Capstone Project

Department of Mechanical Engineering

Final Report

2014

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2 Executive Summary

The engine team is responsible for providing the 2014 MUR Motorsports vehicle an engine system which outputs the required power and torque, through the enhancement of the intake and exhaust systems. On a whole, the system needs to be capable of performing its task reliably at the annual Australasian FSAE competition. For this, the cooling system must dissipate the heat produced by the system, whilst the lubrication system must be able to maintain oil pressure in order to preserve the internal components of the engine. The fuel system is designed to reduce the electrical load of the system and provide more predictable combustion.

The Honda CBR600RR engine has been selected by MUR for its power capabilities, reliability and the extensive knowledge base already developed. The design and manufacture primarily revolves around the intake, exhaust, fuel, cooling and lubrication systems.

The intake manifold is designed to achieve a peak torque of 56.6 Nm at 8500 RPM. Using GT Power as tool to determine performance characteristics, a runner length of 190mm for the outer cylinders were chosen and 180mm for the remaining two. To improve the design 1D and 3D software packages were couple (GT Power/ANSYS Fluent) to obtain details on fluid flow properties in 3D. These results heavily determined diffuser design and a plenum volume of 2.7L which ultimately lead a lower pressure drop between free stream air and the plenum.

The fuel system has been designed to hold 7.5L with a predicted maximum usage of 6.6 L. An internal surge tank was also designed to prevent fuel sloshing and help scavenge fuel under all dynamic conditions.

The Exhaust system was manufactured from 304 Stainless Steel tubing with a wall thickness of 0.89mm employing a 4-2-1 design. The geometries of the the system was determined using 1D flow simulation software, however, ideal results were heavily limited by packaging of the chassis. Additionally, a straight through muffler has been implemented in order to reduce the effects of back pressure and improve engine scavenging.

The cooling system was designed to dissipate the 21.87 kW required. This was accomplished through a cooling system facing spatial constraints and air flow uncertainty introduced by the vehicle aerodynamic package. Heat exchange calculations were developed to model the radiator more accurately and some CFD modelling was introduced to investigate the influence of the shroud and cooling fan.

The lubrication system was overhauled from previous years. The dry sump system was replaced by a wet sump system. A hydraulic accumulator was introduced as a supplementary system in order to slow down any oil pressure drops experienced on track and the amount of routing was considerably reduced. This meant reducing the need for speedflow hoses, reducing mass and improving system maintenance.

Figure 1: Render of Engine Assembly

3 Introduction

The primary objective for the MUR2014 engine team is to focus on implementing evolutionary improvements. This is done by utilising new design techniques and optimising the systems from 2013. The main goals for MUR2014 are:

- **Reliability:** The design for the vehicle should minimise the possibility of failure and have predictable performance characteristics.
- **Performance:** The powertrain of the vehicle should provide a competitive edge, with power maximised from 6,000-9,000 RPM to reduce high end RPM use and improve acceleration out of corners.
- **Driveability:** An engine system which provides experienced drivers with an intuitive driving experience, to obtain maximum vehicle performance.
- **Increase Efficiency:** Designing to increase volumetric efficiency, cooling efficiency, fuel efficiency and routing. The latter case is particularly important for ease of maintenance.

The major subsystems investigated by the engine team are intake, exhaust, fuel, lubrication, cooling, ECU calibration and camshaft design.

Increasing volumetric efficiency is one of the key design targets for the intake system, whilst the exhaust plays an important role in aiding this via the scavenging of exhaust gases.

The cooling, lubrication and fuel systems are important in maintaining a reliable engine system. The cooling system targets the effective dissipation of heat, without compromising vehicle performance through over-designed components. The lubrication system is designed for ease of maintenance and to maintain oil pressure within the system during periods of high lateral and longitudinal acceleration. Similarly, the goal of the fuel system is to ensure continuous fuel scavenging as well as improving fuel economy.

The vehicle is designed in order to compete at the FSAE Australasian competition in December. Team performances are ranked based on a number of static and dynamic events. The static events assess the team's skill in engineering design, cost analysis, and business presentations, and the dynamic events test acceleration, handling and the overall reliability and driveability of the vehicle. Utilising a dynamometer allows these systems to be fined tuned through thorough calibration via the ECU. The primary goals of the Engine team mentioned above revolve around maximising the vehicle's performance at the competition in order to score the highest points total possible.

4 Literature Review

4.1 Intake

In order to grasp the design aspect of Formula SAE it was decided that an excellent way to begin is to learn the basic of a spark ignition engine and the fundamental theory associated with designing a fuel and intake system. Willard (2) provided elementary knowledge with do's and don'ts of simple designs. It failed to acknowledge real systems and how to go about accounting for these losses.

Heywood (1), provided an in depth look at what is happening inside the IC engine. It gives a good development of theory and practice whilst laying a foundation of thermodynamic principals and other relevant theories. He suggests parameters that affect performance such and MEP, spark timing, chamber design and mixture composition and what methods can be used for testing. Although Heywood provides a more in depth design he is still very equation and theoretically based.

Claywell (3) was the foundation where concept selection was done. Claywell and others do comprehensive research into various intake design and reflect on various characteristics. They comprehensively show that a conical spline intake can provide the best performance and tuning capabilities. Claywell concludes that coupling 1D and 3D CFD will benefit students the most as it more accurately predicts the performance of varying intake designs. Although a conical design is preferred there are packaging restraints which may pose limits to performance.

A CFD paper was sourced by Porter (11) who has conducted some basic CFD steady state simulations and demonstrate the effectiveness of various turbulence models. Porter identifies that a side entry intake causes huge pressure losses due to the large bend in the diffuser, whilst also analysing restrictor diffusing angles. Moreover the geometry of these two components have a huge impact on the maximum flow rate into the engine cylinders. Although an excellent starting point, no actual performance or real fluid characteristics are determined.

Based upon Claywell and Porter's research it is evident a coupling of 1D/3D software, specifically focusing on restrictor and diffuser geometry will provide massive increases in torque, power and efficiency. Furthermore, Hamilton's (26) research showed that increasing plenum volume up to 6 times the engines displacement volume can have huge effects on performance. Hence, all three of these characteristics are crucial for an improved design and have been investigated through the coupling of GT Power and ANSYS Fluent.d

4.2 Exhaust

Final reports from previous years may be used (in particular 2013) along with relevant FSAE articles (38), (36), as a guide in order to achieve our goal of maintaining the standard of the existing system. The exhaust is an integral part of the overall engine system and in order to provide a quality engine the exhaust needs to be designed and manufactured accurately.

There are several factors which affect the efficiency and effectiveness of an exhaust system. Certain parameters such as valve sizes, compression ratios, engine speeds are but some of the factors that dictate the efficiency and effectiveness of the system(37). Additionally, the exhaust is linked closely with the intake system and as such needs to be tuned accordingly.

Before any design strategies could be implemented it is important to understand the foundations of IC engines and the basics of each component of the exhaust as well as the engine itself. Willard (2) has delivered the basic knowledge of how the exhaust works and bolstered understanding of how performance can be increased within the engine.

(4) Also tells of exhaust back flow/reverse flow due to intake and exhaust valve overlapping and how higher compression ratios may be able to combat this.
In order for the intake and exhaust to be tuned accordingly (4) and (6) will be used.
Utilising (4) foundational geometries for the system were obtained in order to maximise performance at a specific RPM range

GTI Suite has also been used to enhance understanding of the processes of the engine and has bolstered understanding of the sensitivity of certain parameters and how exactly they alter results. (28) (29) (30)

4.3 Cooling

The requirements of the cooling system is to dissipate the necessary amount of heat so as to maintain optimal operating conditions. This is best approximated as a third of total energy produced (1). With forced convection being the main means of dissipating heat, heat exchanger calculations using the effectiveness-NTU method (34) were used to compute the heat transfer area.

However, if the heat transfer area required is too large for a given expected flow rate of air, a cooling fan can be added to the system. The cooling fan maintains adequate airflow through the radiator core and being thermostatically driven allows for less electrical power usage and some improvement in fuel economy. This is because the engine better maintains an optimum running temperature switching on only when needed, consuming less energy in the process (22).

For a given heat transfer area, the radiator can be even better utilised when mounted with an angle of attack as close to 45 degrees as possible (32). This leads to better heat dissipation as well as taking advantage of the natural convection of air.

Computational fluid dynamic modelling of the radiator, shroud and cooling fan are undertaken in ANSYS Fluent. The radiator is treated as a porous media with the power law model utilised in order to define the cell zone (33). As with any CFD modelling, it is important to have an idea about what values are appropriate and which are misleading. (35) presents pressure drop figures across the radiator of a commercial vehicle and in doing so, provides a sample of what sort of figures would be expected from the CFD modelling undertaken in this report.

4.4 Lubrication

The lubrication system is a vital part of an engine with two functions, which are the cooling and lubrication effect on the moving parts of the engine.

From the specific section of (1), the main purpose of lubrication system design is to ensure the oil pressure is maintained in the oil circuit ensuring there is no engine oil starvation in the crank case for any situation when the car is running. There are two common lubrication systems that are implemented in most vehicles including the Formula SAE; the wet sump system and the dry sump system.

Some basic ideas about dry sump lubrication system design and implementation was gained from (14), (15), (16) and (17). It is conclude that the advantage of implementing a dry sump lubrication system is the relative lower COG, however, extra components are introduced increases the complexity of the system. (18) provides a potential concept of packaging the dry sump lubrication system inside the engine block.

(19) has introduced the design and implementation of a wet sump lubrication system in detail. The wet sump lubrication system is relatively simpler and easier to maintain with less components comparing with dry sump lubrication system, despite the potential rise on COG. Specifically this report made sufficient research on supplementary system design which can resist the oil sloshing issues for the wet sump lubrication system.

4.5 Fuel

From the experience of previous teams since 2009, E85 continues to prove its feasibility in replacing normal RON 98 gasoline in benefiting the team with its flexibility in engine calibration. Using its natural characteristics, custom calibrations are crucial in maximising the output of E85. Ethanol is a compound with low volatility and Reid vapour pressure 3 or 4 times lower than gasoline, which poses a challenge in cold starting (Turner, Pearson, Holland, and

Peck). Complete refinement in cold starting calibration is used to overcome this issue.

The latent heat of evaporation of ethanol is higher which decreases combustion temperature and hence reduces cooling heat loss. As a result Nakata (9) provides further justification for why thermal efficiency and torque can increase. However, given the extra fuel required with blends greater the 20% the volumetric efficiency decreases.

Fuel injection is also a key component of combustion, more specifically atomisation and and mixing are crucial factors in combustion efficiency (Heywood). Spray analysis may be looked into if time permits, and given its potential gains as described in Aoki (10) it is certainly viable to follow up on some sponsors leads. Changing the angle of injection can affect atomisation (1) and will be tested on the dyno.

4.6 Tuning

Most of the tuning will first be simulated using Gt Power. Given the accuracy of last years model, there is confidence that this will work. Using Gamma Technologies resources, (28), (29) and (30), it will be used as the the basis of our tuning. Ignition timing and lambda values will be determined using both design of experiments and optimiser direct.

Creating fuel maps, conducting ignition sweeps, implementing spark advance and choosing a suitable lambda value will be conducted on the Dyno. As recommended from MUR2013 (17) this will be tested on track with the 2014 vehicle and then from their iterate through various selections.

5 Engine

5.1 Summary

The engine is a critical component of any vehicle as it converts chemical energy into the mechanical energy used by the drive-train to propel the vehicle. The internal design of the engine influences its capabilities to produce power and torque, although precise tuning and factors such as the compression ratio and the camshafts used can also have a profound effect.

Relative to the rest of the vehicle, the engine is the heaviest component, and as such has a great deal of influence on vehicle dynamics. Some of the most critical aspects of the vehicle dynamics involves the weight distribution of the vehicle and it's centre of gravity as ideally that is lowest to the ground as possible. All the suspension is set up around these parameters with cornering speed being a particularly key measure of performance.

5.2 Choosing an Engine

To build an engine from the ground up requires more time and resources than are afforded, so the decision is generally made to select a commercially used one. Due to the Honda sponsorship the team have, as well as extensive experience with this particular engine, the CBR 600RR is used. It is a in-line 4 cylinder 599cc engine that falls within the 610cc limit allowed by the FSAE competition.

The only other engines considered were smaller single or twin cylinder engines. This is in keeping with the recent trend in the FSAE competition of using lightweight, lower power engines. However, changing engines is costly as it would require a race engine, a dyno engine and probably a spare to ensure there is cover should an engine issue arise. Coupled with the knowledge base the team possesses for this engine, the Honda CBR600RR was maintained. Please refer to appendix A.3 for more information on engine selection.

5.3 Engine History Spread Sheet

To improve transition between different years, a comprehensive documentation of engine history has been compiled. It has been continued from 2013 and has already provided useful information. It has mainly aided is cataloguing the engine heads, cam cases, chain tensioner cases and engine covers. This is important as each engine head can only be used with its original components. If not any slight variations in bearing finish could result in catastrophic failure. After examining each usable engine, it was found that many engines were unusable as the bearing surfaces had been scored. This could be as a result of mismatched engine components or carelessness in storing engines. These engine have been identified in the catalogue.

The catalogue is also used for noting down when oil has been changed or an oil filter has been replaced. Oil should be changed every 2-3 track days depending on up time.

5.4 Camshafts

Camshafts are an integral component of engine design and is essential for performance characteristics. They allow air and fuel mixture to enter the engine cylinders and exhaust gasses to exit. The CBR600's stock cam profiles are designed such that the peak torque is located at a higher location. The decision to operate between 6000-9000 RPM and increase fuel efficiency has forced us to utilise a different set of camshaft profiles.

The verification of our GT Power model has allowed previous years teams to design CAM shafts. Reviewing both the Wade and 2013 CAM design via GT Valve Train Design software and exporting it to GT Power, we came to the conclusion that the 2013 design is an optimum profile. The key change between the shafts is the reduction in the valve lift. This will increase the mass flow rate at a lower RPM.

The choice to obtain a new pair, was based upon the bearing surfaces of MUR2013's CAM's being scored when placed on an engine head that had slightly scratched surfaces. These surfaces are precision made and hence any imperfections will cause the shafts to rotate of centre which can lead to a lot of issues or even a blown engine. The new CAM shafts have been profiled and is ready to be dispatched. We will test the CAM's thoroughly which will allow the 2015 team more insight to future optimisations.

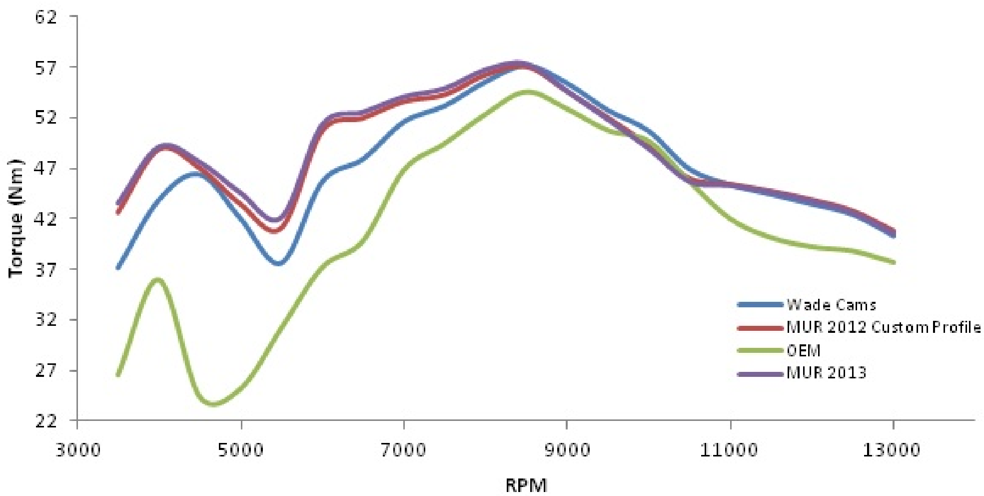


Figure 2: CAM Torque Curve Comparison

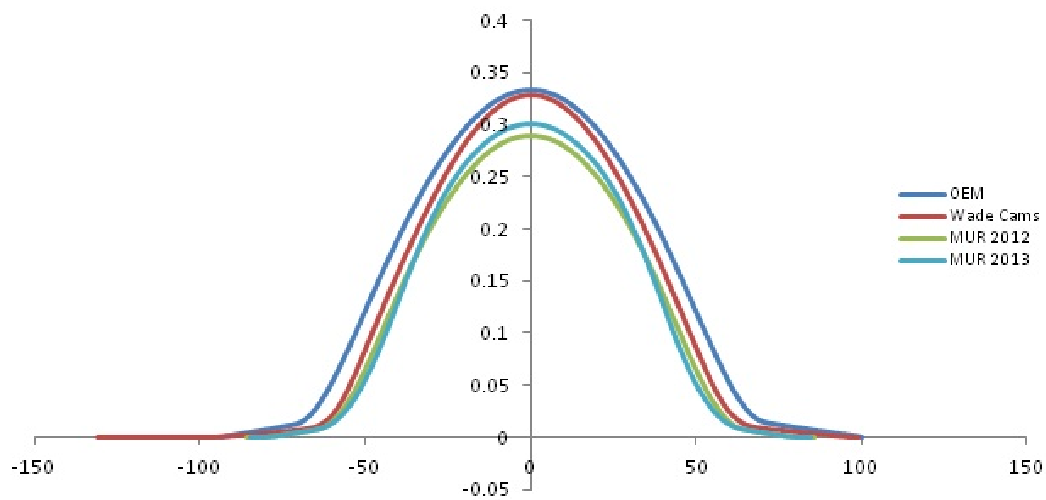


Figure 3: CAM Lift Comparison

6 Engine Simulation and Tuning

6.1 System Overview

The ability to accurately predict the performance of an internal combustion engine is an essential component of engineering design. It allows the team to iterate through various parameters without additional costs. Virtual simulations also allows the development of theory by varying parameters and understanding the effects they have on engine characteristics.

The first means that is used by MUR2014 is creating mathematical models via MATLAB. This type of analysis although has limitations, allowed the team to understand how certain parameters affect performance (see appendix C.2 and F.2). These types of simulations were conducted before the team could understand and use various software packages. In addition, there is an abundance of material that we were able to utilise to help design these codes. The main software package that is utilised by the team is GT Power. GT Power is mainly a one-dimensional solver. The flow model utilised in GT Power involves solving the Navier-Stokes continuity, momentum and energy equations. These quantities are averaged across the flow direction where the whole system is discretised into many volumes. Each flow split is represented by a single volume, and every pipe is divided into one or more volumes. The scalar variables are assumed to be constant and the vector variables are calculated for each boundary.

MUR2014 has taken engine simulation to the next level. GT Power allows co-simulations with various 3D software packages. This type of simulation although computationally expensive, allows more accurate design. The ability to adjust geometry and directly see the effects on engine performance is of extreme value. In particular, it allows us to reduce flow separation, trip boundary layers and evenly distribute flow to the engine cylinders (see section 7.8 & appendix C.11 for more information).

6.2 Engine Modelling

The engine model has been created with the aim to accurately replicate all of the flow components present in the system. MUR2013 showed an extremely close correlation between GT Power and dyno validation. This has allowed this years team to focus more on design parameters as well as introduce the functions of DOE and optimiser into the models. Furthermore, it has allowed us to focus heavily on the optimisation and refining of the new systems.

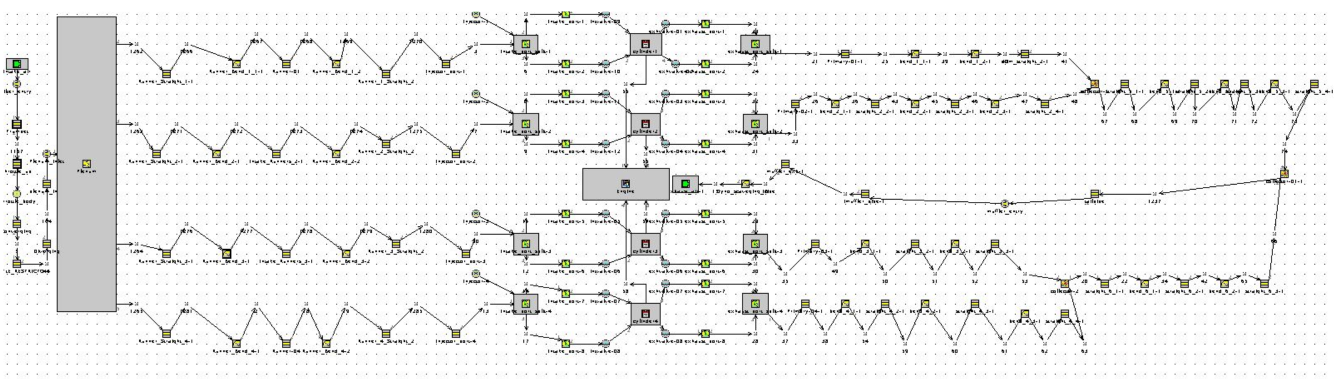


Figure 4: Full Model of Intake, Exhaust, and Engine in GT Power

6.3 Model Validation and Calibration

Although it has been established by MUR2012 and MUR2013 that the GT Power simulation is extremely accurate, changes in components can always have adverse effects and may not be captured. Moreover, tuning the system is crucial as the load the engine experiences in the real world differs. The dynamometer is an excellent means of calibrating the intake and exhaust components.

Two conditions are used to ensure the calibration is performed correctly, wide open throttle and partial throttle. This allows a better resolution in determining the ignition timing and pulse width of the injectors. Acceleration enrichment and accurate fuel mapping is essential. Acceleration enrichment allows the engine to ascertain a better resolution at various throttle positions via the TPS and MAP sensor and hence will prevent any loss in power at various throttle positions. Moreover, the fuel tables are set in discrete increments and hence the ECU interpolates between two adjacent points. With telemetry set up, and quick data transfer, these interpolations will be adjusted on track by fuel trims, altering values or adjusting the graph manually.

As figure 5 shows, the ECU manager is set up to displace both fuel compensation and ignition tables that is over-written at each 500 RPM interval. Different ambient and track conditions will effect the results of tuning. The inclusion a fuel pressure sensor will allow the addition of this parameter into the fuel map. This will reduce the load on the pressure regulator and hence reduce the heat in the system. Adding another variable will smooth out the fuel map making the performance more predictable. Removing heat will also have a similar effect as it will allow more dense fuel to pass through the system.

Figure 5: Dynamometer ECU Screen

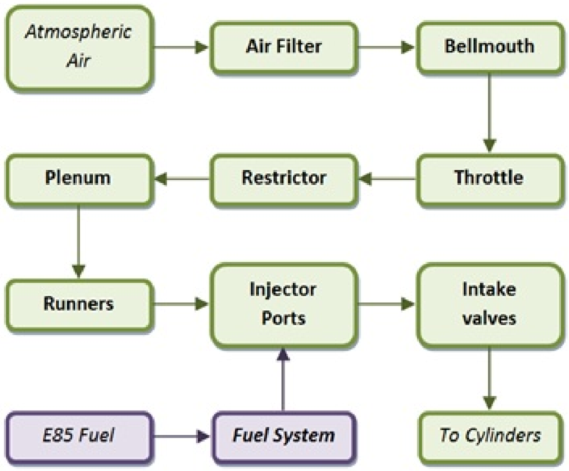
The process of setting up and using the dyno is outlined in appendix B.

This year, there have been major obstacles that have prevented the use of this facility (the predicted plan is shown in the appendix B, table 20). The team is still very hopeful that the testing and tuning will not be affected. Some key areas that are new to tuning is validating GT Power's capability to predict the effect of plenum volume. With MUR2013's intake manifold at 2.0L, a 2.7L and 1.8L will be tested this year.

The team has only been on the dyno a few times with many of these instances involving debuging. The last session showed promising results, however the torque output will need to be calibrated so that it reads realistic results. After this issue is sorted out, MUR2014 should be operating the dyno 2 -3 times a week.

7 Intake System

7.1 Summary



(a) Intake System Flow Chart

(b) Intake System Render with Injector Ports and Injectors

The role of the intake system is to take a desired flow rate of atmospheric air, mix it with the appropriate amount of fuel for combustion, and deliver it to the engine cylinders during their respective intake strokes. The flow rate of air is controlled by the throttle, which provides a variable flow restriction actuated by the driver. A key component of naturally aspirated engines is increasing volumetric efficiency (VE) to obtain better overall torque and power. Below is a list of the major design components that effect volumetric efficiency.

Fuel Type	Engine Speed
Mixture temperature as influenced by heat transfers	Intake and exhaust manifold port design
Ratio of exhaust to inlet manifold pressures	Intake and exhaust valve geometry, size, lift and timing
Compression ratio	

The 2014 intake manifold is designed to aid pressure recovery and ultimately achieve a higher mass flow rate to the engine cylinders. Sharp geometric changes were avoided and CFD analysis was conducted to determine the flow characteristics. Using CFD is a key advancement in 2014 and has helped recover losses in the system and ultimately increased the VE. Although it can be computationally expensive and requires a big time commitment, it's benefits will take MUR to the next level of engineering design. It will also allow the solidification of fluid theory juxtaposing real world applications. Below is a table of the major design specifications for the intake system for MUR2014:

Air Filter	K&N RC-9870 conical cloth filter
Throttle Body	AT Power 28mm aluminium shaftless butterfly throttle with built-in restrictor and bellmouth
Restrictor	Venturi-type with converging half-angle 6° and diverging half-angle 3.1°
Diffuser	Rapid prototyped ABS, 131.2mm length with a 23.32° angle
Plenum	Rapid prototyped ABS conical-spline plenum with volume 2.72 L
Runners	Rapid prototyped ABS curved runners with length 180/190 mm and diameter 34 mm
Injector ports	Rapid prototyped ABS with injection angle 30 deg, 10 mm above cylinder head inlet and OEM Honda rubber couplings to cylinder head

Table 1: Intake System Specifications

7.2 Design Constraints and Considerations

The intake system this year was designed to obtain an even distribution of flow through each runner, whilst increasing the VE of the manifold . With a proven 3D printed intake manifold, a conical spline intake concept was used to reduce cylinder-to-cylinder volumetric efficiency imbalance as well as providing an improved acoustic content. This evenly distributed flow allows a better trapped air-fuel ratio (AFR) which makes muffler design, tuning and ECU calibration much easier. The top of the air filter is positioned just underneath the top-right corner of the main roll hoop, in order to package around the large rules-compliant head restraint and fit within the roll plane defined by the main roll hoop and the outer tyre surfaces (see appendix D.1).

The restrictor throat diameter is predetermined by competition rules. Choosing E85 as our fuel (see secction 8.2) mandates us to utilise a 19mm restrictor where all air must pass through and must be downstream of the throttle body. The restrictor is incorporated within the throttle body.

7.3 Air Filter

Although not a critical design component, the air filter can provide extra gains in the efficiency of the combustion cycle. In previous years a foam filter or a K&N filter has been used. The K&N filter selected however has a chrome cap that interferes with free stream air. A performance foam filter has been selected to compare with the K&N. The test will compare the MAP pressure under the same conditions. By increasing the surface area of the filter there should be no barrier and therefore possibly eradicating any minor losses. Moreover, the foam filter should also allow the pressure at the inlet to be as low as possible. The increase in velocity allows more air to reach the plenum and hence increase volumetric efficiency. MAP pressure will allow an accurate comparison as it can be used to calculate the mass of air available to the engine.

The filters shown in figure 7 show the various filters that MUR2014 are considering to use. The filter on the left is the same as MUR2013 and it will be used as the control. The Pipecross filter has a smaller filtration diameter, however it is 30mm longer. Adjustments have been made to incorporate this filter which has forced a reduction in plenum volume. The third filter has extremely large diameters and will only be tested if sponsorship is received or if spare cash is available. With the delays on the dyno, the team have been unable to validate the differences however, the results will be recorded for future years.

Figure 7: K&N Filter, High Performance Foam Filter & VPW Inverted Air Filter

7.4 Throttle Body and Restrictor

The driver feedback from both competition and driver training days was that a smaller throttle body required more actuation and hence allowed them to ascertain a better resolution between throttle angle and RPM. This is a crucial factor as MUR2014 have tuned both the intake and exhaust to operate between 6000-9000 RPM. In comparison drivers found there was very little difference between a small throttle position and wide open throttle when a larger throttle body is used. The dyno also showed that around a throttle position of 10%, the engine was effectively at wide open throttle. The restrictor and throttle body is also one unit which is why the team has elected to reuse the 28mm throttle body.

A 2D axis-symmetrical simulation was run to determine the pressure loss across the restrictor. This was done in conjunction with a variety of sources that have conducted similar simulations and experiments, (see appendix C.8).

By calculating the mass flow rate that is required for choked flow (see appendix C.4), it was possible to determine that the current restrictor has a pressure loss of approximately 4590.7 Pa. Coupled simulations were run in order to determine which conditions best suited the flow through the restritor. Table 3 shows that the 2014 plenum has the superior pressure recovery. This design had a larger plenum volume and a smaller diffuser (see table 1).

Reducing the pressure drop at the restrictor is the first step to obtain a higher MAP pressure and ultimately create the highest pressure drop between the manifold and the engine cylinders. The mass of air in the plenum is calculated based upon the pressure in the plenum. Ultimately a higher pressure means that engine has a larger reservoir of air available which results in a higher torque and power output. Section 7.8 demonstrates this effect and provides VE and other performance comparisons.

Mass Flow Inlet	0.0647	kg/s
Pressure Inlet	4790.04	Pa
Pressure Outlet	199.7	Pa
Velocity Inlet	14.65	m/s
Velocity Outlet	18.5	m/s
TKE Inlet	0.805	m2/s2
TKE Outlet	30.87	m2/s2
Min Turbulent Viscosity	2.16E-08	kg/m-s
Max Turbulent Viscosity	0.119	kg/m-s

Table 2: 2D Axis-symmetric Simulation Results

	Average Total Pressure (Pa)	**Average Static Pressure (Pa)**
2D Fluent Simulation	4590.7	4658.08
2013 Coupled Simulation	2195.6	4760.6
2014 Coupled Simulation	1347.9	3572.3
2013 Coupled Simulation With Slot	1645.3	3706

Table 3: Pressure Loss Comparison between Fluent 2D and Coupled Simulation

7.5 Diffuser

The diffusers main purpose is to reduce the pressure loss after the restrictor and provide beneficial flow dynamics to the plenum. It is essential for the performance of the engine that this pressure drop is minimised. Information regarding diffuser losses is very minimal however, some understanding was obtained through thermodynamic and fluid mechanic internal pipe flow theory. Sparrow (27) shows how different combinations of diffusing angles and Reynolds number effect flow separation. Different analyses were conducted to reduce separation which, effectively will decrease the pumping effort required by the engine and the cylinder to cylinder imbalance.

Steady state simulations were run utilising a velocity inlet condition of 83.2 m/s obtained from GT Power and a pressure outlet condition with an initial guess of atmospheric pressure (i.e. 100% VE/efficiency) . These results showed no flow separation (see appendix C.9). A coupled simulation was run on the 2013 design as shown in figure 8a. It is evident that a build up of TKE along the bottom of the diffuser contour which justifies the severe case of flow separation seen. The difference between these results is mainly due to the transient behaviour playing a major factor in the performance of the diffuser and hence affirming why accurate transient simulations are required.

Looking at the GT Power results on flow rate through to the runners (figure 8b) it is possible to see that there is a difference of 12% between port 1 and 4. This imbalance is caused by the angle of the diffuser from the centre line of the plenum and hence must be reconciled. The noticeable build up of turbulence and flow separation is also influenced by both the diameter and length of the diffuser. Rotating the diffuser will force us to make it smaller in order to meet competition rules which may negate any improvements.

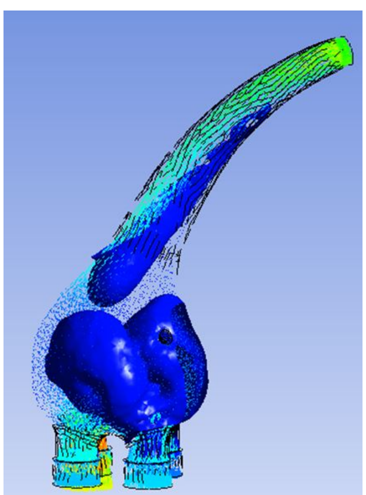
The 2014 design utilises a smaller diverging angle and an outlet diameter of 32mm. This design to go smaller was based upon the relationship that Reynolds stress and diverging angles have with flow separation (see appendix C.9 for more information on effects of Reynolds Number).

This balance of length and diameter versus separation was addressed and its improvements on the overall system is shown in tables 4, 5 and 6. The 2014 diffuser has a smaller outlet diameter and diffusing angle, whilst a 0.82mm protruding surface with a 2mm fillet was placed at the entry of the 2013 simulation with a slot (figure 88). The 2014 diffuser was designed to reduced the rapid expansion and hence allow the air to stick to the contours of the bounding wall. This design achieved a 20% reduction in pressure loss and increased volumetric efficiency (see table 11). The inclusion of the slot also decreased the pressure loss by 82%.

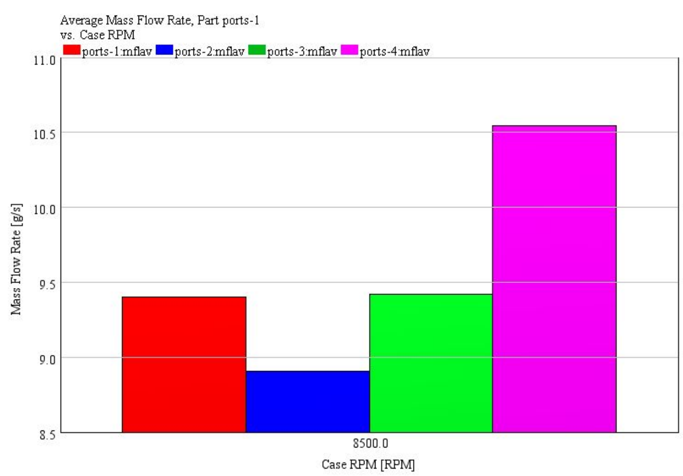
Adding the slot at the entrance of the diffuser was an attempt to reset the boundary layer and decrease flow separation. Although the results show that this design can improve the pressure recovery substantially, it will need to be validated on the dyno.

2013 Coupled Simulation			**2013 Coupled Simulation With Slot**		
	Min	**Max**		**Min**	**Max**
Inlet	-10480	4983	**Inlet**	-2261	-1148
Outlet	-4726	-3687	**Outlet**	-1845	-1032
Max Pressure difference	6793		**Max Pressure difference**	1229	

Table 4: 2013 Diffuser Pressure Drops



(a) Velocity vectors overlay with turbulent kinetic energy



(b) Average mass flow rate at ports before entering runners

2014 Coupled Simulation		
	Min	**Max**
Inlet	-9389	-7556
Outlet	-6332	-3891
Max Pressure difference	5498	

Table 5: 2014 Diffuser Pressure Drops

	Average Pressure Inlet (Pa)	Average Pressure Outlet (Pa)	Pressure Difference
2013 Simulation	-3193.11	-4074.2	-881.09
2014 Simulation	-8168	-4334	3834
2013 Simulation With Slot	-1533	-1414	119

Table 6: Diffuser Average Pressures

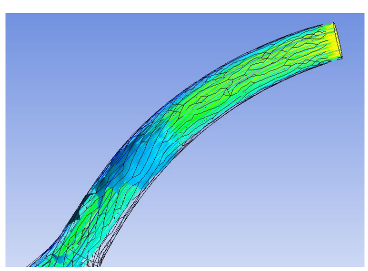
To accurately see the effects of the diffuser more analysis was conducted on the 2013 system as accurate dyno results are available. Two simulations were run to see the effect that area ratio has on divergence. Both simulations had an inlet of 32mm, however the first simulation had a slot at its entrance and an outlet diameter of 40mm. The second simulation had no slot and an outlet of 48mm.

	Torque (Nm)	Power (kW)	Net IMEP (bar)	Gross IMEP (bar)
2013 Slot 32mm to 40mm	56.5	50.3	13.6	14
2013 No Slot 32mm to 48 mm	56.6	50.4	14	14.4

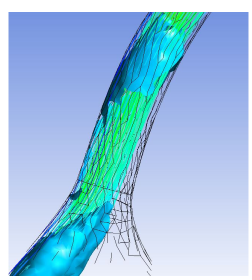
Table 7: Comparison of Small Changes in Diffuser Diameters

The torque output with the slot should show the largest increase in torque, however table 7 shows that this isn't the case. From figure 9a it is evident that the flow has more of a tendency to stick to the surface of the diffuser. Therefore the reason for the loss of torque is due to the diffusing angle not being large enough, which causes the flow to rapidly expand at the exit. This expansion increases the momentum of the fluid and hence it is able to overcome the pressure that is pushing it. This results in turbulence as shown in figure 9b. In order to create better flow characteristics, the development of large TKE must occur near the cylinder ports. Figure 10b shows the development of significant TKE (15 J/kg) in the plenum rather than at the exit of the diffuser. As illustrated in appendix C.9 the area ratio has an influence on the characteristic of the flow. Looking at equation C.17, it is evident that the velocity must be maintained throughout in order to keep the loss coefficient K_d low. These results strongly confirm that the diffuser has a significant impact on fluid characteristics and hence engine performance.

Figure 9: Velocity Stream Line Overlay with TKE Diffuser with Slot

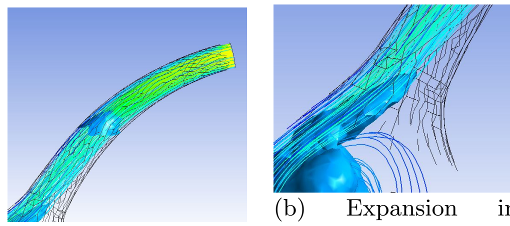


(a) Flow Separation in Diffuser with Slot



(b) Expansion into Plenum in Diffuser with Slot

Figure 10: Velocity Stream Line Overlay with TKE Diffuser with No Slot



(a) Flow Separation in Diffuser with No Slot

(b) Expansion into Plenum in Diffuser with No Slot

These improvements have increased the torque output and has provided future years with the ability to implement parametrisation in Fluent to even further increase the engines performance.

7.6 Plenum

The plenum serves an important factor in resonant charging and the overall peak power and torque curves. Its design is critical in determining the turbulent effects and losses in the system. Larger plenum volumes generally are beneficial for the system, but results in a decrease in throttle response . Using some simple calculations, (see appendix C.2) a decrease in throttle response was determined to small in comparison to the potential gains; a maximum delay of 1.5ms in the designed operating range. Due to 3D printing and dyno facility delays, this effect has not been tested. However, with the onset of sponsors this will be tested on the 2014 vehicle.

Theoretically, the largest plenum volume should provide the greatest average pressure throughout the intake valve open period. The pressure profile that is generated should result in beneficial wave dynamics that occurs in the larger plenum chamber creating better resonance charging. The remanding cylinders may cause interference waves that reduce the strength of the compression waves in the primary runner, hence negating the beneficial effects of resonant charging. With larger plenum volumes the reflections of the expansion waves are not as dominant than those in the smaller plenums and therefore do not degrade resonance charging (see appendix C.2 for information on resonance charging).

Using GT Power, an analysis was conducted to determine the effect that the plenum volume has on performance (figures shown in appendix C.6.1). A close up of the torque curve is shown in figure 11. A plenum volume of 3.0L produces the best improvement over the whole designed operating range with a slight decrease at higher RPM. The 4.8L provides the greatest increase, however it tappers of really quickly. A comprise was made to construct a plenum between 2.4-3.0L to obtain optimal results of both variations. A more in depth look at the effect of our chosen volume selection is shown in appendix C.6.1

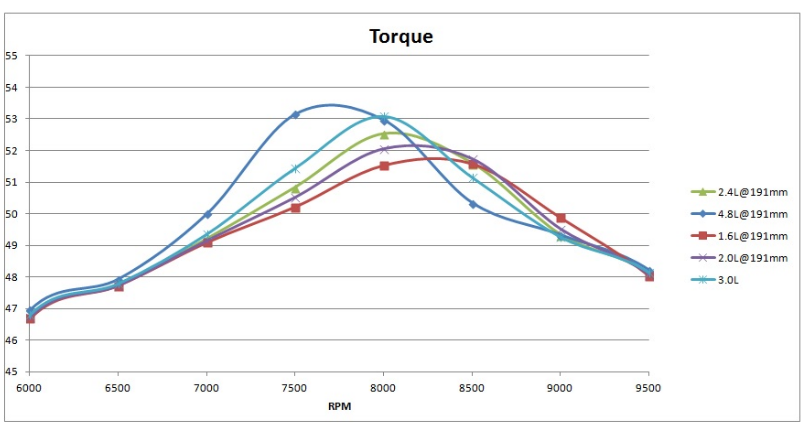


Figure 11: GT Power Results for Torque with Varying Plenum Volume

7.7 Runners

The runner design was selected using GT-Power results as well as Helmholtz equations as preliminary proof of concept (see appendix C.2). The runner geometry is vital in determining the resonant frequency at which the pressure waves superimpose and hence create a maximum torque region. A variety of runner lengths were run in GT Power to obtain torque and power curves (see appendix C.6).

The length of runners are the key feature that determines the location of the torque curve peak as described in appendix C.2. The choice to increase the runner length was based upon our GT power results in figure **??** which showed an increase in torque across the operating range. Table 8 illustrates that between 7000-9500 rpm there is dramatic increase between 180mm and 190mm. Knowing that our average rpm is around 8300 and the the introduction of an aerodynamics package a 190mm length seemed the most viable.

The new aerodynamic features is an area of uncertainty. With no track data and the potential of loss in acceleration at higher speeds due to drag is an area of concern. Looking at the GT Power torque curves it does not look like there is a way to adjust the runner length without accepting a massive reduction in torque. A way around this is maybe to change the fuel MAP at higher RPM to increase the torque output at those points. This will be done on the dyno and tested out on track for verification.

Length(mm)	**170**	**180**	**190**	**200**	**210**	**220**
Average 7000-9500 (Nm)	52.0161	52.86645	53.14185	53.36063	53.51275	53.57983
Average 6000-9000 (Nm)	50.24534	51.20087	51.40221	51.57506	51.78681	51.98501

Table 8: Average Torque vs Length

7.8 Coupled CFD Simulations

The coupled simulation allows the user to utilise both 1D and 3D flow equations. 1D CFD simulations are able to provide essential information on the system level, in particular performance characteristics. It is however, limited in its ability to capture information about the fluid and give a full description of the flow properties (see appendix C.11).

	Torque (Nm)	**Power (kW)**
2013 GT Power Results	57.09	50.822
2013 Dyno Results	55.5	49.43
2013 Coupled Simulation	55.6	49.5
2013 Coupled Simulation With Slot	57.2	50.9
2014 Coupled Simulation	56.6	53.3

Table 9: Torque and Power Results from Simulations

	Error Torque (%)	**Error Power (%)**
GT Power	2.785076195	2.738971312
Fluent/GT Power Coupling	0.179856115	0.141414141

Table 10: Torque and Power Error Comparison with and without coupling

Table 3 illustrates the effect of isolating components in analysing the manifold. The Static pressure results are fairly close to the predicted 2D model in some circumstances. However, given the wave dynamics and development of velocity boundary layers due to moving engine cylinders, the total pressure loss differs. The dynamic conditions are affected by the surface friction, heat transfer and loss coefficients. The improvement of pressure loss upstream signifies a reduction in overall system losses and hence an improvement in torque. Appendix C.8 summarises the methodology utilised and provides more in depth analysis.

1D/3D coupling has been a key advancement in 2014 and has been a major goal that has been achieved by the engine team. Although the team has so far only achieved a 3.5% increase in torque and a 2.75% increase power, these results are promising for future years. It will allow future MUR students to apply more fluid theory and understand boundary layer formation which will allow for better testing methods to be implemented. With testing is yet to be completed on the dyno, the results gathered will help in model tuning and obtain a better understanding of how these losses effect different tuning parameters. Using last years results, the error in the engine model has been reduced such that it can be neglected and can be accounted for by various tuning parameters and natural wear and tear in components such as bearing surfaces and fuel injectors. Obviously, more validation needs to be conducted to verify the model over the entire RPM range.

	Air Flow (kg/hr)	**VE**	**Gross IMEP 360**	**Gross IMEP 720**	**BMEP**
2013 Coupled Simulation	165.08	0.895237	14.3081	13.8263	11.6483
2013 Coupled Simulation With Slot	171.045	0.927584	14.6211	14.169	11.991
2014 Coupled Simulation	170.152	0.922742	14.4566	14.0034	11.858

Table 11: Performance Results from GT Power/Fluent Coupling

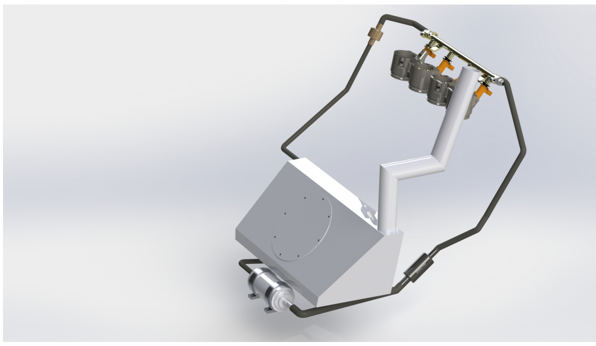
The improvements in the design is apparent when looking at the results in table 11. The increase in both air flow and VE shown is key in illustrating the importance of coupling simulations. Other than torque and power, IMEP (Indicated mean effective pressure) and BMEP (brake mean effective pressure) are key performance characteristics. The development of the new intake manifold shows the potential of the engine to do more work and hence pump fresh air in and remove exhaust gases through the increase of IMEP (See Appendix C.6).

The key area of analysis has been conducted on the diffuser of the manifold. The results obtained substantiates the claim that pressure recovery in this region is crucial for obtaining a higher VE and therefore achieve an increase in torque and power. Various other geometries must simulated to obtain an optimum result. The computational expense can be reduced by implementing a better mesh with fewer elements and running the simulation with parallel proccessing.

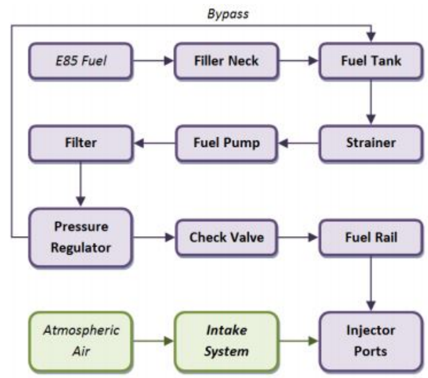
8 Fuel System

8.1 Summary

Figure 12



(a) Full Fuel System



(b) Fuel System Flow Chart

The objective of the fuel system is to store and provide fuel to the engine at a designed pressure and operating conditions. The engine team have the choice between E85 and RON 98 fuel to use at competition (see section 8.2). With a small chassis and even less room than previous years, the team has decided to stick with an external fuel pump and regulator. MUR2014 have also chosen to design a system that has the ability to utilise internal baffles if sloshing becomes an issue with the increased lateral and longitudinal forces. The system has side and front walls that all angle towards the pick up region and hence depending on dynamic testing, the baffles could be left out hence saving some weight and keeping the vehicles CoG to a minimum.

The focus of the fuel system for MUR2014 is to ensure that a lighter system is produced, avoid fuel starvation and increase fuel atomisation.

Fuel Type	E85
Fuel Tank	Custom laser-cut and welded 2mm aluminium tank with cross baffles
Fuel Pump	Off-shelf Bosch 69608 OEM replacement 92 L/hr external in-line pump
Fuel Pressure Regulator	Off-shelf OEM Bosch 100 L/hr external adjustable regulator
Filters	100 µm pre-pump strainer in-tank & Speedflow 10 µm pre-injector EFI filter
Injectors	NipponDenso 12-hole 220 cc/min (Orange, stock from CBR600RR
Lines	Speedflow 200 series stainless steel braided Teflon hose and fittings
Fuel Rail	Stock Honda CBR600RR 2003/04 secondary fuel rail

Table 12: Fuel Components and Specifications

8.2 Choice of Fuel

A preliminary analysis was conducted on potential engine performance, thermal efficiency, emissions and cold starting. The decision to stick with E85 can be summarised by table 13.

The high octane number of E85 will be beneficial with the rebuilding of our higher compression engine. The increased compression ratio allows more flexible tuning with spark advance to obtain a higher torque output. The charge cooling that E85 has results in a decrease in compression temperature during the compression stroke. Furthermore, the decrease in combustion temperature reduces the heat loss and hence achieves a higher thermal efficiency. Volumetric efficiency decreases as the air-fuel ratio is lower. This extra fuel, approximately 40% more, does hinder design as a larger fuel tank is required; however a theoretically power increase of 11.6% (see appendix D.4) with optimised ignition timing is acheiveable. Ethanol has no volatile particles and hence has a fixed boiling point of approximately 78.5 ^{o}C. MUR2014 have created a cold start fuel MAP where the team increased the amount of fuel and delayed the spark.

Advantages	**Disadvantages**
High octane number. This anti-knock property is in part a result of higher flame speeds	Decrease in VE
Utilise a higher compression ratio	Higher specific fuel consumption
Lower Emissions	More difficult to cold start
High evaporative cooling	Lower energy density
Easier to tune	More corrosive
Increases thermal efficency	Potential for vapour lock

Table 13: E85 Advantanages and Disadvantages

8.3 Fuel Tank Design

The fuel tank shape and size is crucial in ensuring constant fuel pressure and avoiding fuel scavenging issues. With the addition of the Rob Black sponsorship MUR2014 were able to get an optimised tank welded as well as machining the pick up component and fuel bungs at Holmsglen (see figures 14a and 14b).

The first step in the fuel tank design is to identify available design space. MUR2014 have chosen to place the fuel tank behind the seat as it can be protected by a firewall and the chassis. Based on competition data from last year, a maximum of 7.5 L of fuel is required to finish the endurance event. At competition, the worst set up for fuel consumption was run; that is, no endurance fuel MAP, (lambda = 0.9) and being restricted to second gear. This combination of high revs and maximum fuel consumption still left approximately 200ml. The team has elected to stay with this volume as fuel MAP will be created or fuel trim will be implemented in this years system.

In-tank pumps are out of the way and sometimes benefit from cooling provided by the fuel, however they are also difficult to access if repairs are required. The pump must also be E85 compatible which is quite rare for OEM automotive pumps. The additional complexity of electrical wiring and components inside the fuel tank supported our decision to stay with the external module.

It was decided to have the fuel pickup for the external pump at the rear of the tank with sloping walls at the front and sides encouraging fuel to flow down and cover the designed pickup. This configuration ensures that under dynamic conditions the pickup should naturally be covered even at low fuel levels due to the sloping hydrostatic level line under acceleration. An internal surge tank was designed so that it surrounds the pickup, featuring one-way swing valves or trapdoors to allow fuel into the surge tank and then seal it inside. Internal baffle plates have also be designed, which will allow direct flow towards the pickup in dynamic sloshing conditions to ensure as much as possible that fuel does not take an easier flow path around the surge tank, rather than entering through the trapdoors.

The tank is made from 2mm thick 5052 Aluminium. 5052 was selected for its corrosion resistance. There was a 2mm restriction due to ease of welding. This still resulted in a system that is 758g lighter than last year.

8.4 Fuel Lines

Speedflow 200 Series Teflon hose was selected for use in the fuel system, as it is one of the lightest hose in its range. It is comprised of a strong stainless steel external braid, and a teflon anti-static coated inner tube. This series is ideal for use with alcohol and unleaded fuels. Size -6AN hose was chosen as it was the smallest diameter which has the anti-static coating required for use with E85 and unleaded, and results in a line velocity of approximately 500mm/s (for calculations see Appendix D.3). It produces a Reynolds number under 3000 and therefore is reasonably laminar. This hose is only compatible with Speedflow 200 series hose-ends and fittings.

The team has also been limited to the lines that MUR2014 currently possess as a $1200 deficit from last year was imposed due to MUR2013 being unable to fulfil their sponsorship deal. This has hindered this MUR2014's ability to procure any new items and as a result the budget is extremely limited.

8.5 Fuel Rail and Injectors

MUR2014 has chosen to stick with using the secondary fuel rail on the 2014 vehicle. This allows us to utilise the bolt on flanges which will allow the team to test different fuel injection angles on dyno as well obtain new injectors depending on the procurement of a new sponsor. The system has one line from the fuel tank entering one side of the rail and the regulator and a bypass line exiting back into the top of the tank on the other side (see Figure 12a).

The fuel system operates around 3.5 bar which produces 220cc/min with the stock injectors. A lot of emphasis has been put out using mathematical models to determine the effect the flow rate has on the system and hence determine fuel requirements, tuning aspects (pulse width modulation) , and analyse the legitimacy of new fuel injectors. Various Matlab codes are used to determine the mass and flow rate of the fuel. We will not require more than 35 L/hr (see apendix D.2 for more calculations). Two different mathematical models were utilised to determin the maximum fuel flow rate. These methods did yield two different results. The first method didn't include pulse width within the calculation. It is determined by utilising an idle point and hence underestimates the fuel consumption (see appendix D.2).The effect that pulse width (PW) has on various parameters was also looked analysed. The model predicted an operating pulse width PW between 38ms to 17ms.
Utilising the the information from the models, it was determined that the injectors are adequate for the operating conditions. However, they also show that air-fuel mixing is a key design constraint of fuel injection systems.

The required characteristics of a port fuel injection system are:

Accurate fuel metering	Small sac volume;
Desirable fuel mass distribution pattern for the application;	Good low-end linearity between the dynamic flow and the fuel pulse width;
Minimal spray skew for both sac and main sprays;	Small pulse-to-pulse variation in fuel quantity and spray characteristics;
Good spray axis-symmetry over the operating range;	Minimal variation in the above parameters from unit to unit.
Minimal dripage and zero fuel leakage, particularly for cold operation;	

Table 14: PFI system Requirements

To promote fuel-air mixing, MUR2014 sought to create a swirl generator or procure a swirl-type injector. In the swirl-type injector, the pressure energy is effectively transformed into rotational momentum, which enhances atomisation. The mass distribution of a swirl-type injector is generally more axis symmetric. Increasing the swirl in the injector promotes fuel air mixture through the injector port. The toroidal vortex ring that is initially generated near the injector tip, grows larger as it moves away from the injector and is responsible for better mixing. This increased mixing of fuel-air mixture is why this type of injector will help balance the combustion process, making tuning easier as well as potentially reducing fuel consumption (5). This type of swirl generator is preferred over the swirl port design (see figure 13). The swirl type injectors will not restrict the flow of air into the cylinder which will influence the power and torque curves. However, these designs will hopefully be compared if funds/sponsorship is obtained. It will also be tested on the dyno before the end of the year.

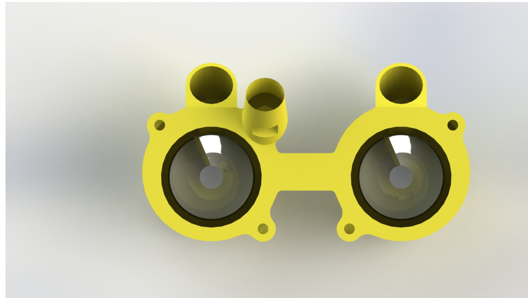
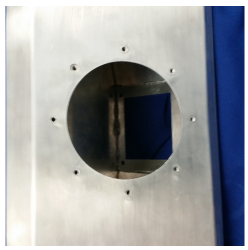
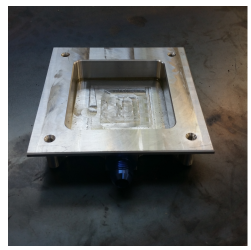


Figure 13: Proposed Swirl Ports



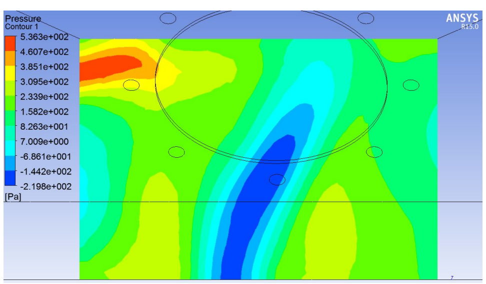
(a) Fuel Tank



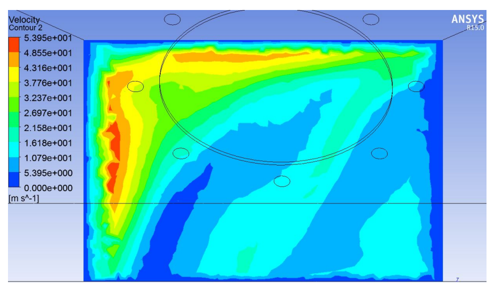
(b) Fuel Pickup Render

8.6 Fuel Tank Simulations

Figure 15: Fuel Tank Simulation Results



(a) Pressure Contour at 2Gs



(b) Velocity Contour at 2Gs

A transient fuel tank simulation was run using CFX's buoyancy model with thermal properties enabled. Ideally the results of the simulation would show the hydrostatic pressure gradient under the extreme condition of 2Gs. However, a contour plot was created on a plane near the pick up to show the fluid properties. Figure 15a shows a high pressure region in top left corner, however a low pressure does enclose a section of the pick up. The low pressure does signify sloshing which is also shown in the velocity contour plot in figure 15b. The sloshing region is occurring around the pick up region which may cause some scavenging issues at a low levels of fuel. Given this simulation is conducted at prolonged high forces, this is a situation that is not a concern. These results do support the idea that baffles or a surge may not be required.

These results are preliminary and need fine tuning. Fuel tank results will be logged for future years so these simulations can be further developed to acertain accurate results.

9 Exhaust

9.1 Summary

The objective of the exhaust system is to efficiently expel the gaseous products from the engine cylinders that result after combustion. This is done by utilising various tubes of specified lengths and diameters that create pressure and expansion waves. The 2014 exhaust system is designed to scavenge exhaust gases at the designed operating RPM range. This will allow fresh air to fill the engine cylinders and therefore maximise the VE, power and torque outputs.

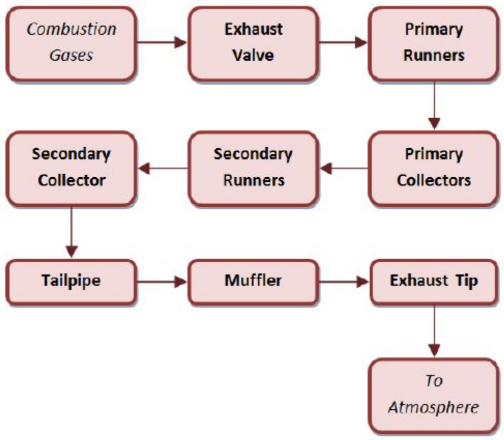
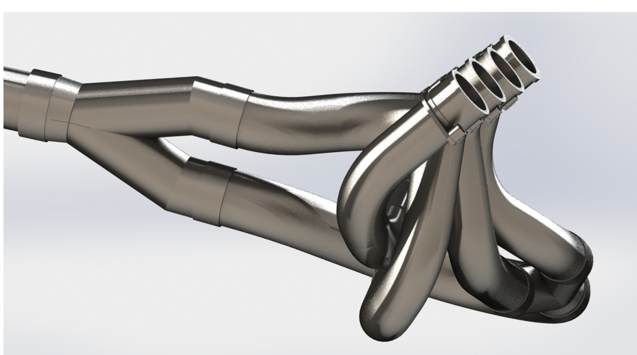


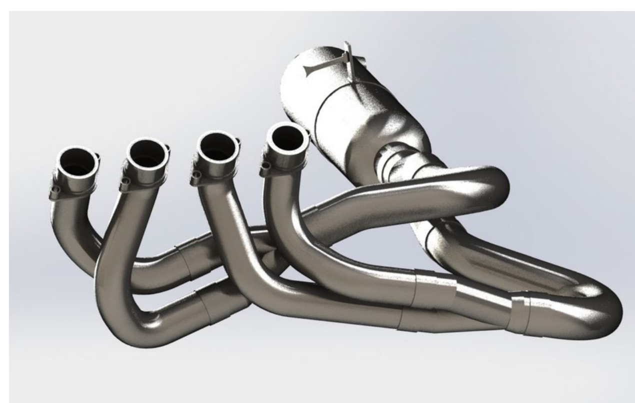
Figure 16: Flowchart of the Exhaust system

A 4-2-1 design has been maintained from the previous year. This system provides a broader power band than the 4-1 system, as well as being more beneficial for mid range RPM torque.

Optimal performance coincides with the RPM range that is operated in most, as well as benefits in fuel consumption with this configuration. Despite packaging constraints ultimately being a limiting factor, the more complex arrangement was selected in order to maximise performance.



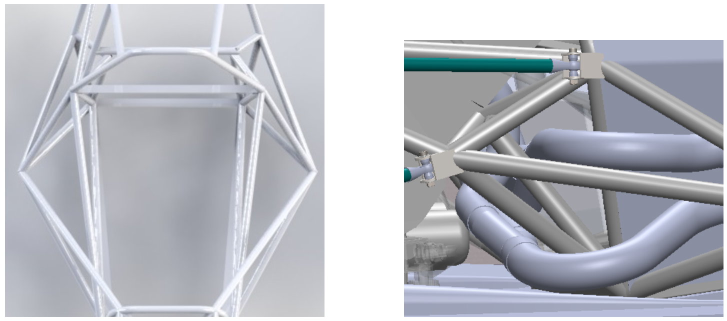
(a) Side View of Exhaust



(b) Rear View of Exhaust

9.2 Design Constraints and Considerations

The aim of the 2014 exhaust was to optimise for a specified RPM range, whilst also being able to manufacture a system that suitably fits in the chassis. The chassis this year was designed with out-riggers, protruding tubes, that aid the torsional rigidity of the vehicle but consume a large portion of the design space available. This can be seen in figure 18b.



(a) Chassis Outriggers (b) Exhaust with undertray

Additionally, the introduction of an aerodynamics package, has provided some uncertainty in how engine performance is affected. The system has been designed to account for this change. Even though the exhaust has been designed towards a specific power and torque output range, some flexibility is still retained through the intake manifold and tuning.
The under-tray has provided an additional volume constraint and has required the exhaust to be designed with minimal clearance between the out-riggers. The heat radiated from the exhaust towards the composite based components can be detrimental, as the resin used on the carbon fibre has a lower tolerance to heat than that used in previous years.

Competition rules also mandate that the gases produced are to be directed away from the driver and the noise level produced to be below 110 dB.

9.3 Runner Length and Diameter Selection

Runner length and diameters are crucial design parameters that are selected based upon the pressure pulses that are generated. The geometries of the tubing enable us to manipulate the torque and power outputs such that they reflect the overall team goals.

Utilising basic equations (see appendix E.2) to design for a peak torque of around 8500 RPM, provided a foundation for preliminary values to be utilised in GT Power. The results from the equations are shown in table **??**.
In order to determine the ideal pipe specifications, however, MUR2014 utilised GT Power to obtain performance characteristics. Setting up a DOE in GT Power allowed for iterating through large ranges of runner lengths and diameters so the team could observe the effect of varying parameters on overall engine performance.

RPM	6500	7000	7500	8000	8328	8500	9000
P	654.5385	602.3429	557.1067	517.525	494.141	482.6	451.5556
P1	381	381	381	381	381	381	381
P2	273.5385	221.3429	176.1067	136.525	113.141	101.6	70.5556
P2-ID	24.339	25.2577	26.1443	27.0017	27.5496	27.8327	28.6396
CL and TL	730.7385	678.5429	633.3067	593.725	570.341	558.8	527.7556
CL	2.9257	3.0362	3.1428	3.2458	3.3117	3.3457	3.4427
TL	727.8127	675.5067	630.1639	590.4792	567.0293	555.4543	524.3128
TL-ID	32.7814	34.0189	35.2129	36.3677	37.1058	37.487	38.5738

Table 15: Initial Geometries for different optimizing RPM ranges

Below are the ideal geometries which enabled an exhaust to be designed that fits within the chassis.

	Length (mm)	Diameter (mm)
P1	250	35
P2	250	35
P3	250	35
P4	250	35
S1	400	45
S2	400	45
TL	570	40

Despite analysing the efficiency that a tailpipe would add to the system, an ideal geometry has not been implemented as there is physically no space within the constrained design volume to allow for the 600 mm length required.

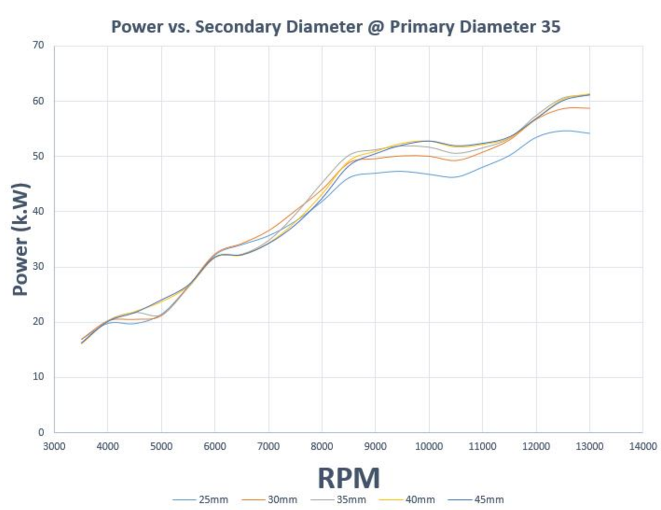


Figure 19: Power vs Secondary Diameter with Primary Diameter of 35mm

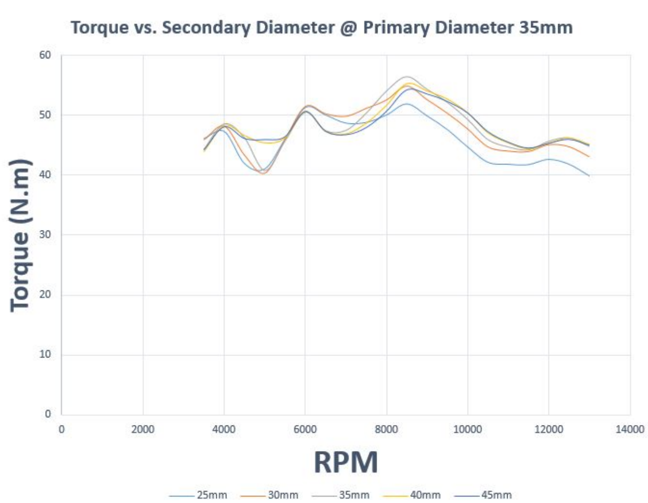


Figure 20: Torque vs Secondary Diameter with Primary Diameter of 35mm

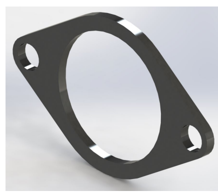
Figures 19 and 20 show that for a primary diameter of 35 mm, a secondary diameters of 45 mm is best suited for our application. Refer to appendix E.4.

9.4 Collectors

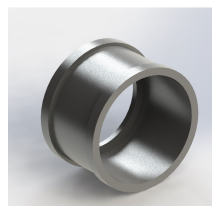
The collectors have also been made in house, in order to save money and save weight. The end sleeves of the collectors have been CNC machined at Holmesglen tafe, while the body of the collectors consist of SS 304 profiled tubes that have been welded together. A simple merge collector design is opted for which has been quite complex to manufacture, requiring two sets of jigs to maintain accuracy. See Appendix for justification E.3.

9.5 Manufacturing

304 Stainless Steel with a 0.89mm WT is used for its superior material properties and due to its current availability. Refer to Appendix E.6 for relevant justification. Despite complicating the manufacturing process, the 0.89mm WT was chosen to maximise spatial constraints as well as keeping the system relatively light weight. The port adaptors similarly are made of 304 Stainless Steel, as they're welded to the runners at one end. They have been CNC machined at Holmesglen TAFE. The purpose of the adaptors are to connect the primary runners to the Cylinder head, with adequate sealing to prevent exhaust leakage and maintain performance. The flanges job is simply to constrain the primary runners to the exhaust ports, as such, an existing design has been modified and laser cut from 3mm mild steel in the Engineering Workshop.



(a) Flange

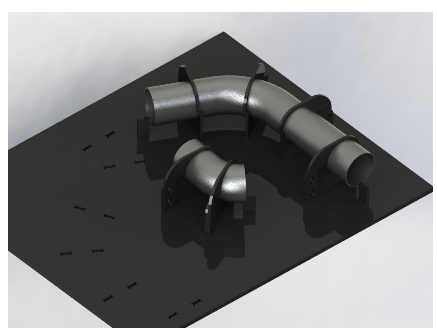


(b) Port Adaptor

9.6 Jig

During welding heating of the metal causes thermal expansion and contraction to occur around the weld bead. This results in distortion or 'warping' of the welded elements which could cause detrimental inaccuracies to the system especially due to the quite small wall thickness. Therefore, an exhaust jig was required to weld all components of each primary and secondary, as well as a larger jig to accommodate the engine itself such that the entire system could be welded together correctly relative to the chassis.

(a) Exhaust sytss



(b) Render

Above all else, packaging constraints of the chassis is what ultimately dictates the maximum performance of the exhaust system; as all ideal cases are limited by the ability to physically manufacture them. After multiple iterations of CAD, the best design ensured approximately a 14mm clearance between the lower firewall and engine block, with tubing having to be quite intricate in order to satisfy the volume constraints. Figures a and b, show a section of a primary tube constrained in a jig and the primary and secondary runners being held in a larger jig with the engine respectively. See Appendix E.6 for all pictures of jig plates. Furthermore, SS 304 is susceptible to oxidation and can often be witnessed during the welding of an exhaust system. The inner surface of the piping unlike that of the outside is not protected by the inert Argon gas and undergoes oxidation, allowing for crystalline layers to form along the weld bead on the inner surface and consequently affect the integrity of the exhaust flow. Therefore in order to minimise this a purging process was utilised to weld the tubing together.

9.7 Ceramic Coating

With the chassis being extremely small there is minimal clearance between the exhaust tubing and various components in the car which is an area of concern. In particular the lower firewall, which is structurally integral to the car, separating exhaust tubing and the fuel tank are millimetres away from each other and cannot be removed. With exhaust temperatures knowing to reach in excess of 700 degrees celsius, deterioration of the carbon is a stark possibility. Therefore, in order to combat the exhaust has been heat coated through a sponsor, Jet-Hot. The coating acts as an insulative jacket and allows for the exterior temperatures of exhaust components to be drastically reduced, increase the fatigue stress limit as well as increase corrosion resistance.

9.8 Muffler

The most important attribute that the muffler must do is comply with the competition regulation of reducing the noise level to less than 110dB (see appendix E.5). Over the last 3 years MUR have been utilising a chamber muffler and have not made any modifications. The aim for this year, has been to change to a straight through muffler. A freer flowing muffler allows the reduction of back pressure in the system and therefore increases scavenging and performance. Initially a muffler was to be built in house out of cheap and free materials readily available; simply for the purposes of testing the effects of varying parameters such as diameter, length, number of perforations etc. However, without complete access to a working dynamometer it could not be followed through. Instead MUR2014 have been able to acquire a straight through muffler via sponsorship from MIDAS Yarraville. The straight through muffler is expected to be quite dense in order to compensate for its small size so that packaging and noise level constraints could be met. However, the benefits of the straight through muffler have come without any significant weight gain. The 2014 muffler weighs approximately 1.6kgs, which is approximately the same weight as the chamber muffler. The quantitative increase in efficiency and performance still needs to be verified on the dynamometer.

Chamber	**Straight Through**
	Advantages
No maintenance required	Easier to manufacture in house
Greater noise attentuation	Light Weight
Potentially reduced insertion loss	Increased flow rates
	Cheaper to manufacture
	Disadvantages
Usually heavier	Requires periodic maintenance
Restricted flow	Increase in noise (Detrimental if noise test failed)
Complex design practice required	
More expensive to purchase	
Harder to package	

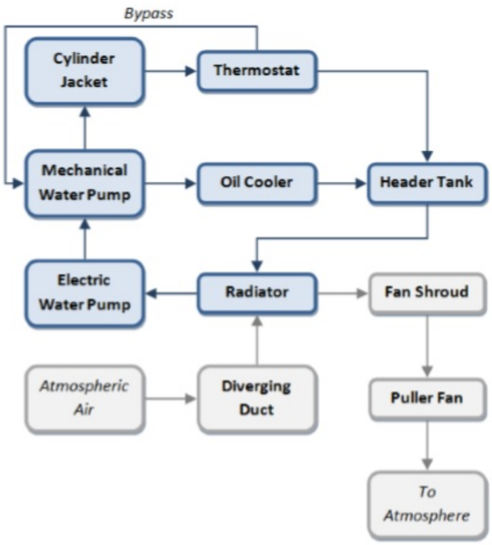
(a) 2014 Straight Through Muffler

(b) 2013 Chamber Muffler

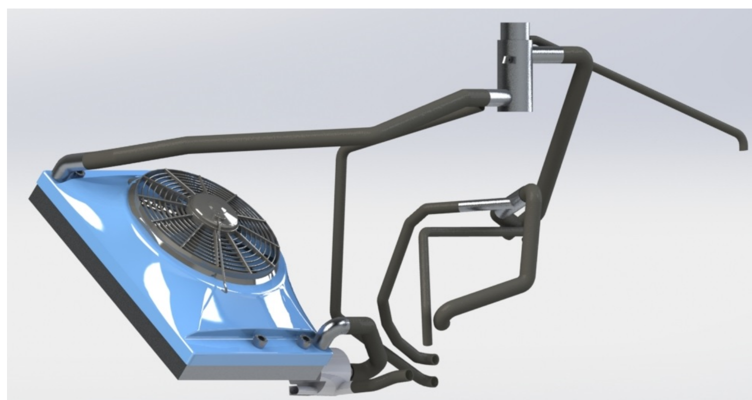
10 Cooling

10.1 Summary

The main function of the engine cooling system is to dissipate the heat produced by the engine so as to maintain optimal operating temperatures. This is accomplished by using an air to liquid heat exchanger; hot water is input into the system from the engine, cooled by the heat transfer that the air across the radiator produces and the cooler water then re-enters the engine. Designing the cooling system requires an approximation for the amount of heat to be dissipated, which will then lead to an appropriate radiator design. Assessing the capabilities of the system is undertaken in conjunction with CFD analysis on the radiator, shroud and cooling fan.



(a) System flow chart



(b) Full Cooling System CAD

10.2 Design Constraints and Considerations

There are numerous design constraints that determined aspects of the system design. Some of these constraints are governed by FSAE competition rules, whilst others are dictated by the composition of the vehicle influenced by the other sub-teams. Wind tunnel testing was also not available this year.
1. Water is the only permitted coolant to be used at competition.
2. Available design space dictated by chassis and other sub-teams' components.
3. No component on the vehicle can protrude outside the widest wheel base, in this case the front wheel base, of the vehicle.
4. All components must be at least a tennis ball width away from the tyres.

Other considerations that influence the potential performance of the system include the chassis out-riggers and the decision to introduce a full aerodynamic package. The chassis out-riggers are the extra set of steel tubing that protrude out the side of the chassis. They increase the torsional rigidity of the vehicle, aiding the overall dynamic performance. However they also reduce the design space outside the chassis where cooling components such as the radiator are located. The aerodynamic package is comprised of a front and rear wing as well as an under-tray. Of these, the front wing has the largest influence on the cooling system. In particular, the front wing re-directs the airflow profile around the vehicle and can potentially restrict the amount of air-flow that reaches the radiator. As such, the 2013 cooling system was analysed under conditions of reduced airflow, in essence simulating the presence of a front wing. This was done by blocking off the inlet duct through to the radiator to varying extents by using a corflute sheet. The results of this experiment, conducted on a track day, are shown below.

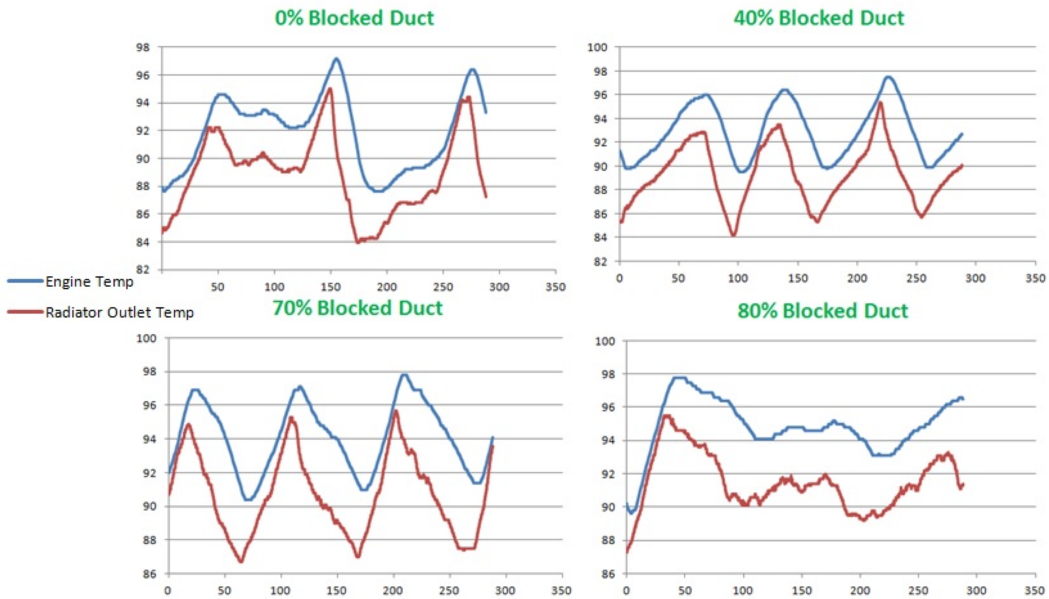


Figure 25: Engine temperature (blue) and radiator outlet temperature (red) for different duct blockage conditions

As would be expected, the cooling performance diminishes with higher peak temperatures reached and longer times spent at these higher temperatures. An obvious knock on effect is that the cooling fan runs longer leading to additional power being drained from the battery. In any case, the key result is that the cooling system was still able to dissipate enough heat from the system, signalling that the system that will be designed can be robust and reliable should air flow be limited.

10.3 Heat dissipation

In order to design the system, the amount of heat the system needs to dissipate must be found. The manner in which this is done here is by first faggot analysing MAP versus engine RPM. MAP can be used to indicate how much power the engine is producing as it relates to the air/fuel mixture entering the cylinders. Power curves produced by the dynamometer are measured at full throttle which is not reflective of driving conditions. As such the near linear relationship between throttle position and MAP means the MAP data can be normalised relative to throttle position. For example, if the power curve indicates 40 kW of power at 7000 RPM, that value is scaled to 20 kW when the normalised MAP percentage is 50%. This is replicated at each instance in time and an average is taken. This will be taken as the average power produced by the engine. Heywood's rule of thumb is that energy from combustion is split into thirds between usable mechanical motion, energy dissipated via the exhaust and heat dissipation. Analysing this data from a routine track day, it was found that the average power output of the engine was 19.89 kW. A 1.1 safety factor is introduced that reflects a slight buffer for design but also the confidence in the system variables, so that the overall heat dissipation requirement of the system is 21.87 kW. See appendix F.1 for calculations.

10.4 Radiator

The radiator is the main component in the system, taking in the hot engine water and cooling it mainly via convection when the cooler air flows across it. The radiator is traditionally placed in one of the side pods due the configuration of the vehicle and to direct a consistent stream of air to it. Due to the FSAE rule that no component can protrude outside the widest wheel width and the chassis out-riggers, the radiator was configured to be a rectangle with its height being the major side. This lead to the decision to use a single pass radiator instead of a dual pass because the design space available meant a dual pass radiator would have its core area dramatically reduced and hence less surface area of tubing would have been available. There were only two realistic positions to place the radiator from a packaging point of view. One was in front of the out-riggers behind the front left wheel, and the other was behind the out-rigger closer to the engine. The latter is the most desirable due to it meaning that the routing would be shorter and hence less water in the system meant conserving mass. CFD simulation of the entire vehicle with the aerodynamic package incorporated was undertaken by the chassis team and lead to the decision to place the radiator in the position closest to the engine. Figure 26 shows the streamlines manoeuvring around the front of the car before they converge back towards the vehicle and into the strategically placed inlet duct. The velocity through the duct is approximately 12 m/s.

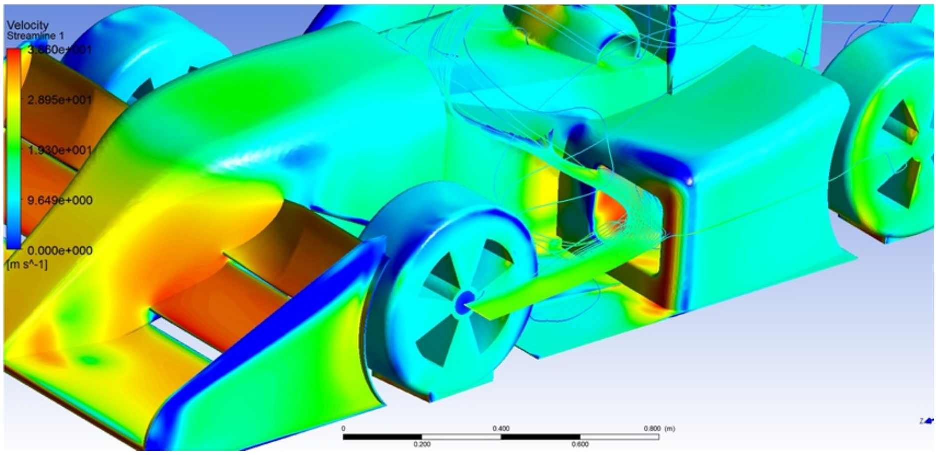


Figure 26: Predicted air flow through inlet duct of radiator

In order to determine a suitable radiator size, the heat transfer area was determined via the effectiveness-NTU method which is outlined in the appendix F.2. Some assumptions were required regarding the speed of the air flow through the radiator, particularly due to the fact that during the design phase, the body kit and aerodynamic devices had not been fully designed. A conservative approach was taken by not factoring in the impact of the fan and a 1.1 safety factor was again used. This lead to computing a heat transfer area of 5.87 m^2. This would be the surface area of tubing required for heat dissipation if the system was to be reliant only on the air flow produced by the movement of the vehicle. This would require a radiator of dimensions greater than that available in the design space. This justifies the use of the fan and is explained in the Cooling Fan and Shroud section. The final radiator dimensions were made based on the available space and the expected contributions of the fan and shroud to produce a pressure drop across the radiator that would be appropriate to meet the heat dissipation requirements. The dimensions are 400 mm$\times$ 280 mm $\times$ 50 mm. The radiator is constrained to the chassis with two tabs on one side designed to resist worst case bending loads and by the body kit on the other side to avoid any rotation. The aluminium tabs were laser cut and then welded on the radiator by the radiator manufacturer, Race Radiators.

(a) Radiator

(b) Radiator aluminium tab

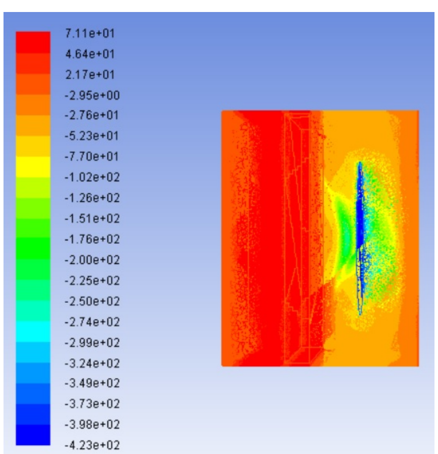
10.5 Cooling Fan and Shroud

As sufficient heat dissipation can not be accomplished by relying purely on the airflow through the radiator generated from the vehicle's motion, a cooling fan is used. It aids to pull more air at a faster rate through the radiator core, improving the heat transfer capabilities of the system. The fan used is a 596 cfm (cubic feet per minute) 9" straight blade Spal SPEF3500 fan. The ECU condition is for the fan to switch on when engine temperature reaches 96 °C and switch off back off at 92 °C. This ensures the engine is operating consistently at optimal temperatures, aiding fuel efficiency and saving power.

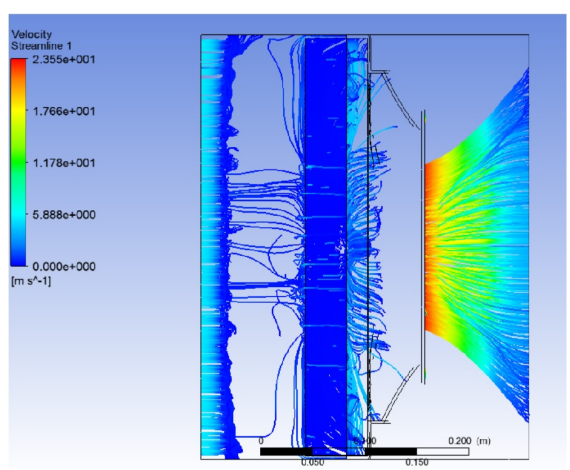
To utilise the full radiator core, a shroud is used between the radiator and fan. From literature, the optimal distance to place the fan from the radiator is between 1.5" and 3". With the available design space and allowing for any manufacturing inaccuracies, the fan is placed 2" from the radiator with the shroud dimensions reflecting this. It provides a sufficient seal so as to avoid airflow leakage as well as providing a mounting point for the fan. Upstream of the radiator is the diverging inlet duct which creates the higher pressure region, and downstream of the radiator is the converging shroud and fan which creates the low pressure region. This is what promotes flow through the radiator. Several shroud designs were analysed using CFD, with the best one selected. This was based on which design produced the best low pressure region, and hence highest air velocity through the radiator.

To justify that the radiator would be capable of dissipating the heat required, CFD simulations were run in ANSYS Fluent in order to compute approximately what the pressure drop across the radiator, shroud and fan would be. From this pressure drop, the velocity of air through the radiator can be computed and a heat transfer area can be found. The inlet duct was treated as an enclosure with the radiator and shroud placed inside it. The duct is completely rectangular which is only an assumption made to simplify the modelling. In reality the duct has more curvature and also has a chassis outrigger just in front of the radiator. The dimensions of the duct are 400 mm$\times$ 280 mm $\times$ 250 mm, allowing for 100 mm upstream and downstream of the radiator. After meshing the geometry a solution was produced using a density based solver with the energy equation and Transition SST models used.

Two different boundary conditions were used at the inlet but only the most reasonable assumption is shown in this section. Refer to appendix F.4 for the other case modelled. The boundary condition used for the inlet here is to make it a pressure-inlet, using the pressure of 120 Pa upstream of the radiator, supplied by the chassis team. The fan was accounted for by the exhaust fan boundary condition, with 400 Pa used as the pressure drop across it (31). The radiator was modelled as a porous media, with porosity of 0.7 and the power law model used to fully define it. These parameters were taken from other automotive radiator simulations found as they would be comparable to the simulations being run here. Finally the duct outlet was treated as a pressure-outlet with pressure set at atmospheric. The results of the CFD analysis for the pressure and velocity can be seen in figure 28a and figure 28b respectively.



(a) Pressure contour of inlet duct, radiator and shroud



(b) Velocity streamlines

The pressure drop is found to be 494 Pa. Based on available information for other CFD models run for similar applications, this result can be viewed as reasonable. The residuals for the various parameters of the simulation are shown in figure 29. As can be seen some of them are still larger than ideal, with the solution not fully converged after 100 iterations. Modelling the cooling fan is one of the more difficult aspects of the simulation. Alternative means to treat the cooling fan are discussed in appendix [.

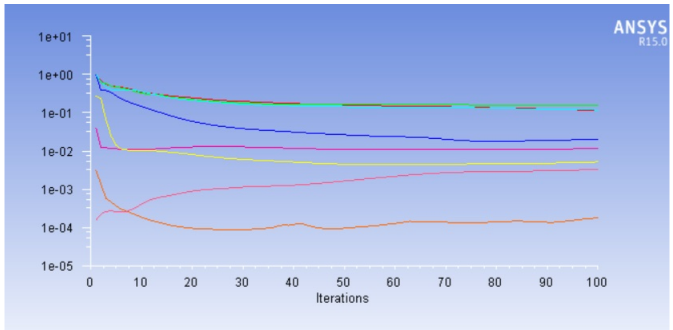


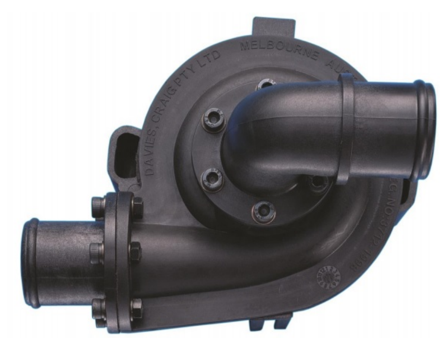
Figure 29: Scaled residuals of the simulation: Continuity (white), x-velocity (red), y-velocity (green), z-velocity (blue), energy (cyan), k (purple), omega (yellow), intermit (orange), retheta (pink)

From the use of Bernoulli's equations the velocity can be computed to be 29.65 m/s. Using the same heat transfer calculation methods used previously and the same factor of safety, a heat transfer area of approximately 4.11 m^2 is computed. This means the radiator manufactured, which had an approximate heat transfer area of 4.23 m^2 is sufficient but also that the system is efficient and not over-designed.

10.6 Electrical and Mechanical Water Pump

Water is pumped through the system primarily by the engine's stock mechanical water pump. The pump is driven off the engine crankshaft and as such the flow rates it provides are in direct relation to the Engine RPM. The results of a flow test conducted in 2012 display the Engine RPM and water flow rate relationship. This plot can be found in appendix F.5.

(a) Stock mechanical water pump



(b) Davis Craig EWP80

The mechanical water pump provides large flow rates, in particular at the higher RPM range compared to most affordable electrical water pumps, but it doesn't address the issue of heat soak. Heat soak is the phenomenon by which the engine can continue to heat up when the cooling performance drops off, like in instances when the vehicle is shut off whilst heat is still being generated but no longer being dissipated at the rate it was. This issue will be averted in the same manner as in the past, through the use of a small electrical booster pump. It switches on only when the vehicle is turned off so as to maintain a flow of water through the system allowing it to continue dissipating heat for a short period after the engine is turned off.

11 Lubrication

11.1 Summary

The lubrication system is mainly designed for maintaining constant oil pressure in the engine block to ensure the oil circulates in the internal oil galleries, so as to lubricate and cool down the moving components. The two types of lubrication systems investigated were the wet sump and dry sump configurations. The comparison of the two are discussed in Appendix G.1.

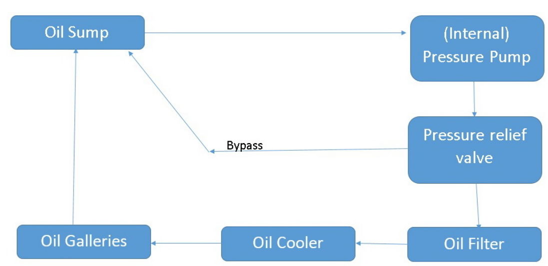
The dry sump lubrication system stores oil externally in an oil tank and uses a scavenge pump to create an external oil circuit. As the oil is not stored in the sump pan, it can be designed to have a relatively small thickness. This allows the engine block to sit closer to the ground and thus improve the dynamic performance of the vehicle via the lowering of the COG. However, this system has drawbacks. The biggest disadvantage observed from previous iterations of this system is its routing complexity which leads to a higher risk of leaking and maintenance difficulties.

The 2014 lubrication system design will be to develop a wet sump system that meets the lubrication requirements of the vehicle. The system will need to combat the addition of the aerodynamic package which will lead to cornering at faster speeds, more lateral G Force and hence a greater deal of oil sloshing within the sump that will be sustained for longer periods of time. The contingency will be a dry sump system similar to previous iterations.

11.2 Design Overview

Based on careful research, a wet sump lubrication system was designed to improve the reliability and simplify the routing compared to the previous dry sump system. The wet sump system stores the oil in the sump pan and enables oil to circulate internally in the engine block. This allows simpler packaging of the system and removes many potential leakage areas. Previous iteration dry sump systems struggled with oil sloshing, so one would expect that it may cause even more of an issue in a wet sump system without an external oil source. This has lead to the addition of a hydraulic accumulator which acts to equalise oil pressure between itself and the engine. Once the oil pressure in the engine drops below a pre-charge amount set in the accumulator, it releases oil back into the engine block, retarding the pressure drop until the vehicle returns to normal operating conditions. This is expected to maintain oil pressure within the system at all times. This report will mainly focus on the wet sump lubrication system design, meanwhile, the process of designing the dry sump lubrication system will also be briefly discussed. Figure 31a and figure 31b shows the system iteration chart and flow chart of the wet sump lubrication system.

(a) Wet sump lubrication system iteration



(b) Flow chart for wet sump lubrication system

11.3 Sump Pan

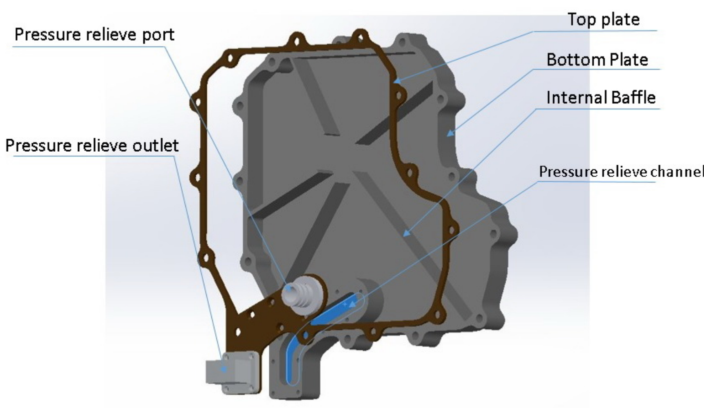
After analysing different options for the wet sump pan design (refer to appendix G.2), it was concluded that the flat sump configuration was the most appropriate concept to investigate because the bottom surface of the sump is the lowest point of the engine block. Since the sump pan acts as the oil container for the wet sump system, a wet sump that is too thin may cause oil scavenging issues. The height of the sump pan is designed to be 25 mm after calculation based on the cross section area of the engine block bottom and oil volume requirement of CBR600 RR engine (the calculation is discussed in detail in appendix G.3). To prevent the oil sloshing issues in the wet sump pan, baffles are designed inside the sump pan.

11.3.1 Flat Plate Sump Design

Considering the design of a flat sump involves modifying the stock sump of the engine, there are a few constraints that should be satisfied.

1. The top plate of the flat sump should be flush with the engine block bottom and the oil pump inlet.
2. A well-sealed interface is required between the top and bottom plate of the sump to cover the external pressure relief channel.
3. The pressure relief outlet from the engine block is almost flush with the bottom surface of the block, a method of relieving pressure from the block is needed.
4. The oil pump inlet of the engine block is located near the centre of the sump, hence baffles should be designed around this fact.

The flat sump pan design is shown in figure 32a



(a) Flat sump pan iteration

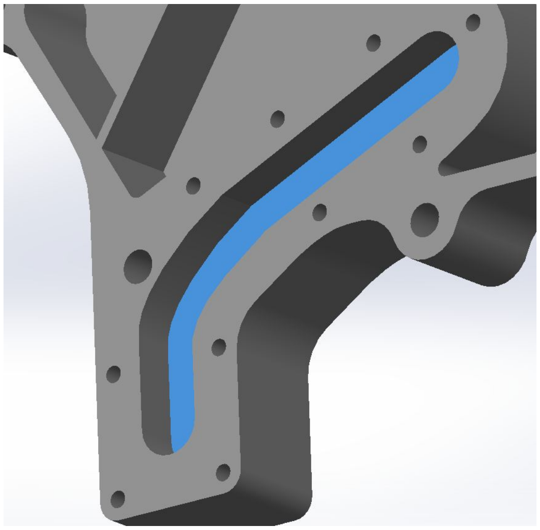

(b) Extended pick up for internal oil pump

The flat sump is designed in two parts, the top plate and the bottom plate(refer to appendix G.4 for details). The bottom plates is made into a oil pocket pool to contain more oil, with a 21 mm thickness whereas the top plate is 4 mm.

11.3.2 Oil supply and Pressure relief

The oil is picked up directly from the sump via the pick up port which has been extended by 20 mm towards the base of the sump. The extended oil pick up was machined to have 3 legs with 5 mm clearance from the sump base to allow for oil pick up. It is shown in figure 32b.

A proper way of relieving high pressure from the engine block should be designed because the HONDA stock internal pressure relive valve is not compatible. The height of the valve is approximately 35 mm which is too high for the 25 mm wet sump. An internal pressure relieve valve was designed in 2013, and it has been tested in a specific testing rig that was designed this year (for further detail refer to appendix G.5). The testing results have shown that the 2013 designed internal pressure relief valve can not satisfy the flow rate requirement of the engine block, hence it is decided that an external pressure relief configuration on the wet sump pan is required. A 9 mm deep channel is designed on the sump bottom plate for pressure relieving. The channels extend towards the rear corners of the sump pan and it is sealed by the top plate with paper gasket between them. Figure 33a demonstrates the design.



(a) Pressure relieve channel at flat sump pan

(b) Aviaid external pressure relieve valve

The pressure in the engine block should be regulated below 7 bar. An Aviaid external in-line pressure relief valve was purchased and is used to relieve the excess pressurised oil back into the engine block. The external pressure relieve is shown in figure 33b.

In order to route the relieved oil back to the engine block, an adaptor is designed to fit the engine block sight glass port and route the relived oil back to the oil sump. The sight glass port adaptor is shown in figure 34.

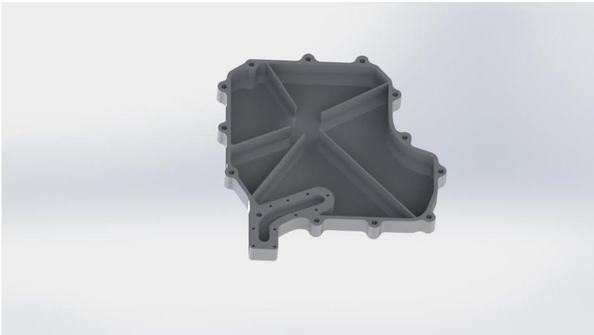
Figure 34: Sight glass port adapter

11.4 Sump baffle design

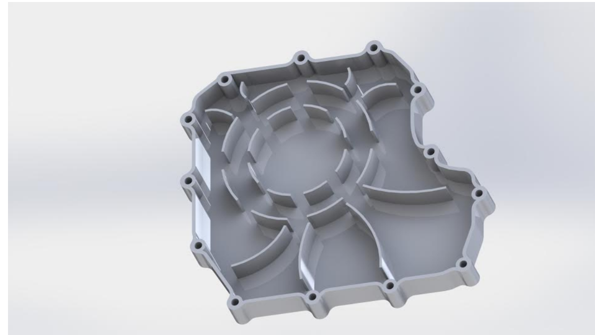
11.4.1 Basic design of oil sump baffles

Since the oil is stored internally in the oil sump, oil sloshing will occur when the vehicle is turning, accelerating or braking. This will increase the risk of picking up air from the internal oil pump. To prevent this, baffles are needed in the wet sump pan to limit the free sloshing of oil.

The baffle design is based on the position of the internal oil pump entrance, leading the oil to pass through the oil pick up point. Two configurations of internal baffles are proposed in figure 35a and figure 35b. Both designs consider the oil pick up point at the centre of the baffle configuration, creating channels which lead oil to slosh through the centre port under dynamic conditions.



(a) Wet sump internal baffle design concept 1



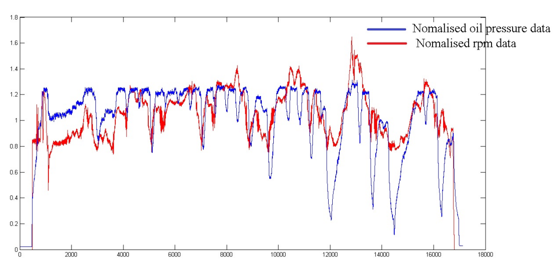
(b) Wet sump internal baffle design concept 2

Considering ease of manufacture and the function of baffles, the baffle configuration shown in figure 35a is applied to the wet sump pan. This configuration has divided the sump pan into four sections; front, back, left and right. Under dynamic conditions, the oil will slosh from the front section to back section when accelerating and flow from back section to front section when braking. Similarly, oil will flow from right section to left section during clockwise turning and from right section to left section during anticlockwise turning. Also as relieved oil returns from the sight glass port, a separate channel is designed to ensure that oil flows through the centre to the pick up point.

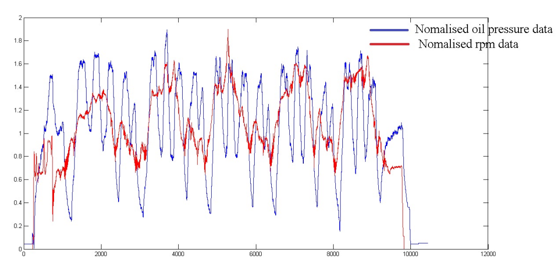
11.4.2 Modifying baffle design based on track data

Since the area of the divided baffle sections has a large influence to oil pressure performance, the baffle configuration has been re-considered based on the track data. During the oil sump testing, the vehicle was running in both clockwise and anticlockwise directions to test the left and right section distribution which is significant to maintain oil pressure.

The engine rpm data and the oil pressure is normalised by dividing by their average value. The normalised RPM and oil pressure data for both clockwise and anticlockwise driving is plotted on the same graph shown in figure 36a and figure 36b.



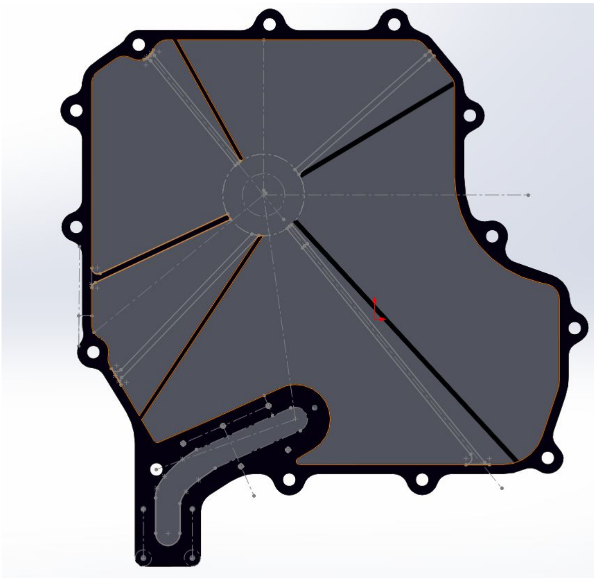
(a)



(b)

Figure 36: (a)Normalised rpm and oil pressure data in CCW running test
(b) Normalised rpm and oil pressure data in CW running test

It could be clearly observed from the data that the anticlockwise running performance is better than the clockwise performance. This is shown by the tracking of the RPM and oil pressure in the CCW case as it would be expected that the oil pressure would increase at higher RPM and vice versa. The CCW case also has smaller fluctuation bands and higher troughs. Hence the left and right section should be re-distributed based on the performance data. The left section area is increased by approximately 15% whereas the right section is reduced by the same amount. The redesigned baffle configuration is shown in figure 37a.



(a) Internal baffle modification design

(b) Flexiable baffle design

Considering ease of manufacture, a flexible baffle configuration was designed. Using a triangular net designed on the top plate enables different baffle configuration to be welded on and tested with relatively low manufacturing cost. This design is shown in figure 37b.

11.5 Oil pressure supplementary system design

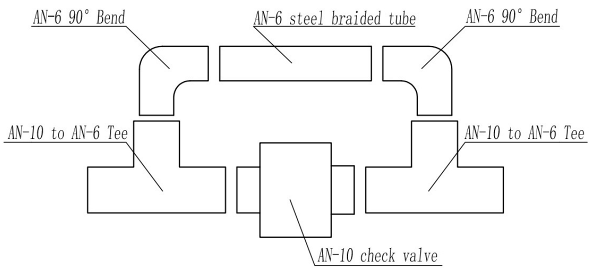
Although baffles are designed in the flat sump plate, with a 25 mm shallow sump pan, the engine is still susceptible to risks of picking up air due to the oil sloshing. Hence, a supplementary system which can help maintain oil pressure is required. Initially approaches such as designing a mechanism which can eliminate the influences of centrifugal force on oil sloshing, was considered. The details of these concepts are discussed in detail in appendix G.6. However, the limitation of the height of the sump meant that additional components within the sump will decrease the volume of oil allowable.

Another solution is to use a hydraulic accumulator as a supplementary device for the wet sump lubrication system. The detail of the working principle of a hydraulic accumulator is discussed in appendix G.7. The hydraulic accumulator is a cylinder with an active piston which divides the cylinder into two chambers. One is filled with gas and the other will be connected with the main engine gallery. This allows the accumulator to charge at high engine oil pressure and discharge at low oil pressure. This discharging of oil into the engine retards the drop in oil pressure for a period of time which will ideally allow for the vehicle to maintain oil pressure until it returns to normal operating conditions. There are various hydraulic accumulators in the market with the hydraulic accumulator unit from Canton Racing selected for its specific racing application. the final selection for the hydraulic accumulator is a 1 Qt (0.94 litre) unit. The accumulator applied to the wet sump lubrication system is shown in figure 38.

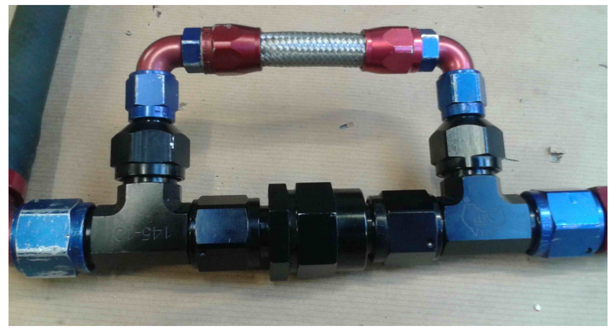
Figure 38: Hydraulic accumulator 1 Qt unit

11.5.1 Hydraulic accumulator flow rate redistribution design

The utilisation of the hydraulic accumulator can effectively reduce the dropping rate of oil pressure during the oil starvation periods, however it still suffers from the risk of discharging all stored oil in extreme conditions, leaving the system without oil for a period of time. The basic principle of flow rate redistribution is designing a hydraulic circuit which enables the accumulator to charge with a relatively high flow rate, meanwhile, discharging at a lower flow rate.



(a) Flow redistribution route design



(b) Check valve and routing

As is shown in figure 39a and 39b, the connection pipe line between the accumulator and the engine main oil gallery is divided into two sub lines; an AN-10 (Pipe diameter 15.8 mm) sub-line and a AN-6 (Pipe diameter 12.7 mm) sub-line. A check valve is utilised on the AN-10 line. This configuration allows the accumulator to charge through both AN-10 and AN-6 line and with the check valve it is only able to discharge through the AN-6 line. This design will ensure the accumulator being charged at a faster rate and discharge at a slower rate.

11.6 Connection and routing design

The hydraulic accumulator is connected to the main oil gallery, with a 10 mm M6 standard thread hole on the engine selected for the connection point. Since all the routing lines are connected with standard AN-thread connectors, an AN-10 to M6 adaptor was required.

The entrance hole of hydraulic accumulator is using 1/2 NPT thread, so another 1/2 NPT to AN-10 adaptor is needed.

11.7 Hydraulic accumulator mounting

The hydraulic accumulator should be mounted on the vehicle with the following being satisfied:
1. The accumulator should be clamped at both ends as described in the Accusump manual.
2. The total weight of the accumulator when full is approximately 2.5 kg meaning it should be mounted relatively low on the chassis for COG purposes.
3. There should be at least 90 mm of clearance for the oil routing at the exit port of the accumulator.

The engine connection port for the accumulator is located on the right side of the vehicle, thus, the accumulator is positioned on the right to reduce length of routing. It is positioned near the engine with tabs attached to the bulkhead. Figure 40 shows the position of the accumulator on the vehicle.

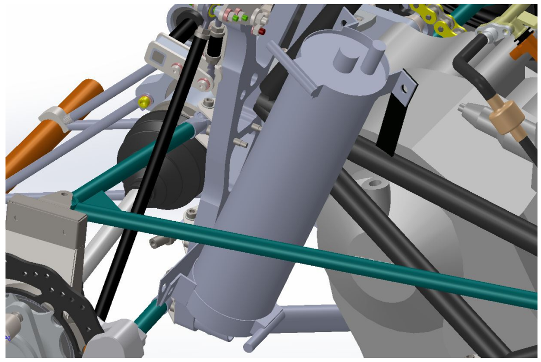


Figure 40: Chassis mounting for accumulator

11.8 Track performance analysis and tuning

The wet sump lubrication system was tested on track, with a 1 bar pre-charge for the accumulator. The vehicle ran both clockwise and anticlockwise directions to acquire the oil pressure data under lateral G-force. Baffles have been re-designed based on the data, as discussed in section 10.5.2. The RPM vs oil pressure map is plotted to assess the performance of the wet sump system.

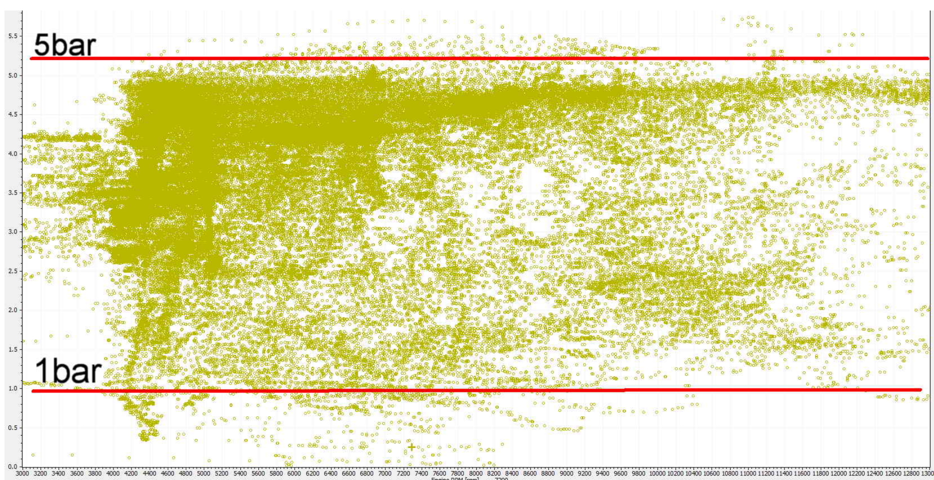


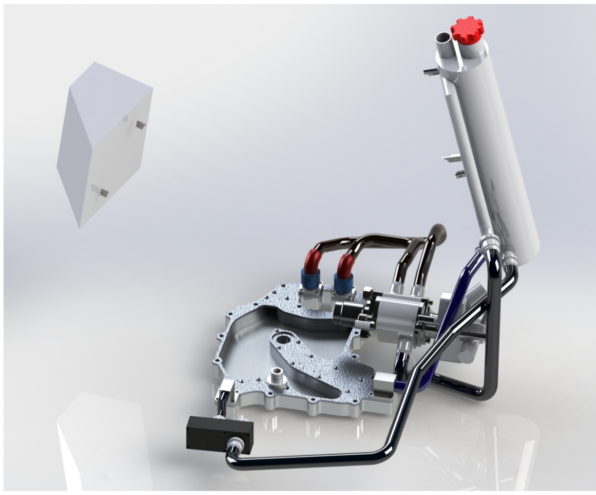
Figure 41: RPM vs Oil pressure(bar) map

As is shown in figure 41, the oil pressure falls into the range between 1 bar and 5 bar which is ideal for engine lubrication. However, instances with oil pressure below 1 bar are still observed.

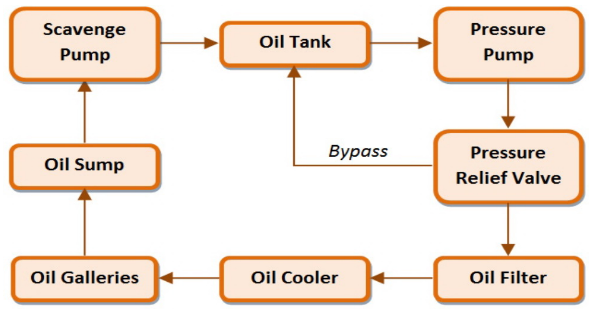
To improve the performance of the wet sump lubrication system, the following strategy will be applied.
1. This will mean the accumulator will have more instances in which it is discharging. The higher pre-charge should mean the oil pressure would be starting from a higher base pressure before any significant pressure drops occur. A potential drawback here is during these major pressure drops, the accumulator will likely have less oil content to discharge at any given time.
2. Increasing the relief pressure of the system:
As the relieving pressure during testing was set at approximately 4 bar, and with the HONDA CBR 600RR oil galleries able to sustain pressure up to 7 bar, a higher relieving pressure can be selected for the same reason as above.

11.9 Dry sump lubrication system design

To ensure the vehicle has a reliable lubrication system, a dry sump lubrication system is also designed. The dry sump iteration and flow chart are shown in figure 42a and figure 42b. For more dry sump design information refer to Appendix??.



(a) Dry sump lubrication system iteration

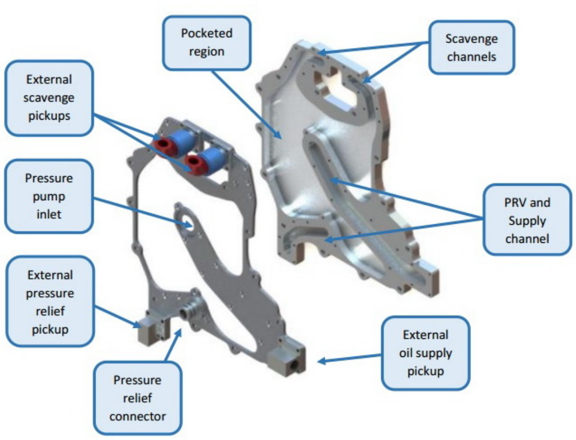


(b) Flow chart for dry sump lubrication system

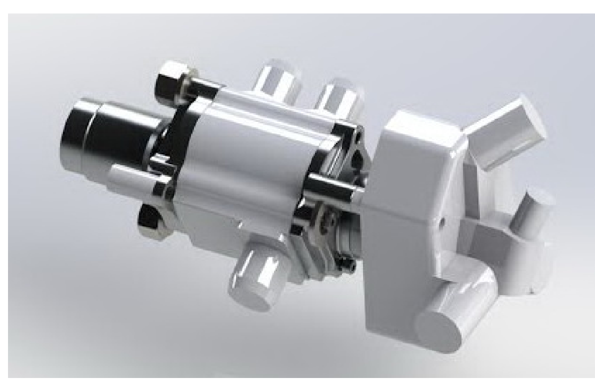
11.9.1 Dry sump pan design

The dry sump pan has an oil pool pocket with two scavenge pickups located at the rear of the engine as with previous iterations of the dry sump system.

Since a 25 mm wet sump pan was used for the wet sump system, the thickness of the dry sump pan will be designed to also be 25 mm so as to stay consistent for suspension setup. It has a 22 mm bottom plate and a 3 mm top plate. Most of the sump pan is pocketed out 20 mm deep, with a pressure relieve channel 9mm deep. For detail of the dry sump pan design refer to appendix G.8. The configuration of the dry sump pan is shown in figure 43a



(a) Dry sump pan iteration



(b) Scavenge pump assembly

12 Conclusion

The goals set out by the Engine team were designed to maximise the vehicles performance at the FSAE competition.

Improved performance was achieved via the full analysis of the intake system, in which volumetric efficiency was improved via the reduction of pressure losses in the manifold. This was aided by designing an exhaust manifold with geometries that facilitate effective exhaust scavenging at the designed operating range. This has allowed MUR 2014 to achieve a theoretical peak torque of 56.6 Nm, which be validated on the dynamometer.

Increased efficiency was achieved in the cooling, lubrication and fuel systems. The cooling system dissipates the required amount of heat without over-designing the heat exchanger dimensions. The lubrication system routing is vastly improved, cutting maintenance time dramatically. The inclusion of RPM driven MAP and fuel pressure will allow a more consistent amount of fuel injected into the cylinders and improve the vehicle fuel efficiency score at competition.

Utilising RPM data from the previous year, peak torque is positioned in the drivers average operating RPM range. The effects of the aerodynamic package on vehicle performance have been considered in the shifting of the torque curve in order to improve overall driveability. Through driver feedback and testing, the 28 mm throttle body allows a better resolution of throttle position and RPM, allowing the driver to comfortably remain in the designed operating range.

Reliability is an ongoing goal that is mainly achieved through on-track testing, in the lead up to the December competition. The wet sump lubrication system will need to be validated through skid-pad testing, as this is the worst case scenario the system will experience. Initial testing shows that the oil pressure is less volatile than previous systems and that this system exhibits stable performance in an autocross or endurance run. Cooling performance will be subject to the success of the aerodynamic package in delivering a reasonable level of air flow to the system. The overall consistency of the engine's performance will be validated and tuned on track in the coming months.

Meeting these goals should ensure the vehicle's capability to complete the endurance event. In doing so, traditionally, a top ten finish is assured. Hitting the performance targets set through effective tuning and on-track testing should allow the team to extract full potential of the vehicle. This will allow the team to achieve a maximum points total at competition.

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°ABDC	Degrees after Bottom Dead Centre
°BTDC	Degrees before Top Dead Centre
°CA	Degrees Crank Angle
A/F	Air/Fuel
ABDC	After Bottom Dead Centre
AT	Air Temperature
ATDC	After Top Dead Centre
BBDC	Before Bottom Dead Centre
BDC	Bottom dead centre; piston at point of maximum cylinder volume
BTDC	Before Top Dead Centre
CFD	Computational Fluid Dynamics
COG	Centre of Gravity
CR	Compression ratio
E85	Fuel with 85% ethanol and 15% unleaded petrol by volume
EBP	Electric Booster Pump
ECU	Engine Control Unit
EFI	Electronic Fuel Injection
ET	Engine Temperature
EWP	Electric Water Pump
FEA	Finite Element Analysis
FSAE	Formula SAE
FSAE-A	Formula SAE - Australia
Lambda	Air/fuel ratio normalised by stoichiometric ratio (λ ¡ 1: Fuel rich; :λ ¿ 1 Fuel lean)
MAP	Manifold Absolute Pressure
MBT	Maximum brake torque, setting to achieve peak torque
MWP	Mechanical Water Pump
OEM	Original Equipment Manufacturer
RON98	98 Research Octane Number unleaded petrol
RPM	Revolutions per minute
SAE	Society of Automotive Engineers
TDC	Top Dead Centre; piston at point of minimum cylinder volume
TPS	Throttle Position Sensor
WT	Wall Thickness

Table 16: Add caption

A Engine

A.1 Engine Block

A.2 HONDA CBR600RR 2003-2006 Engine Specification

Engine Type	4-cylinder in-line
Displacement	599 cc
Bore/Stroke	67.0 mm x 42.5 mm
Compression Ratio	12.0:1
Valve Train	DOHC, 2 intake + 2 exhaust valves per cylinder
Fuel Delivery	Dual-stage electronic fuel injection
Cooling	Liquid-cooled
Starter	Electric start
Lubrication	Wet sump

Table 17: Honda Engine CBFR600RR Specifications

A.3 Other Engine Choices

HONDA CRF 450

- Single cylinder, 450cc, 4 strokes, liquid cooled, 5 speed, standard compression ratio 12:1
- Over-squared with bore and stroke of: 96x62.1 mm
- Electronic controlled fuel injection (model year 2009 onward)
- Kick started (CRF450R) / Electric started (CRF450X)
- Regarding the peak power, please refer to figure 46 for a dyno plot
- Construction materials: aluminium and magnesium with a weight of around 25kg
- Single Overhead Camshaft with 36mm titanium intake valves and 31mm steel alloy exhaust valves
- Water cooling system required with coolant volume of 1.2 litres
- Oil cooling system not required
- According to forum topics suitable for E85 fuel
- Price for new complete engine: around 4000 AUD
- Price for used complete engine: less than 2000 AUD
- Link for some used engines http://www.oemdirtbikeparts.com

Important to notice that there are 2 different models available: CRF450R is the kick-started version, light flywheel since there are no lights and with more high end power; CRF450X is the electric started version, heavier flywheel to power lights and has more torque compared to the other model.

Important to notice that because of the high performances reached by this engine, in order to maintain its reliability, it is necessary to perform some engine controls and piston replacement is recommended after 20 hours of competition use. Personal experience suggests that piston replacement is suggested after 50 hours and highly recommended before 100 hours. The life of the connecting rod can be assumed as doubled compared to the piston, while valves are to be checked at the half of the piston life.

KAWASAKI KXF 450

- Single cylinder, 450cc, 4 strokes, liquid cooled, 5 speed, standard compression ratio 12.5:1
- Over-squared with bore and stroke of: 96x62.1 mm
- Electronic controlled fuel injection (indirect injection from model year 2009)
- Kick started
- Regarding the peak power, please refer to figure 47 for a dyno plot
- Construction materials: aluminium and magnesium with a weight of around 25kg
- Double Overhead Camshaft with 4 valves
- Water cooling system required with coolant volume of 1.2 litres
- Oil cooling system not required
- According to forum topics suitable for E85 fuel
- Price for new complete engine: around 4000 AUD
- Price for used complete engine: less than 2000 AUD
- http://www.oemdirtbikeparts.com
- http://www.moto-recyclers.net
- http://www.ebay.com/itm/2009-09-Kawaski-KX450F-KX-450-Engine-Motor-Stator-assembly-/300588297582

SUZUKI RM-Z 450

- Single cylinder, 450cc, 4 strokes, liquid cooled, 5 speed, standard compression ratio 12.5:1
- Over-squared with bore and stroke of: 96x62.1 mm
- Electronic controlled fuel injection (indirect injection from model year 2009)
- Kick started (RM-Z 450) / Electric started (RM-X 450)
- Regarding the peak power, please refer to figure 48 for a dyno plot
- Construction materials: aluminium and magnesium with a weight of around 25kg
- Double Overhead Camshaft with 4 valves
- Water cooling system required with coolant volume of 1.2 litres
- Oil cooling system not required
- According to forum topics suitable for E85 fuel
- Price for new complete engine: around 4000 AUD
- Price for used complete engine: less than 2000 AUD
- http://compare.ebay.com.au/like/170750900097?ltyp=AllFixedPriceItemTypes
- http://www.moto-recyclers.net

Figure 44: Suzuki Engine

YAMAHA YZF/WR 450 (MONOCYLINDER)

- Single cylinder, 450cc, 4 strokes, liquid cooled, 5 speed, standard compression ratio 12.5:1
- Over-squared with bore and stroke of: 97x60.8 mm
- Electronic controlled fuel injection (indirect injection from model year 2009)
- Kick started (YZF 450) / Electric started (WR 450)
- Regarding the peak power, please refer to figure 49 for a dyno plot
- Construction materials: aluminium and magnesium with a weight of around 25kg
- Double Overhead Camshaft with 4 titanium valves
- Water cooling system required with coolant volume of 1.2 litres
- Oil cooling system not required
- According to forum topics suitable for E85 fuel
- Price for new complete engine: around 4000 AUD
- Price for used complete engine: less than 2000 AUD
- http://www.oemdirtbikeparts.com

The main difference of this engine with the other Japanese made engines consists in the fact that the intake manifold is located in the front of the engine, instead of the back. This solution should allow a better allocation of the exhaust, as well as a decreased heat transmission to the driver's seat and other car components.

APRILIA RXV/SXV 550 (TWO CILINDERS)

- -twin with an angle of 77 degrees, 550cc, 4 strokes, liquid cooled, 5 speed, standard compression ratio 12.5:1
- Over-squared with bore and stroke of: 80x55 mm
- Electronic controlled fuel injection
- Electric started
- Construction materials: aluminium and magnesium with a weight of around 32 kg
- Each cylinder has a single overhead cam with 4 valves each

- Water cooling system required with coolant volume of 1.1 litres
- Oil cooling system not required
- According to forum topics suitable for E85 fuel
- Price for new complete engine: around 5000 AUD
- Price for used complete engine: around 2500 AUDd

Please note that, even if this engine gives optimal performances compared to its weight and dimensions, it needs constant maintenance, even more compared to a Japanese built engine, in order to avoid possible failures. In addition the availability of spare parts in Australia should be a bit more complicated compared to the above mentioned engines.

KTM EXC 500

- Single cylinder, 510cc, 4 strokes, liquid cooled, 6 speed, standard compression ratio 12.5:1
- Over-squared with bore and stroke of: 95x72 mm
- Electronic controlled fuel injection
- Kick-started and electric started
- Construction materials: aluminium and magnesium with a weight of around 32 kg
- Double Overhead Camshaft
- Water cooling system required with coolant volume of 1.2 litres
- Oil cooling system not required
- According to forum topics suitable for E85 fuel
- Price for new complete engine: around 5000 AUD
- Price for used complete engine: around 2500 AUD

Figure 45: KTM Engine

SUZUKI LT-R 450 The engine used on the ATV Suzuki LT-R 450 is nearly the same used on the motocross and enduro motorbike with the difference that it already start with electric starting and the ECU had been set for more low end power instead of pure high end like in the motorbike case. The negative of this engine will probably be to be a bit heavier than the motorbike one (it is just a supposition since no official data had been found), but the reliability and the time between each component substitution will be higher, since the maximum peak power and torque will be lower.

HONDA TRX 450 As well as in the Suzuki LT-R, the engine that equips the Honda TRX 450 is directly derived from the off road motorbike. It has been put in this review in order to know that is possible to consider this engine since it starts with electric start and will be a bit less expensive from a maintenance point of view, since the maximum peak power and torque will be lower.

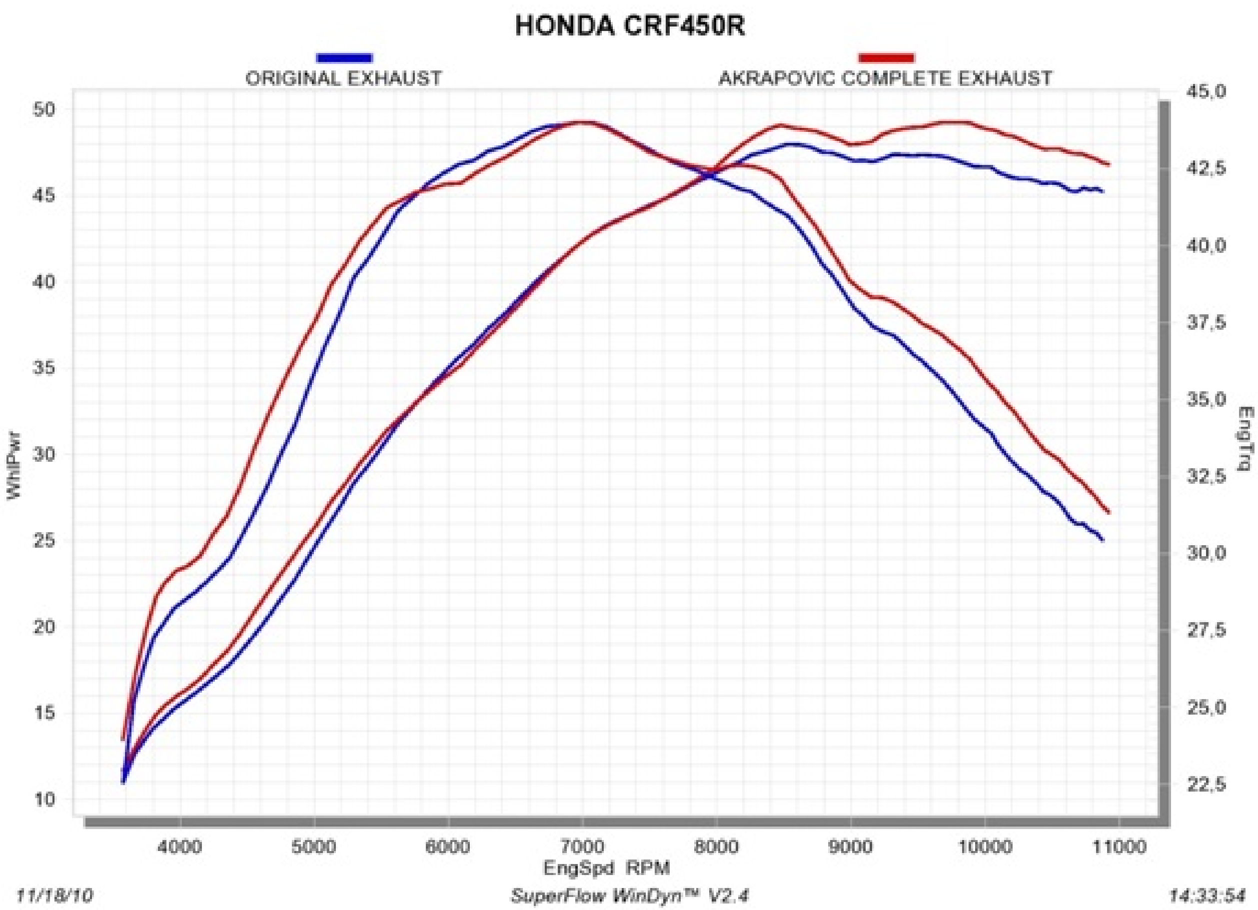


Figure 46: Honda CRF 450R Tourque and Power

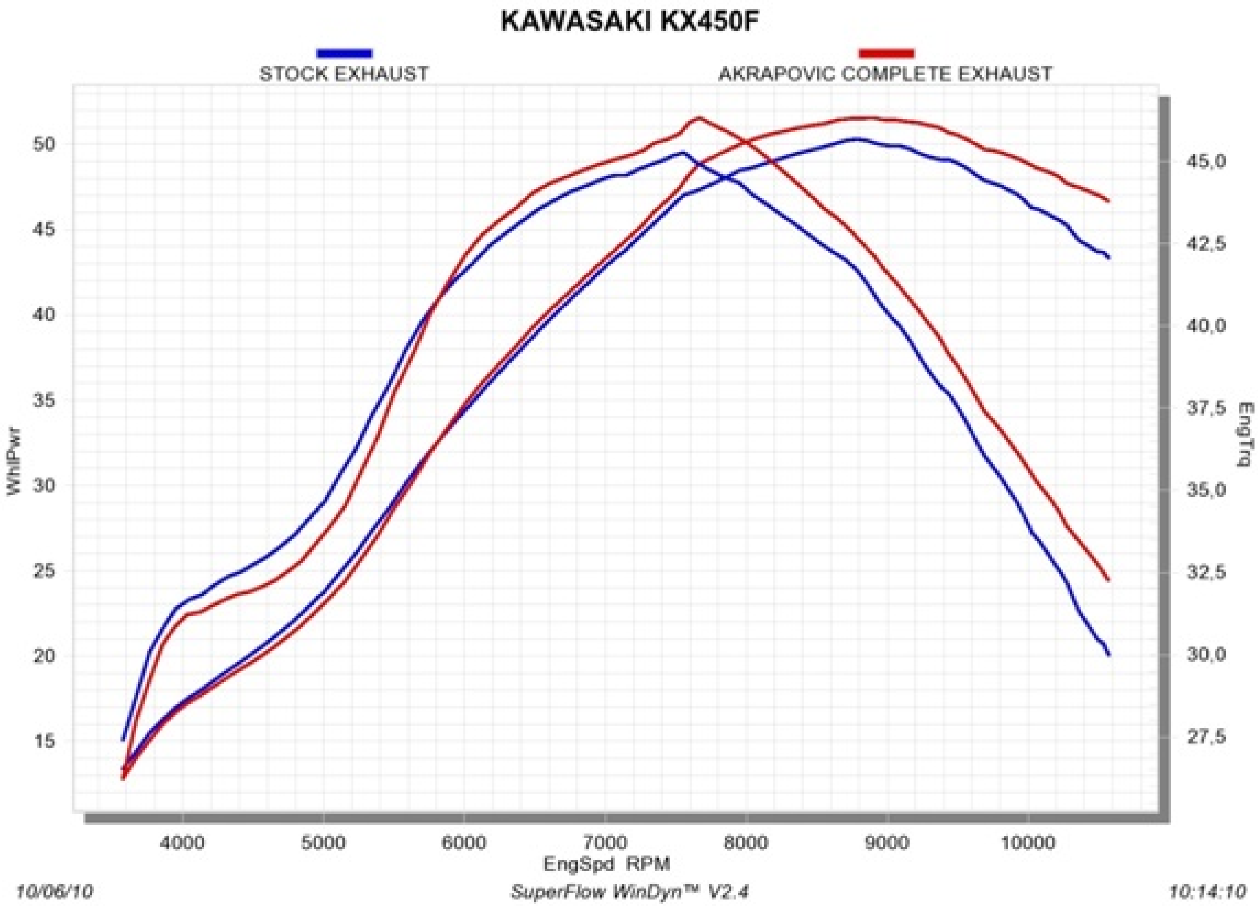


Figure 47: Kawasaki KX450F Torque and Power

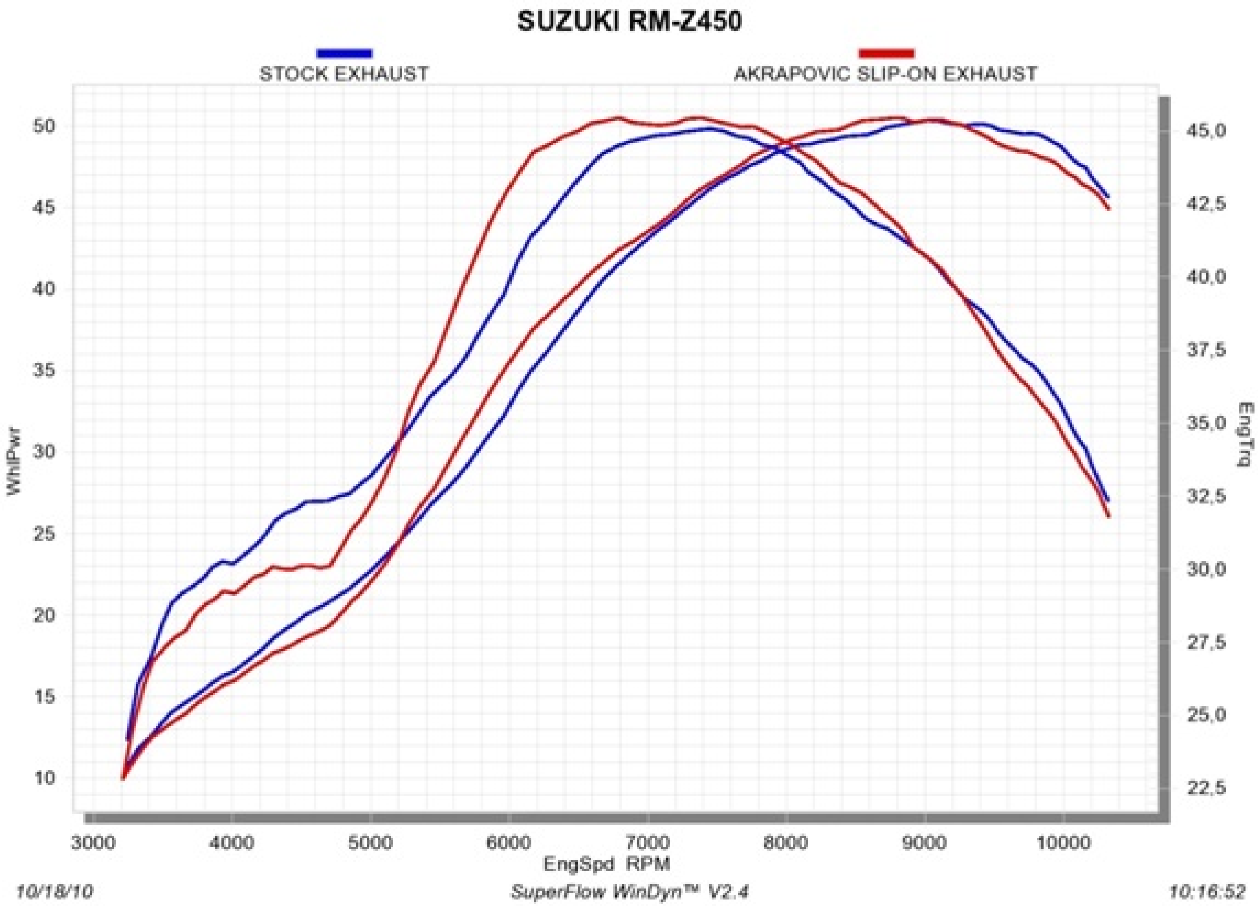


Figure 48: Suzuki RM-Z450 Torque and Power

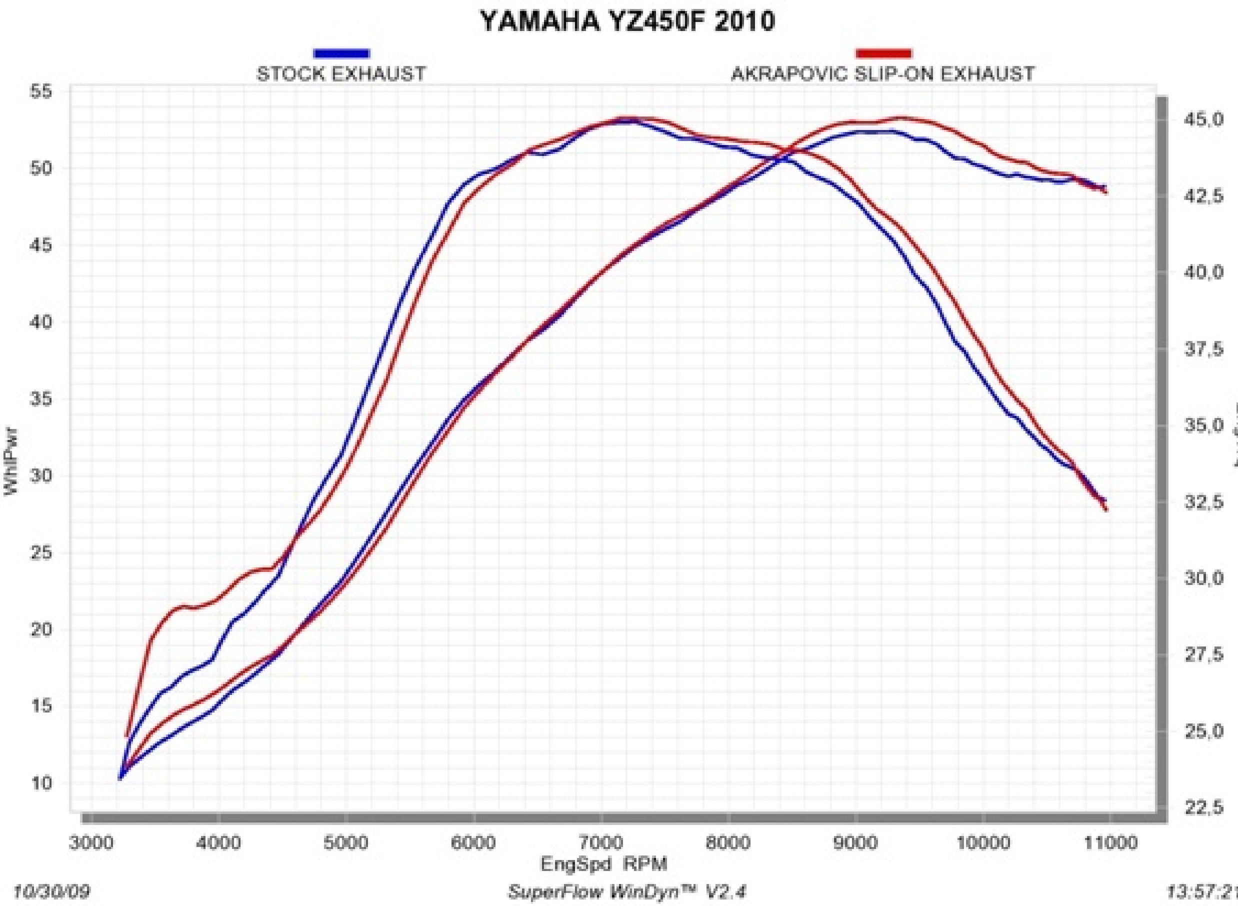


Figure 49: Yamaha YZ450F Torque and Power

A.4 CAM Profile Data

A.4.1 Intake Lift Profile

Cam Angle(deg)	*Lift (in)*	*Cam Angle(deg)*	*Lift (in)*	*Cam Angle (deg)*	*Lift (in)*
-85	0	-46	0.099451	-7	0.285369
-84	4.64E-06	-45	0.105737	-6	0.286612
-83	3.52E-05	-44	0.112145	-5	0.287657
-82	0.000112	-43	0.11867	-4	0.288506
-81	0.000251	-42	0.125302	-3	0.289162
-80	0.000459	-41	0.132035	-2	0.289628
-79	0.000741	-40	0.138861	-1	0.289907
-78	0.001093	-39	0.145773	0	0.29
-77	0.001508	-38	0.152721	1	0.289907
-76	0.00197	-37	0.159542	2	0.289628
-75	0.002459	-36	0.166224	3	0.289162
-74	0.002955	-35	0.17276	4	0.288506
-73	0.003453	-34	0.179142	5	0.287657
-72	0.003952	-33	0.185365	6	0.286612
-71	0.004451	-32	0.191424	7	0.285369
-70	0.005202	-31	0.197315	8	0.283923
-69	0.006264	-30	0.203036	9	0.282272
-68	0.007631	-29	0.208586	10	0.280414
-67	0.009295	-28	0.213963	11	0.278348
-66	0.011248	-27	0.219167	12	0.276074
-65	0.013483	-26	0.224197	13	0.273591
-64	0.015993	-25	0.229054	14	0.270903
-63	0.018769	-24	0.233738	15	0.268014
-62	0.021804	-23	0.238249	16	0.26493
-61	0.025091	-22	0.242588	17	0.261656
-60	0.028622	-21	0.246753	18	0.258198
-59	0.032389	-20	0.250744	19	0.254559
-58	0.036386	-19	0.254559	20	0.250744
-57	0.040603	-18	0.258198	21	0.246753
-56	0.045035	-17	0.261656	22	0.242588
-55	0.049673	-16	0.26493	23	0.238249
-54	0.05451	-15	0.268014	24	0.233738
-53	0.059537	-14	0.270903	25	0.229054
-52	0.064749	-13	0.273591	26	0.224197
-51	0.070136	-12	0.276074	27	0.219167
-50	0.075693	-11	0.278348	28	0.213963
-49	0.08141	-10	0.280414	29	0.208586
-48	0.08728	-9	0.282272	30	0.203036
-47	0.093297	-8	0.283923	31	0.197315
Cam Angle(deg)	*Lift (in)*	*Cam Angle(deg)*	*Lift (in)*	*Cam Angle(deg)*	*Lift (in)*
32	0.191424	51	0.07013	70	0.005201
33	0.185365	52	0.064743	71	0.004451
34	0.179142	53	0.059531	72	0.003952
35	0.17276	54	0.054504	73	0.003453
36	0.166224	55	0.049667	74	0.002955
37	0.159542	56	0.04503	75	0.002459
38	0.152721	57	0.040599	76	0.00197
39	0.145772	58	0.036381	77	0.001508
40	0.138859	59	0.032385	78	0.001093
41	0.132032	60	0.028618	79	7.41E-04
42	0.125298	61	0.025088	80	4.59E-04
43	0.118665	62	0.021802	81	2.51E-04
44	0.11214	63	0.018767	82	1.12E-04
45	0.105731	64	0.015991	83	3.52E-05
46	0.099445	65	0.013482	84	4.64E-06
47	0.09329	66	0.011247	85	0.00E+00
48	0.087274	67	0.009294	86	0.00E+00
49	0.081403	68	0.007631		
50	0.075686	69	0.006264		

Table 18: Intake Lift Data

A.4.2 Exhaust Lift Profile

Cam Angle(deg)	*Lift (in)*	*Cam Angle(deg)*	*Lift (in)*	*Cam Angle (deg)*	*Lift (in)*
-81	0	-42	0.099946	-3	0.259189
-80	2.53E-11	-41	0.106894	-2	0.25964
-79	4.47E-05	-40	0.113911	-1	0.25991
-78	2.45E-04	-39	0.120929	0	0.26
-77	5.13E-04	-38	0.127879	1	0.25991
-76	8.02E-04	-37	0.134712	2	0.25964
-75	0.0011	-36	0.141387	3	0.259189
-74	0.0014	-35	0.147881	4	0.258557
-73	0.0017	-34	0.154178	5	0.257744
-72	0.002	-33	0.160271	6	0.256747
-71	0.0023	-32	0.166161	7	0.255567
-70	0.0026	-31	0.171854	8	0.254202
-69	0.002917	-30	0.177358	9	0.252652
-68	0.003325	-29	0.182683	10	0.250915
-67	0.003898	-28	0.18784	11	0.24899
-66	0.004683	-27	0.192835	12	0.246877
-65	0.005711	-26	0.197673	13	0.244575
-64	0.006993	-25	0.20235	14	0.242083
-63	0.008528	-24	0.206861	15	0.239401
-62	0.010316	-23	0.211202	16	0.23653
-61	0.012359	-22	0.215368	17	0.23347
-60	0.014655	-21	0.219356	18	0.230221
-59	0.017205	-20	0.223162	19	0.226785
-58	0.020009	-19	0.226785	20	0.223162
-57	0.023067	-18	0.230221	21	0.219356
-56	0.026378	-17	0.23347	22	0.215368
-55	0.029943	-16	0.23653	23	0.211202
-54	0.033762	-15	0.239401	24	0.206861
-53	0.037836	-14	0.242083	25	0.20235
-52	0.04217	-13	0.244575	26	0.197673
-51	0.046768	-12	0.246877	27	0.192835
-50	0.051634	-11	0.24899	28	0.18784
-49	0.056774	-10	0.250915	29	0.182683
-48	0.062188	-9	0.252652	30	0.177358
-47	0.067874	-8	0.254202	31	0.171854
-46	0.073826	-7	0.255567	32	0.166161
-45	0.080033	-6	0.256747	33	0.160271
-44	0.086474	-5	0.257744	34	0.154178
-43	0.093124	-4	0.258557	35	0.147881
Cam Angle(deg)	*Lift (in)*	*Cam Angle(deg)*	*Lift (in)*	*Cam Angle(deg)*	*Lift (in)*
36	0.141387	56	0.026378	76	8.02E-04
37	0.134712	57	0.023066	77	5.13E-04
38	0.127879	58	0.020009	78	2.45E-04
39	0.120929	59	0.017205	79	4.47E-05
40	0.113911	60	0.014655	80	5.20E-11
41	0.106894	61	0.012359	81	0
42	0.099945	62	0.010316		
43	0.093123	63	0.008527		
44	0.086474	64	0.006993		
45	0.080032	65	0.005711		
46	0.073826	66	0.004683		
47	0.067874	67	0.003897		
48	0.062187	68	0.003325		
49	0.056774	69	0.002917		
50	0.051634	70	0.0026		
51	0.046767	71	0.0023		
52	0.04217	72	0.002		
53	0.037836	73	0.0017		
54	0.033762	74	0.0014		
55	0.029943	75	0.0011		

Table 19: Exhasut Lift Data

B Tuning

MUR2014 will follow the below steps in operating the Dyno as describe by MUR2013:

1. Calibration of throttle position sensor (TP) and lambda sensor at the beginning of every test session;
2. Engine fluids check;
3. Engine warm up to operating T;
4. Engine parameters check;
5. Set up of desired rpm on the dyno control screen;
6. Set up desired TP (WOT or partial throttle);
7. Allow engine to settle at the desired rpm;
8. Modify fuel map values to obtain desired lambda (usually 0.9 for maximum torque or 1.1 for fuel efficiency);
9. Modify ignition timing map to obtain MBT;
10. Collect desired data (e.g. torque, ignition advance, fuel duty)
11. Set up of the next desired rpm on the dyno screen and proceed repeating points from 7-10

Start Date	**End Date**	**Task**
5/06/2014	**13/06/2014**	Fix Fuel Injectors
19/05/2014	**13/06/2014**	Print Intake components
7/06/2014	**19/06/2014**	Acetone Vapour Bath Intake
13/06/2014	**23/06/2014**	Run Basic Dyno tests on 2012 Components
23/06/2014	**27/06/2014**	Run Ignition Sweeps with new intake
		See effect of spark advance
		Perform lambda testing for optimal power at 8500 RPM
		Create a fuel MAP for maximum power and test at Track
		Fuel MAP for endurance
		Torque and Power Tests
30/06/2014	**4/07/2014**	Test 2013 new cams
		Power and Torque comparison
		Volumetric efficiency comparison
		Print Increased volume of intake
		Torque and Power Tests 2014 Intake V1
		Fuel MAP for cold starts
		Lambda testing for optimal power
		Spark Advance Tests
7/07/2014	**11/07/2014**	Volumteric Efficiency tests
		Throttle Response comparison
		Fuel MAP for Autocross
		Fuel MAP for acceleration
		Fuel MAP for Skid Pad
		Fuel MAP for Endurance Pad
		MAP Reference PWM
		Ignition Timing
		Brake Specific Fuel Consumption Analysis
14/07/2014	**18/07/2014**	Hook up 2014 Accusump and test
		Order Fuel Injectors (if sponsored)
		Torque and Power test on 2014 Intake v2
		Fuel MAP for Autocross
		Fuel MAP for acceleration
		Fuel MAP for Skid Pad
		Fuel MAP for Endurance Pad
		Ignition Timing
		Brake Specific Fuel Consumption comparison
		Purchase new Filter
		Volumetric Efficiency Comparison
21/07/2014	**25/07/2014**	Compare effect of air filters
		Torque and power comparison
		Fuel Consumption
		Volumetric Efficiency Comparison
		Test new fuel injectors
		Print of Final 3D Plenum
		Validate ANSYS Model
28/07/2014	**1/08/2014**	Ignition MAP
		Spark Advance Tests and Tuned
		Ignition Timing
		Fuel MAP for Autocross
		Fuel MAP for acceleration
		Fuel MAP for Skid Pad
		Fuel MAP for Endurance Pad
		Throttle Response comparison
		MAP Reference PWM
		Brake Specific Fuel Consumption Analysis
		Iterate based upon Track Day Results with aerodynamics Kit on
4/08/2014	**27/08/2014**	Extra time for repairs if needed
		If extra time is obtained then the following will be tested: Throttle Body Tests (if sponsorship permits) ,Fuel Injection Angle tests, Put higher compression engine on Dyno and test new plenum

Table 20: Predicted Dyno Timeline and tasks

B.1 Fuel

Figures 50, 51 and 52 show GT Power results for a lambda sweep between 0.7-1.1. The purpose was to help verify a lambda value that will produce the highest torque. It is evident without proper ignition timing, these results show that a value of 0.9 shows the best value.

A spark timing DOE was run, however each time an attempt was made to open GT Post, errors would occur in the Java script. This would be a good idea for next years team to implement and compare it with dyno results.

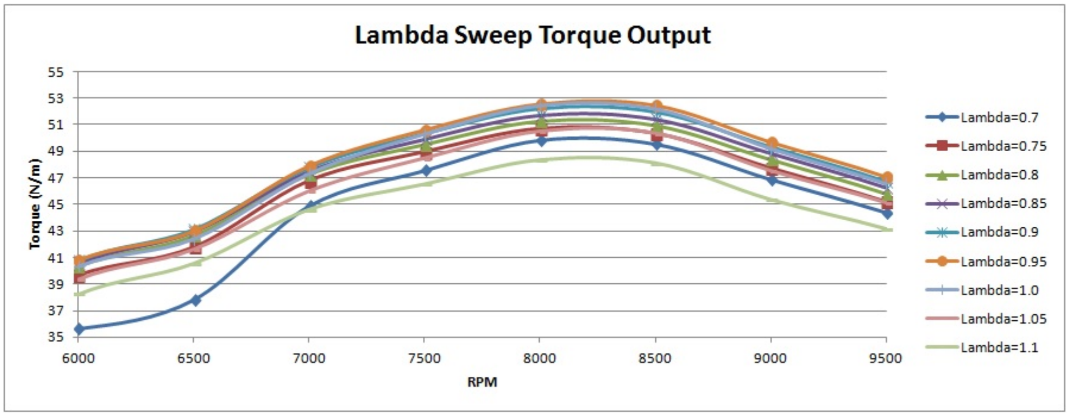


Figure 50: Effect of Lambda on Torque

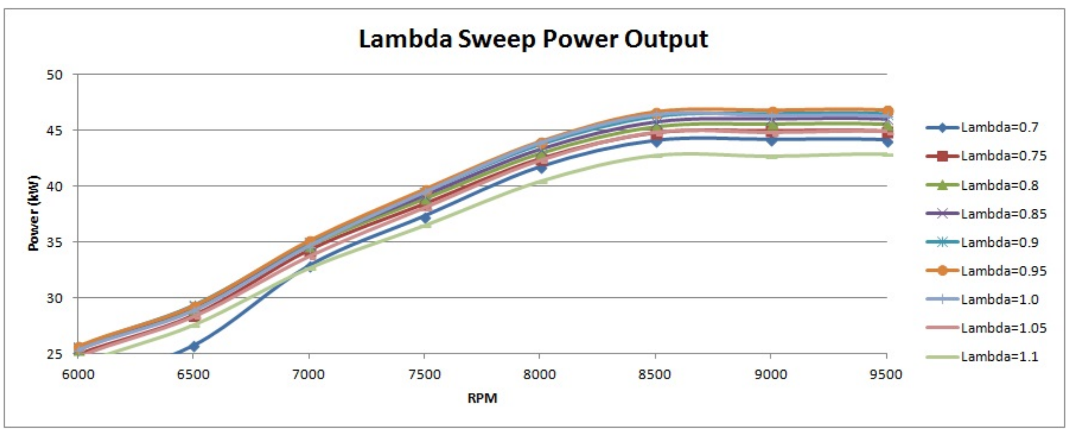


Figure 51: Effect of Lambda on Power

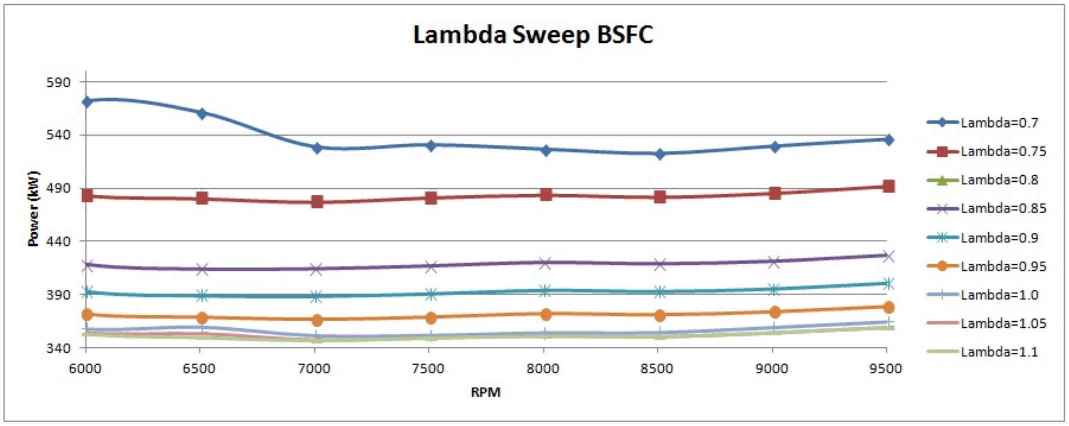


Figure 52: Effect of Lambda on Brake Specific Fuel Consumption

C Intake

C.1 Packaging Restrictions

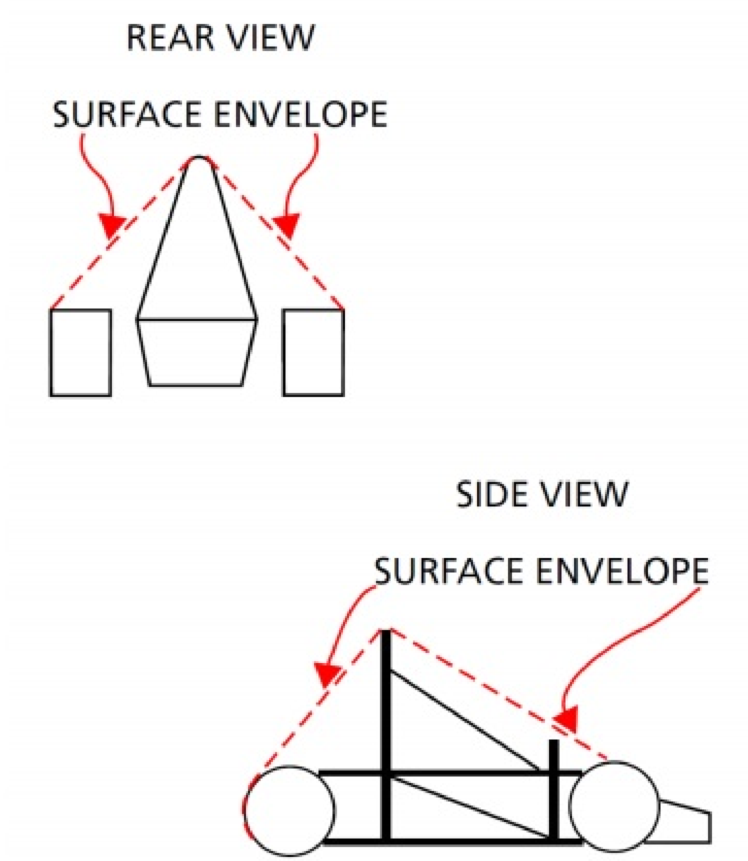


Figure 53: Intake Packaging Restriction

C.2 MATLAB Analysis

A major goal of the 2014 Engine team was to build upon previous years designs and their techniques to advance the engine performance by understanding its limitations. The Helmholtz resonator models has been the corner stone for predicting runner length and diameter and hence is essential for tuning the torque output of the engine. This analysis is conducted for one inlet stroke. The pressure wave is produced at the inlet valve during the downward motion of the piston, which produces a rarefraction wave and travels down the intake pipe and is reflected as a compression wave. A positive tuning effect occurs when the compression wave arrives just before the intake valve closes.

$$f_1 = \frac{1}{2\pi}\sqrt{\frac{A - B}{2 \times IND_{Ratio} \times CAP_{Ratio} \times IND_{Primary} \times V_{eff}}} \tag{C.1}$$

$$f_2 = \frac{1}{2\pi}\sqrt{\frac{A + B}{2 \times IND_{Ratio} \times CAP_{Ratio} \times IND_{Primary} \times V_{eff}}} \tag{C.2}$$

Equations D.1 and equation D.2 represent the induction frequency resonances. An electrical circuit is used to derive the equations where capacitors represent volumes and inductors represent pipes. The primary resonance is given by equation D.3

$$f_p = \frac{1}{2\pi}\sqrt{\frac{1}{V_{eff} \times IND_{Primary}}} \tag{C.3}$$

It is the primary ratio that is used to calculate the frequency ratios by equations C.4 and equation C.5

$$X_1 = \frac{f_1}{f_p} \tag{C.4}$$

$$X2 = \frac{f_2}{f_p} \tag{C.5}$$

The tuning peak is then calculated using the the engine characteristics which is showen in equation

$$N_p = 642 \times Sound_{Vel} \times \sqrt{\frac{A_1}{Runner_L + Port_L} \times Cyl_{Disp} \times \frac{Cr - 1}{Cr + 1}} \tag{C.6}$$

Finally the intake tuning peaks are derived as ratio of both the frequency ratios and the tuning peak.

$$N_1 = X_1 \times N_p \tag{C.7}$$

$$N_2 = X_2 \times N_p \tag{C.8}$$

The tuning effect or resonant supercharging that occurs in the manifold leads to increased volumetric efficiency and hence increased torque and power. However, the expansion waves will also travel to the inactive runners with closed valves and be reflected as expansion waves back to the active runner. These expansion waves reduce intake runner pressure and hence reduced volumetric efficiency.

Our results show that based upon the 2013 design we expect two resonant peaks that will produce maximum torque at 3989 rpm and 8677 RPM. Comparing this with the 2013 data both from GT Power and from Dyno testing this is slightly over estimated by around 8"%". The Helmholtz resonator does not account for many of the losses that occur in pipe flow, such as surface roughness, heat losses, turbulence effects and pressure losses. This model however did provide us with preliminary design constraints for the intake manifold.

A recommendation from last year was to analyse the effect that plenum volume has on the torque output. Figure 54 shows how different runner length torque curves will shift as we increase the plenum volume. Figure 55 indicates that the torque output has a slight shift as the runner length increases at the first frequency. The location of the first frequency is around the idle rev range. Moreover the reduction in design space with an increased runner length and the increase in flow disruption to the rear wings does not justify a volume change.

Figure 56 and figure 57 shows an extremely good correlation that plenum volume has very little effect on location of peak torque. This is in contradiction to the what Hamilton and Lee [(26)] predicted. Their experimental evidence did show ta relationship that can quite clearly seen above 6500 RPM; moreover, strong evidence for a distinct rise in torque.

Figure 58 and figure 59 showed how the effect of increasing plenum volume at a desired runner length increases the location the peak frequency more accurately. Looking at this data, it clear that in order to achieve our operating range of 6000-9000 RPM we require runner lengths between 18-26cm. Figure 60 shows a relationship between plenum volume and throttle response. There is a decrease in throttle response of 1.5ms in the designed RPM range. This difference needs to be tested out on track to ensure that if there is an increase in torque that it is not made redundant by the lag associated by resonant charging.

As you can see this type of analysis is adequate for initial design parameters. It fails to give any information about torque output which is a key design parameter. Furthermore, a lot of prototypes iterations would be required to obtain accurate results.

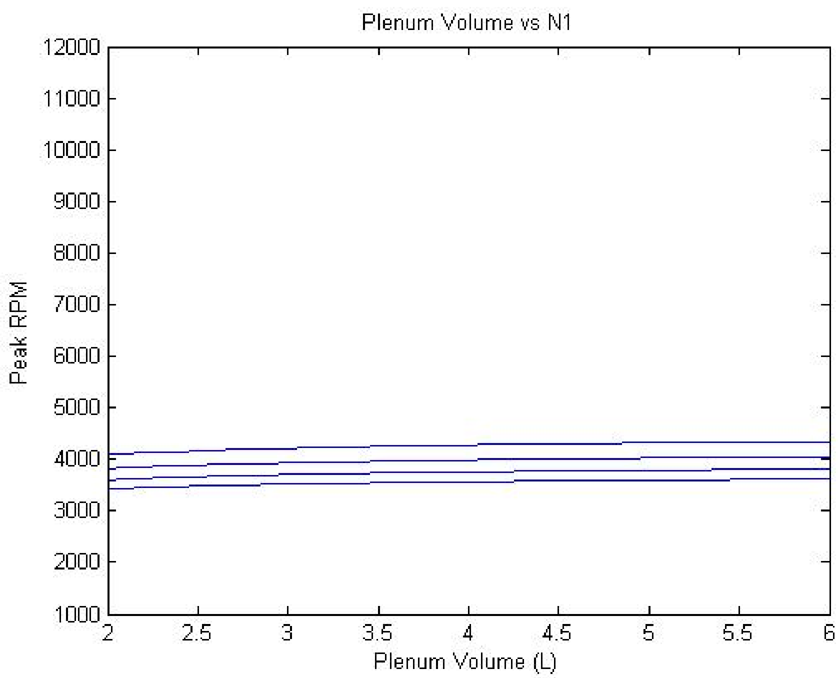


Figure 54: Runner Length vs First Frequency

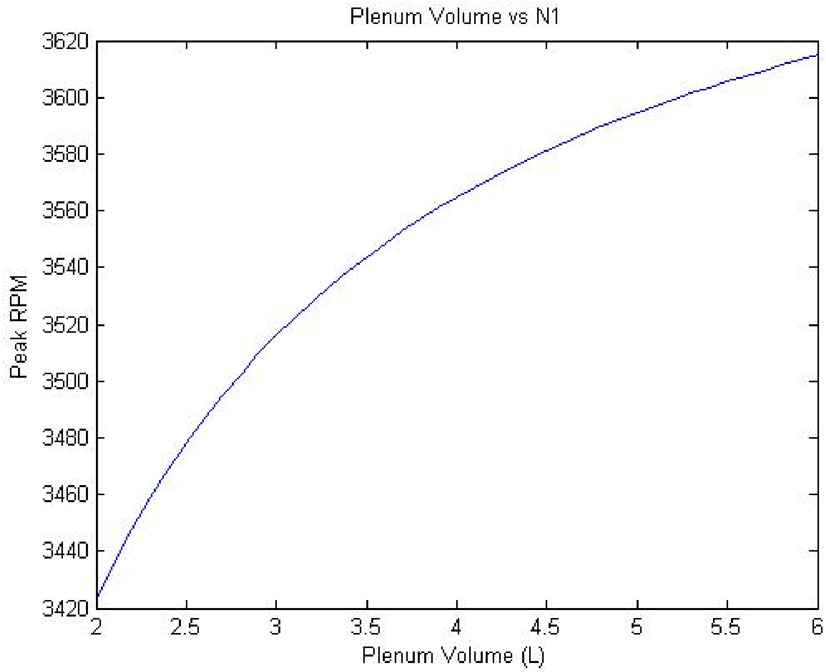


Figure 55: Plenum Volume vs First Frequency

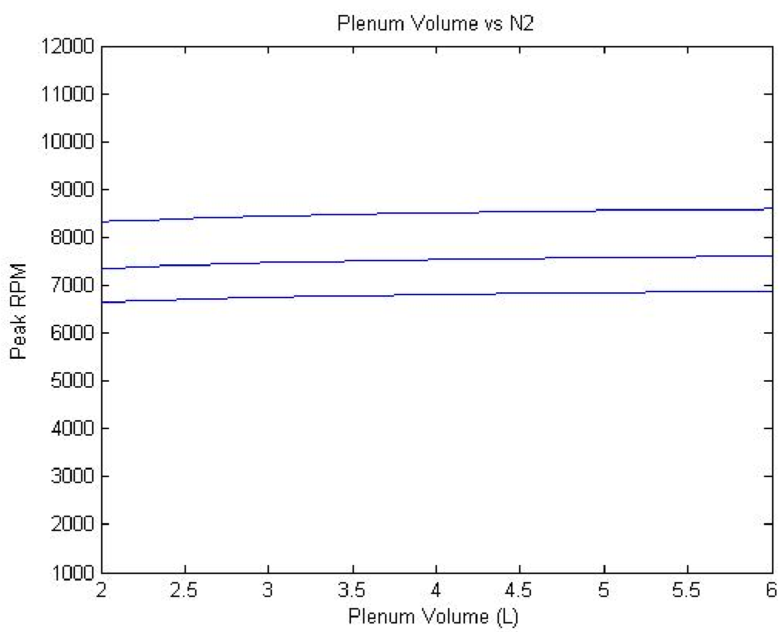


Figure 56: Runner Length vs Second Frequency

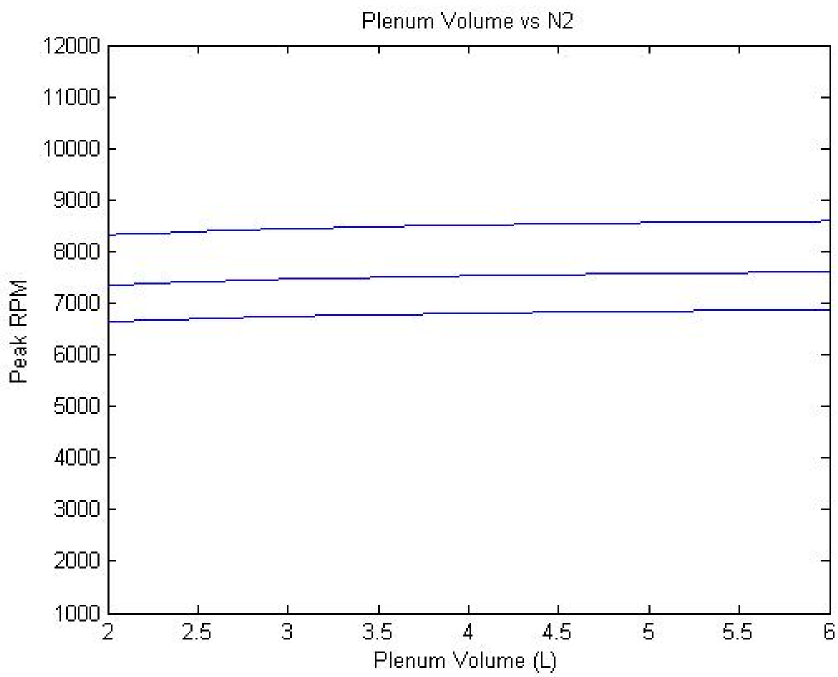


Figure 57: Plenum Volume vs Second Frequency

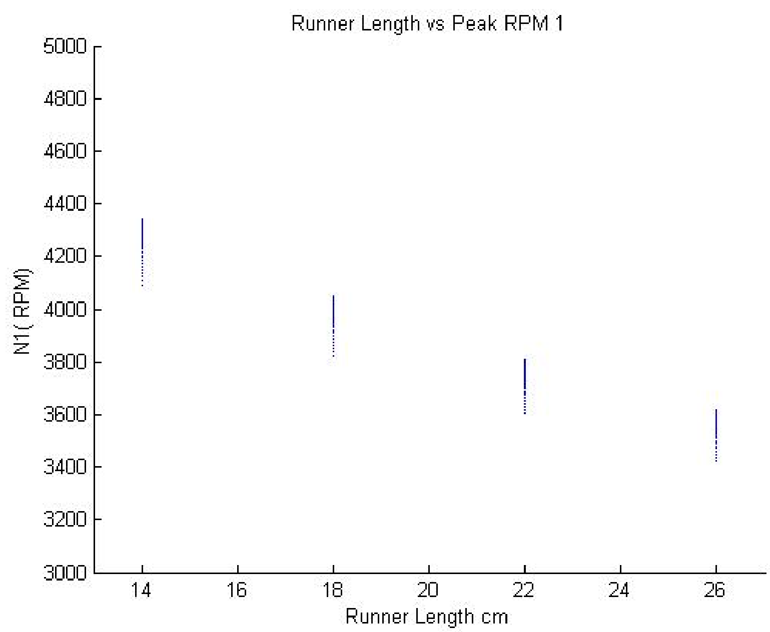


Figure 58: How Runner length and volume affects the location of the peak RPM of the first frequency

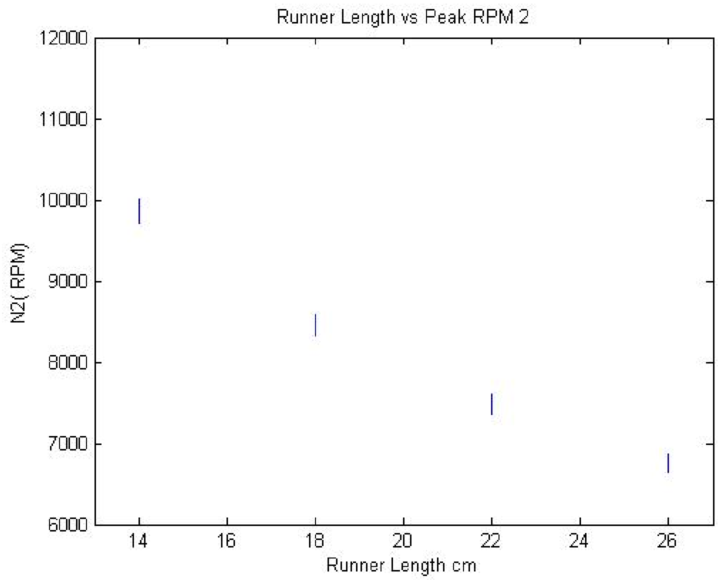


Figure 59: How Runner length and volume affects the location of the peak RPM of the 2nd frequency

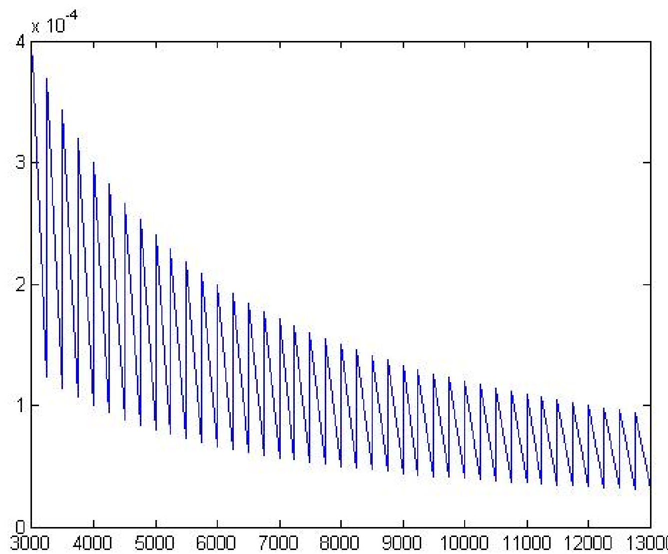


Figure 60: How Throttle Response is affected by Volume:

C.3 Flow Calculations

```
clear all
close all
clc

V_d1=(599/100^3)/4; % swept volume of each cyilnder m^3 for one cyilinder
V_d=(599/100^3); % swept volume of each cyilnder m^3 for whole system
rho_a=1.181;
N=1000:100:12000;
```

```
n_v=0.95;
n=2;
lamda=0.85;
phi=1/lamda;
AF_st=9.95;
AF=AF_st/phi;

C_D=0.7;
An=(pi*(7.95e-3)^2)/4;
rho_f=898.7; %kg.m^-3
P_rail=87000; %Pa
MAP=90000; %Pa

m_dot_a=zeros(1,length(N));
m_dot_f=zeros(1,length(N));
m_f1=zeros(1,length(N));
t=zeros(1,length(N));
omega=zeros(1,length(N));
td=zeros(1,length(N));

tic

for i=1:length(N)
    %air flow rate into one cyilinder at WOT
    m_dot_a(i)= (rho_a * V_d1 * (N(i)/60) *n_v) /n;

    %fuel flow rate into one cylinder .: the flow rate through the injector
    m_dot_f(i)=m_dot_a(i)/AF;

    %fuel flow into one cylinder for one cycle
    m_f1(i)=(m_dot_f(i)*n)/(N(i)/60);     %kg/cycle

    %time of injection
    t(i)=m_f1(i)/m_dot_f(i) ;%(s)

    %rotational speed
    omega(i)=(N(i)/60)*360; %(degrees/sec)

    %time of injection
    td(i)=omega(i)*t(i); %(degrees)


end

figure(1)
plot(N,t)
xlabel('RPM')
ylabel('Pulse Width (s)')
title('RPM vs Pulse Width (Single Cylinder)')

figure(2)
plot(N,m_dot_f)
xlabel('RPM')
ylabel('Fuel Flow rate kg/s')
title('RPM vs Fuel Flow rate (Single Cylinder)')
%%
%Fuel Injector actual

delta_P=0:400000/110:400000; %delta_p=(P_rail-MAP)

m_dot_ff=zeros(1,length(N));

for i=1:length(N)

    m_dot_ff(i)=C_D*An*sqrt(2*rho_f*(delta_P(i)));

end
```

```
figure(3)
plot(m_dot_ff,delta_P)
xlabel('Mass Flow rate (kg/s)')
ylabel('Pressure Difference (Pa)')
title('Mass Flow rate vs Pressure Change (Single Cylinder) Fuel Injector')

%% Fuel for wholse system
m_dot_a_s=zeros(1,length(N));
m_dot_f_s=zeros(1,length(N));
m_f1_s=zeros(1,length(N));
t_s=zeros(1,length(N));
omega_s=zeros(1,length(N));
td_s=zeros(1,length(N));

for i=1:length(N)
    %air flow rate into  whole system at WOT
    m_dot_a_s(i)= (rho_a * V_d * (N(i)/60) *n_v) /n;

    %fuel flow rate  whole system .: the flow rate through the injector
    m_dot_f_s(i)=m_dot_a_s(i)/AF;

    %fuel flow  wholse system for one cycle
    m_f1_s(i)=(m_dot_f_s(i)*n)/(N(i)/60);     %kg/cycle

    %time of injection
    t_s(i)=m_f1_s(i)/m_dot_f_s(i) ;%(s)

    %rotational speed  whole system
    omega_s(i)=(N(i)/60)*360; %(degrees/sec)


    %time of injection  whole system
    td_s(i)=omega_s(i)*t_s(i); %(degrees)

end
figure(4)
plot(N,t_s)
xlabel('RPM')
ylabel('Pulse Width (s)')
title('RPM vs Pulse Width (Whole System)')

figure(5)
plot(N,m_dot_f_s)
xlabel('RPM')
ylabel('Fuel Flow rate kg/s')
title('RPM vs Fuel Flow rate (Whole System)')

E85_flow=zeros(1,length(N));
m_dot_ffL=zeros(1,length(N));


for i=1:length(N)

   E85_flow(i)=(m_dot_f_s(i)/rho_f)*3600;
   m_dot_ffL(i)=(m_dot_ff(i)/rho_f)*3600;

end

figure(6)
plot(N,E85_flow)
xlabel('RPM')
ylabel('Fuel Flow rate L/hr')
title('RPM vs Fuel Flow rate L/hr (Whole System)')

plot(N,m_dot_a_s)
toc
```

C.4 Restrictor Calculations

```
clc
clear all

gamma=1.4;
M1=0.1;

Dstar=0.019;
D1=0.034;

Astar=(pi*Dstar^2)/4;
A1=(pi*D1^2)/4;

A1onAstar=A1/Astar;
ratio1=0;

% Subsonic Solution
while ratio1 < (1/A1onAstar)

term1 = (1/M1);
term2 = (2/(gamma+1));
term3 = (1 + ((gamma -1)/2)*M1^2);

ratio1 = 1/(term1*(term2*term3)^((gamma + 1)/(2*(gamma - 1))));
M1=M1+0.00001;

if ratio1 >= (1/A1onAstar)
    M1
end


end



%Super Sonic Solution
D2=0.031;
A2=(pi*D2^2)/4;
A2onAstar=A2/Astar;

M2=2;
ratio2=0;

while ratio2 < (1/A2onAstar)

term1 = (1/M2);
term2 = (2/(gamma+1));
term3 = (1 + ((gamma -1)/2)*M2^2);

ratio2 = 1/(term1*(term2*term3)^((gamma + 1)/(2*(gamma - 1))));
M2=M2+0.0001


if ratio2 >= (1/A2onAstar)
    M2
end

end
```

C.5 Throttle Body Calculations

```
clear all
clc

%Estimate of Engine response by evaluating engine breaking
N = 3000:250:13000;

 EngineDisp = 600;
```

```
for t = 1:length(N)
    Plenum3L = 3000;
    Plenum2point2L = 2200;


    %Number of displacement of the plenum volume over the engine displacement
    DispNum1 = Plenum3L/EngineDisp;
    DispNum2 = Plenum2point2L/EngineDisp;

    %Intake stroke per minute ISPM
    ISPM = N(t)/2;

    %Engine Braking Response Time [ 1/ISPM * DispNum * 60 ]
    Response1(t) = 1/ISPM * DispNum1 * 60;
    Response2(t) = 1/ISPM * DispNum2 * 60;
end

% plot(N,Response1,'r'); hold on
% plot(N,Response2);

plenum=2.0:0.1:6.0;
i=1;

for t=1:length(N)
    ISPM = N(t)/2;

    for vol=1:length(plenum)
        Disp=plenum(vol)/EngineDisp;
        response(i)=1/ISPM * Disp * 60;
        response_N(i)=N(t);
        i=i+1;
    end


end


plot(response_N,response)
```

C.6 GT Power Simulations

Figure 61 shows the intake and engine model utilised by the 2014 team. A key addition of this model is the input of bends in the system. These losses are minimal, and hence have been utilised as the computational power increase isn't noticeable and therefore included for completeness.

GT Power is an extremely useful tool that models the engine system extremely accurately. This model has been utilised to calculate the ideal runner lengths and diameters, plenum volume, diameter of ports and diffuser lengths.

As you can see from the below figures, a detailed analysis was conducted on the effect on diameters of the intake ports. The main goal of this simulation is to ensure the torque output peaks in the designed operating range. We can see that a 34mm diameter suits the best. It has the broadest torque output between 6000-9000 RPM.

A detailed study was also conducted on the mean effective pressure of the engine. MEP is used as it is independent on engine displacement and hence provides a relative engine performance measure. It is calculated by dividing the work per cycle by the cylinder volume displaced per cycle.

Work per cycle $= \frac{Pn_r}{N}$

n_R = number of crank revolutions for each power stroke. ($n_R = 2$ for four stroke engines)
N = Crankshaft rotational speed P = Power (kW)

We then divide this by displaced volume (V_d)

$$mep = \frac{Pn_r}{V_d N} \tag{C.9}$$

MEP can also be expressed in terms of Torque (Nm):

$$mep = \frac{6.28 n_r T}{V_d} \tag{C.10}$$

It is effectively calculating the pressure in the cylinder, where IMEP is the average cylinder pressure over the cycle. Gross IMEP, is the amount of work delivered to the piston over the compression and expansion strokes, per cycle per unit displaced volume. Net IMEP is the work delivered to the piston over the entire four strokes of the cycle, per unit displaced volume. Given the above definitions it is evident, that these two characteristics directly correlate to the engines ability to do work. Power and torque aren't the only the factors considered in designing an effective system. Fuel consumption must also be considered which is normally measure by fuel flow rate, $\dot{m}_f$. A more effective means is the specific fuel consumption (sfc(g/kW .h)). It measures how efficiently an engine is utilising the supplied fuel to produce work.

$$sfc = \frac{\dot{m}_f}{P} \tag{C.11}$$

A smaller value is more desirable when using this parameter. Figure 62 has two interesting areas. The first is the location for where the team has designed to operate in and second shows divergence after 11000 RPM. Obviously the discrepancy is expected in the designed range as the the runner lengths have a effect on the peak torque and its location. It is evident that lengths between 160-210mm are the most beneficial. To select an appropriate length torque and power behaviour within this region needs to be analysed. Figure 65 eliminates 160mm and 170mm lengths as these torque curves are more beneficial if a broader torque curve is required. With evidence suggesting a faster course, the 210mm and 220 mm curves taper of to quickly which may cause issues at hgih speeds where MUR2014 have calculated a high drag situation. To decide on the correct runner length it can seen from figure 66 a length of 190mm produces more power between 6000-9000 RPM.

To further support this selection, figures 63 and 63 show both the net and gross IMEP are highers for the 190mm length. As mentioned above this means that a higher pressure is contained within the engine cylinders. This increases the engines ability to do work and hence increases efficiency more importantly volumetric efficiency as shown in figure 67.

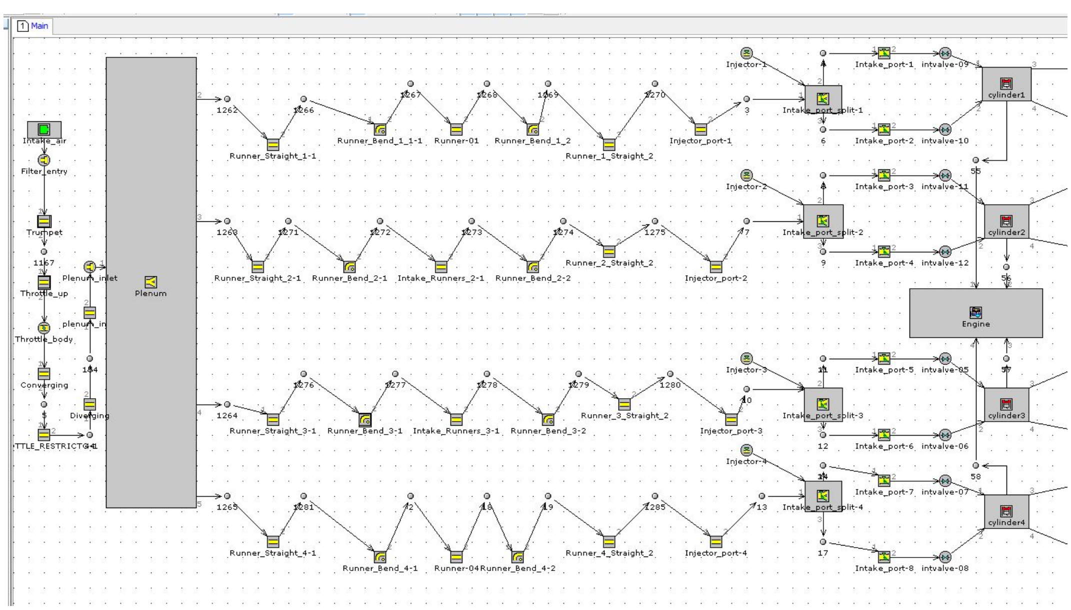


Figure 61: GT Power Model of Intake and Engine

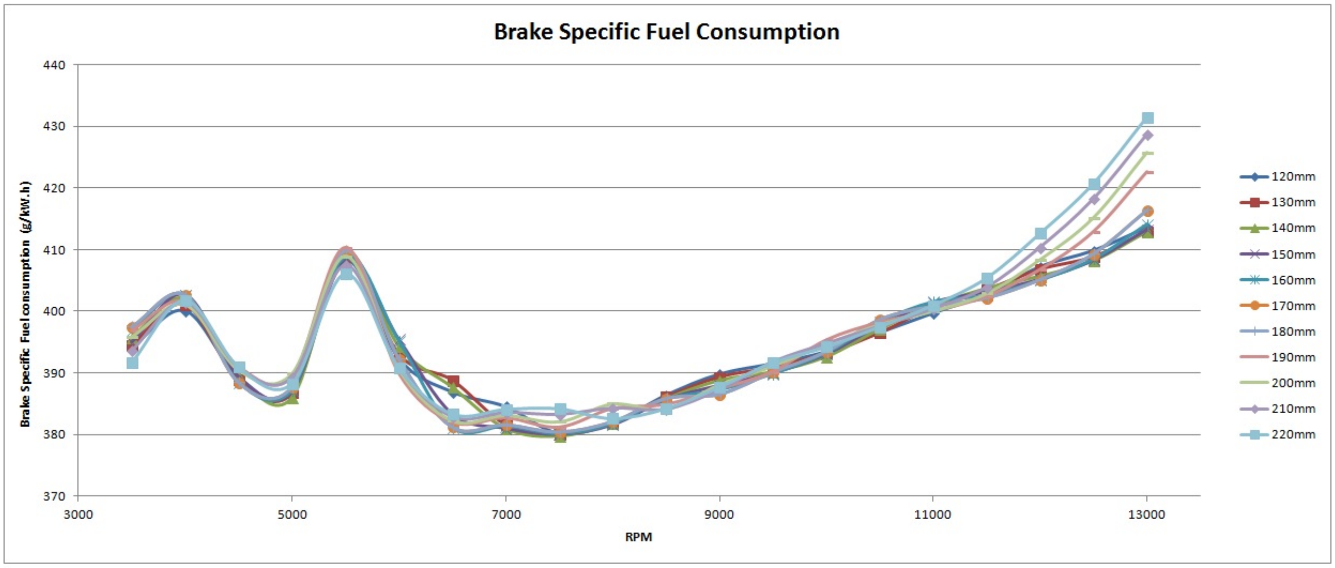


Figure 62: GT Power Results for Brake Specific Fuel Consumption with Varying Runner Length

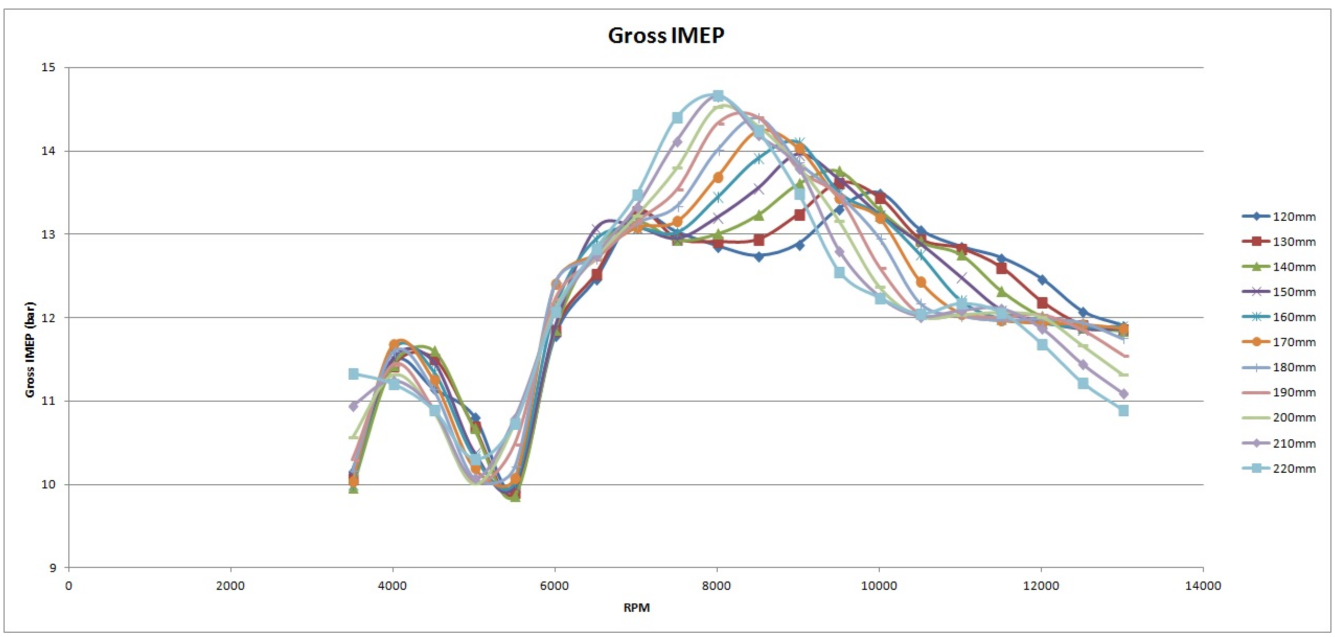


Figure 63: GT Power Results for Gross Indicated Mean Effective Pressure with Varying Runner Length

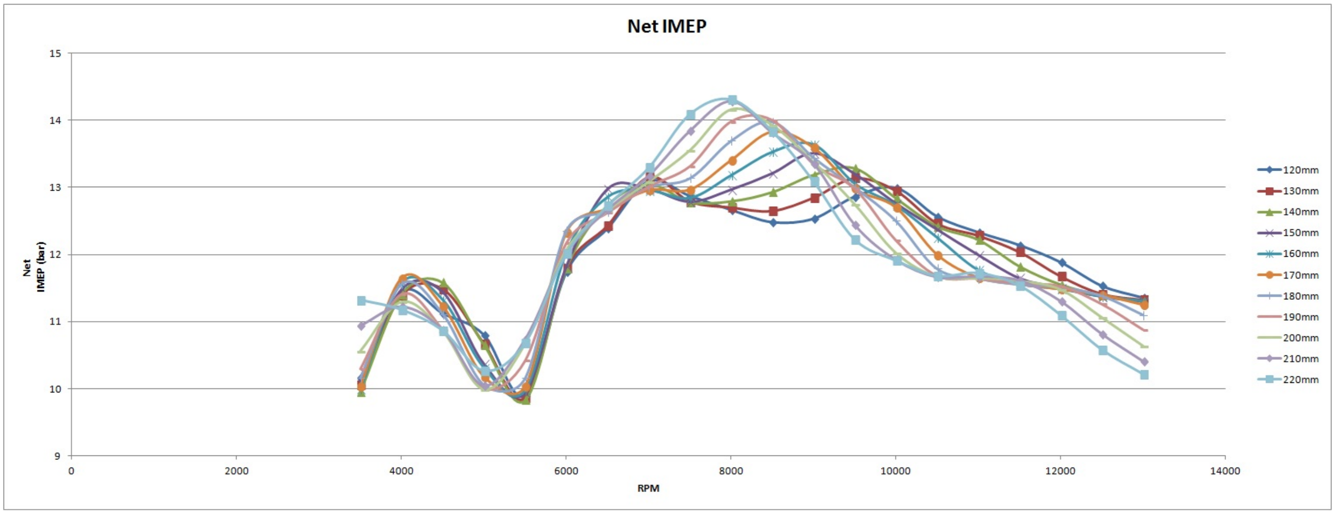


Figure 64: GT Power Results for Net Indicated Mean Effective Pressure with Varying Runner Length

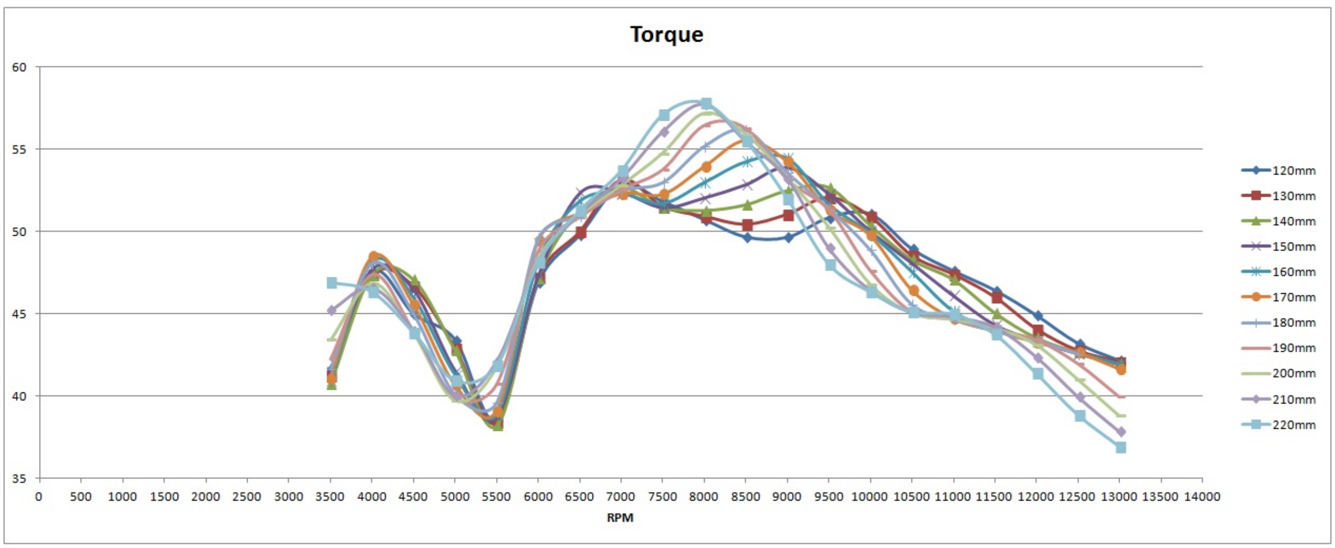


Figure 65: GT Power Results for Torque with Varying Runner Length

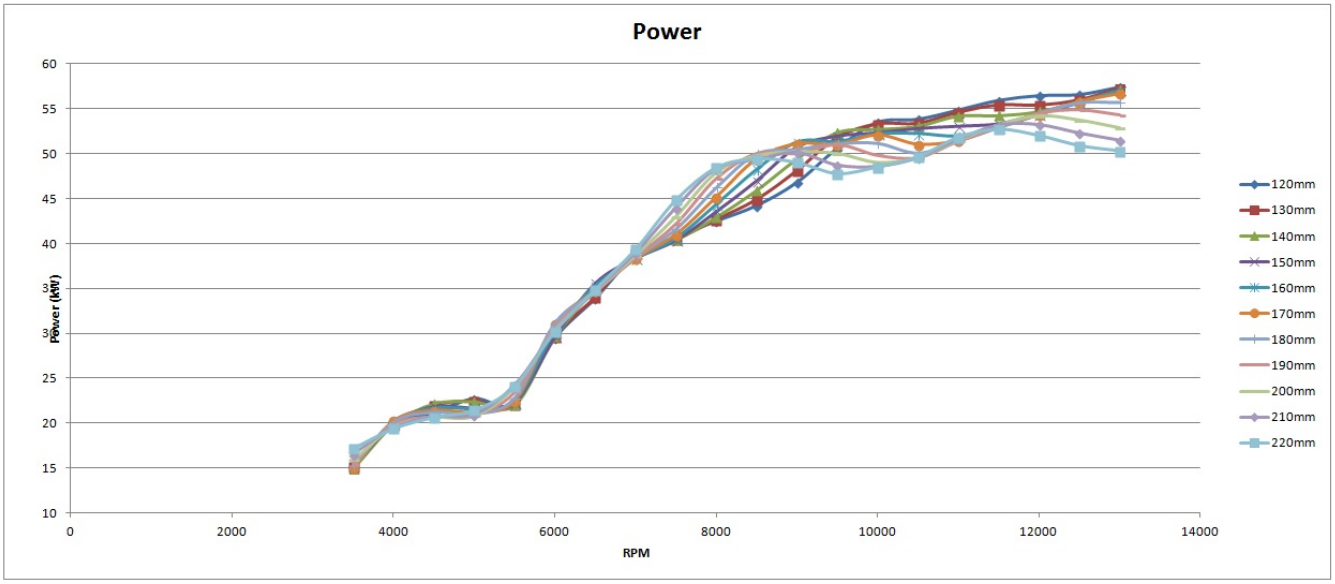


Figure 66: GT Power Results for Power with Varying Runner Length

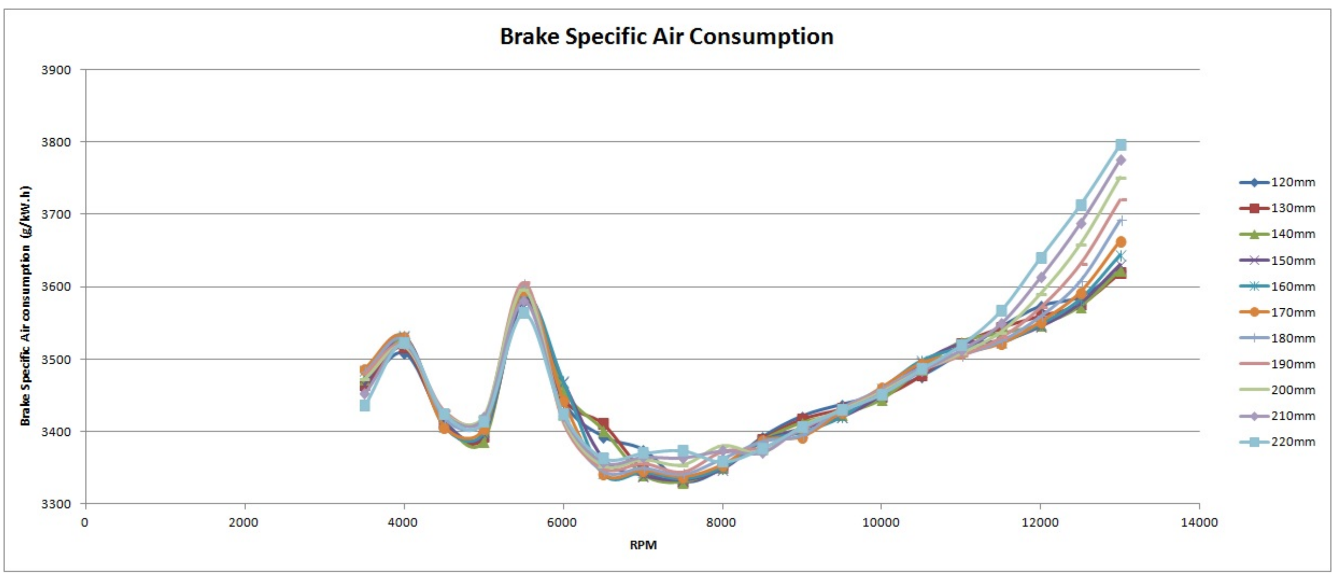


Figure 67: GT Power Results for Brake Specific Air Consumption with Varying Runner Length

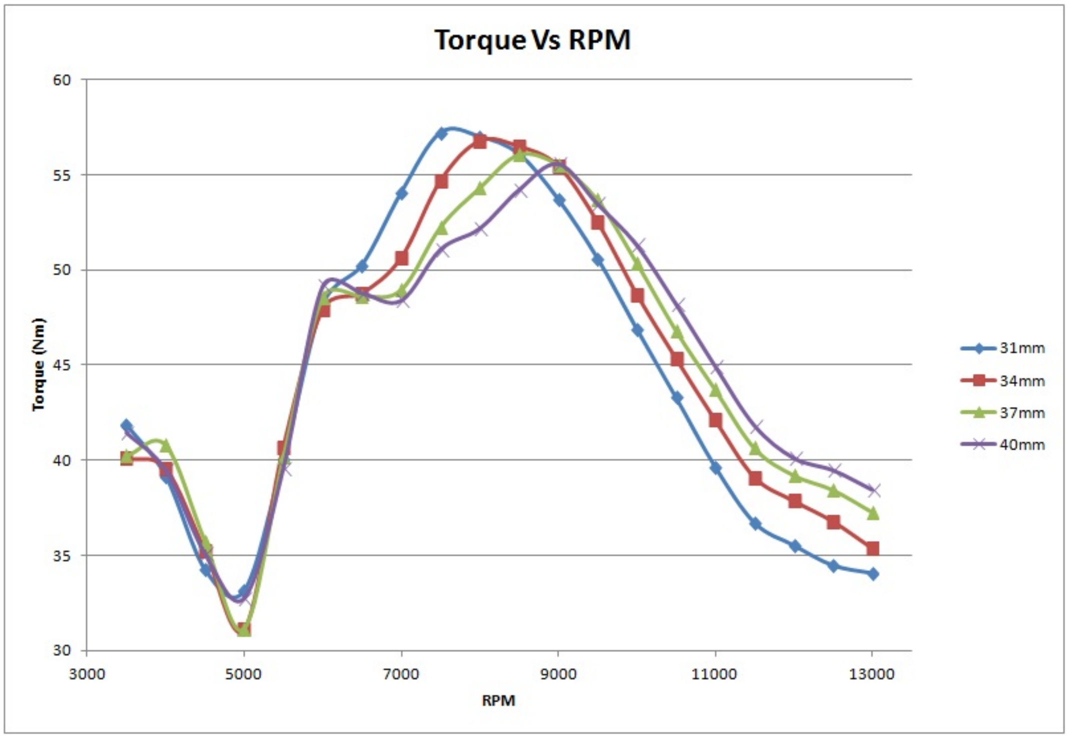


Figure 68: GT Power Torque Results with Varying Intake Port Diameter

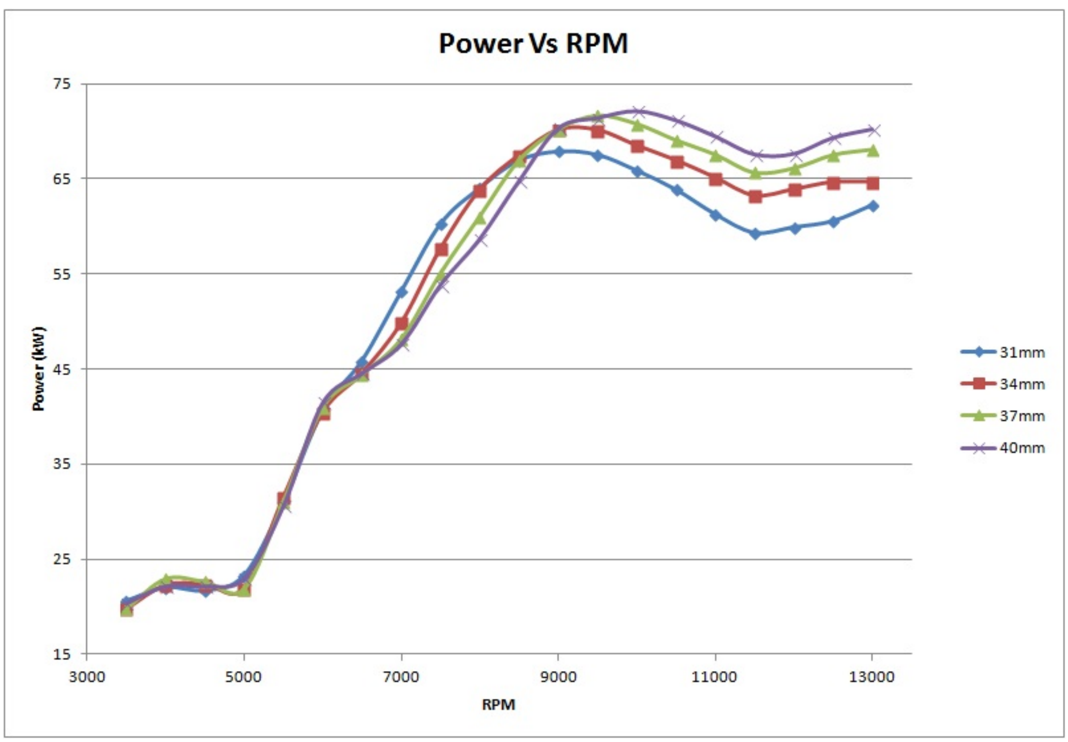


Figure 69: GT Power, Power Results with Varying Intake Port Diameter

Length(mm)	170	180	190	200	210	220
Average 7000-9500 (kw)	44.97543	45.6868	45.905	46.0671	46.15462	46.15877
Average 6000-9000 (kw)	39.80447	40.55653	40.71731	40.85929	41.03164	41.18164

Table 21: Average Power in Operating Ranges

C.6.1 2014 GT Power Plenum Volume Analysis

As mentioned previously, volumetric efficiency (VE) is a key component in designing naturally aspirated engines. Ideally a coupled simulation would yield more accurate results, however, given the computational expense of such an analysis, analysing a variety of volumes is unreasonable. Also given that the purpose of these simulations was to illustrate the differences in designs quantitatively, it became more apparent that such an analysis would be useless. Theoretically we would expect that an increase in volume increase should increase the volumetric efficiency. Using a simple filling and emptying air flow model we can determine the rate of change of mass in the system as shown in equation C.12.

$$\frac{dm_{a,m}}{dt} = \dot{m}_{a,th} - \sum \dot{m}_{a,cyl} \tag{C.12}$$

We can use a quasi-steady approximation to determine the manifold response to to various throttle and load inputs.

$$\sum \dot{m}_{a,cyl} = \frac{\eta_v \rho_{a,m} V_d N}{2} \tag{C.13}$$

Assuming a constant temperature we can use the ideal gas law for a manifold, equation C.14 we get a relationship for how the mass changes with time, equation C.15

$$p_m V_m = m_{a,m} R_a T_m \tag{C.14}$$

$$\frac{dp_m}{dt} + \frac{\eta_v V_d N}{2V_m} p_m = \dot{m}_{a,th} \frac{RT_m}{V_m} \tag{C.15}$$

More mass results in denser air and hence an increase in volumetric efficiency. Figure 72 confirms our predictions, but also provides insights where the equations fail. The above equations cannot determine accurately the distribution of VE over the RPM range due to a lack of information of how the above parameters change over RPM. Figure 71 includes fuel in its VE calculations.

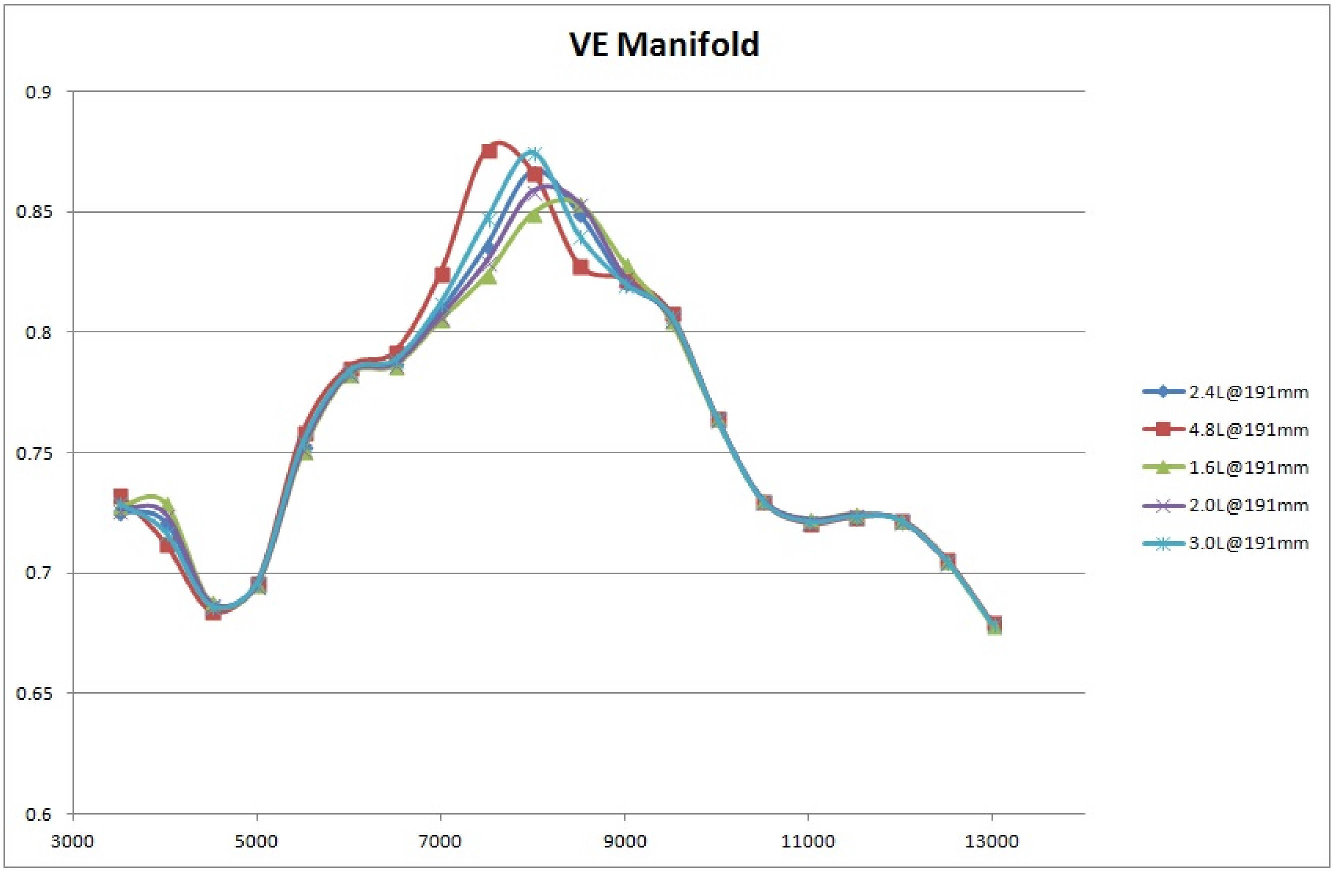


Figure 70: GT Power Results for Manifold Volumetric Efficiency with Varying Plenum Volume

Figure 70 shows a direct correlation between volume and the maximum VE. A larger volume will provide a greater amount of air available to engine hence a higher pressure difference for longer. Using E85 will reduce the VE that is achieved. This is due to a larger quantity of fuel required to be injected which will displace the amount of air in the engine cylinders. Figure 71 is an essential performance measurement as it shows the VE distribution of the fuel-air mixture.

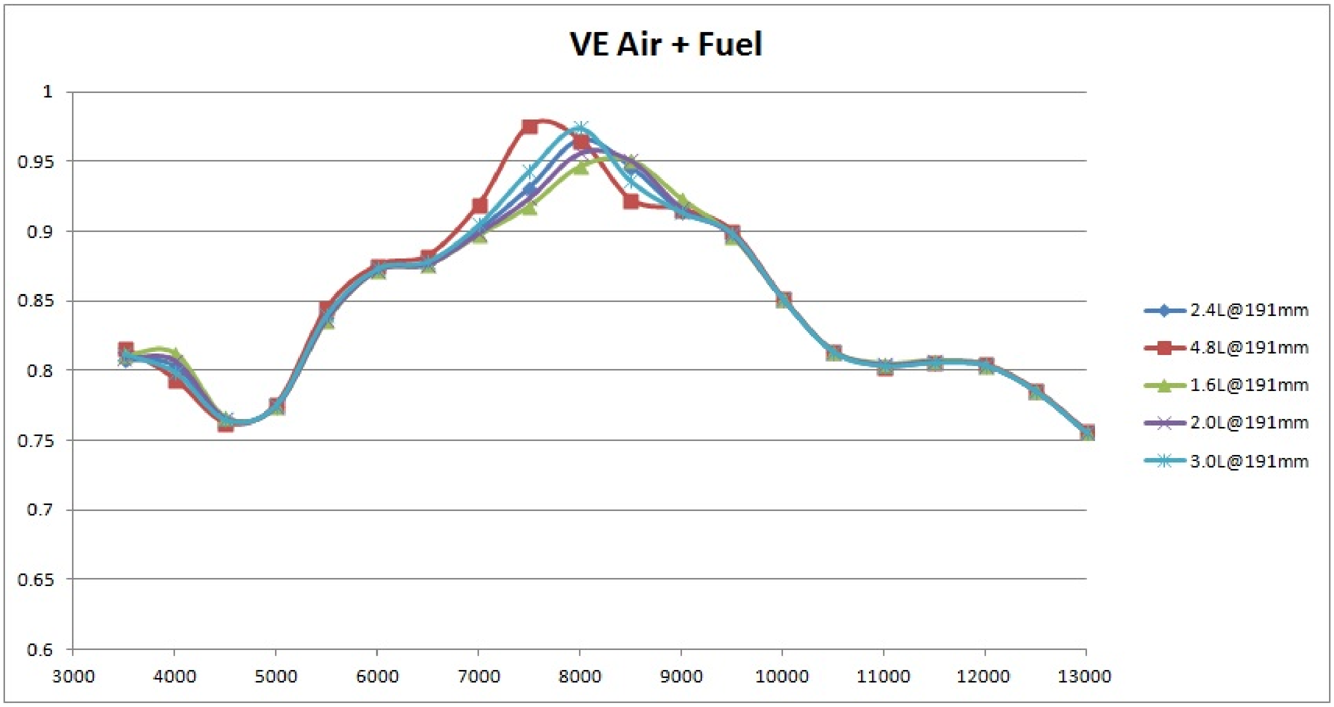


Figure 71: GT Power Results for Air and Fuel Mixture Volumetric Efficiency with Varying Plenum Volume

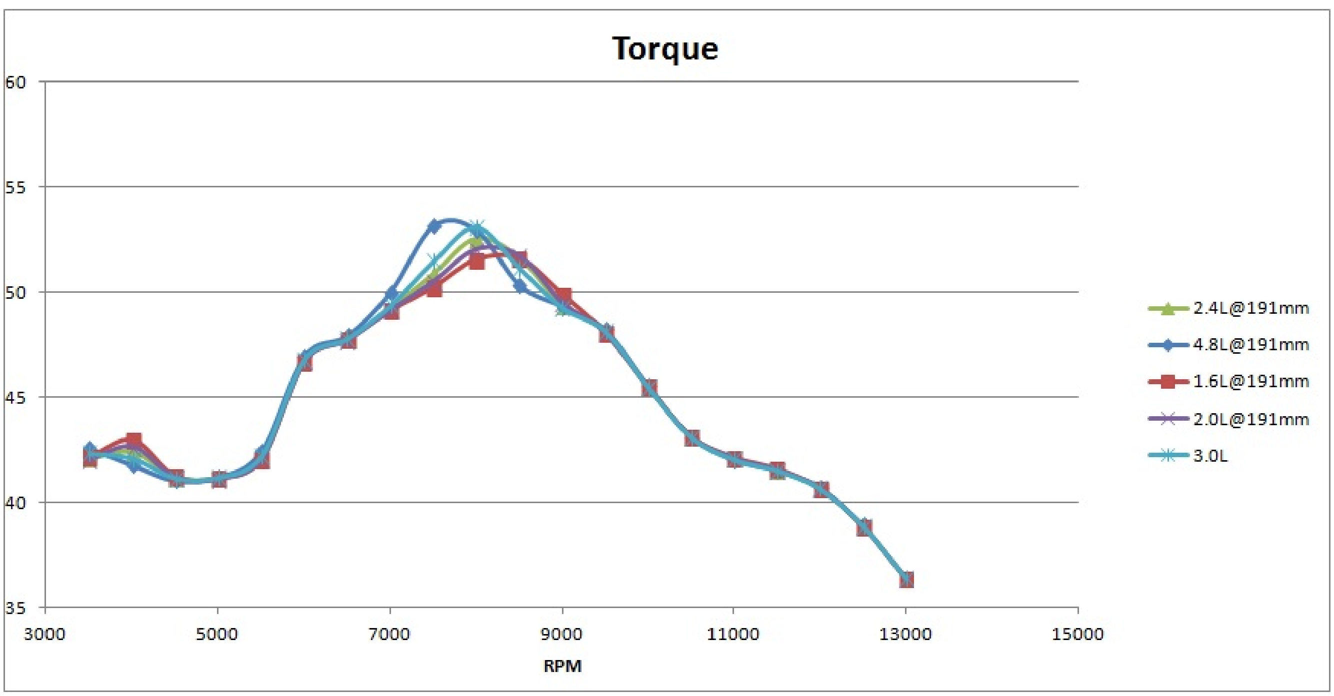


Figure 72: GT Power Results for Torque with Varying Plenum Volume

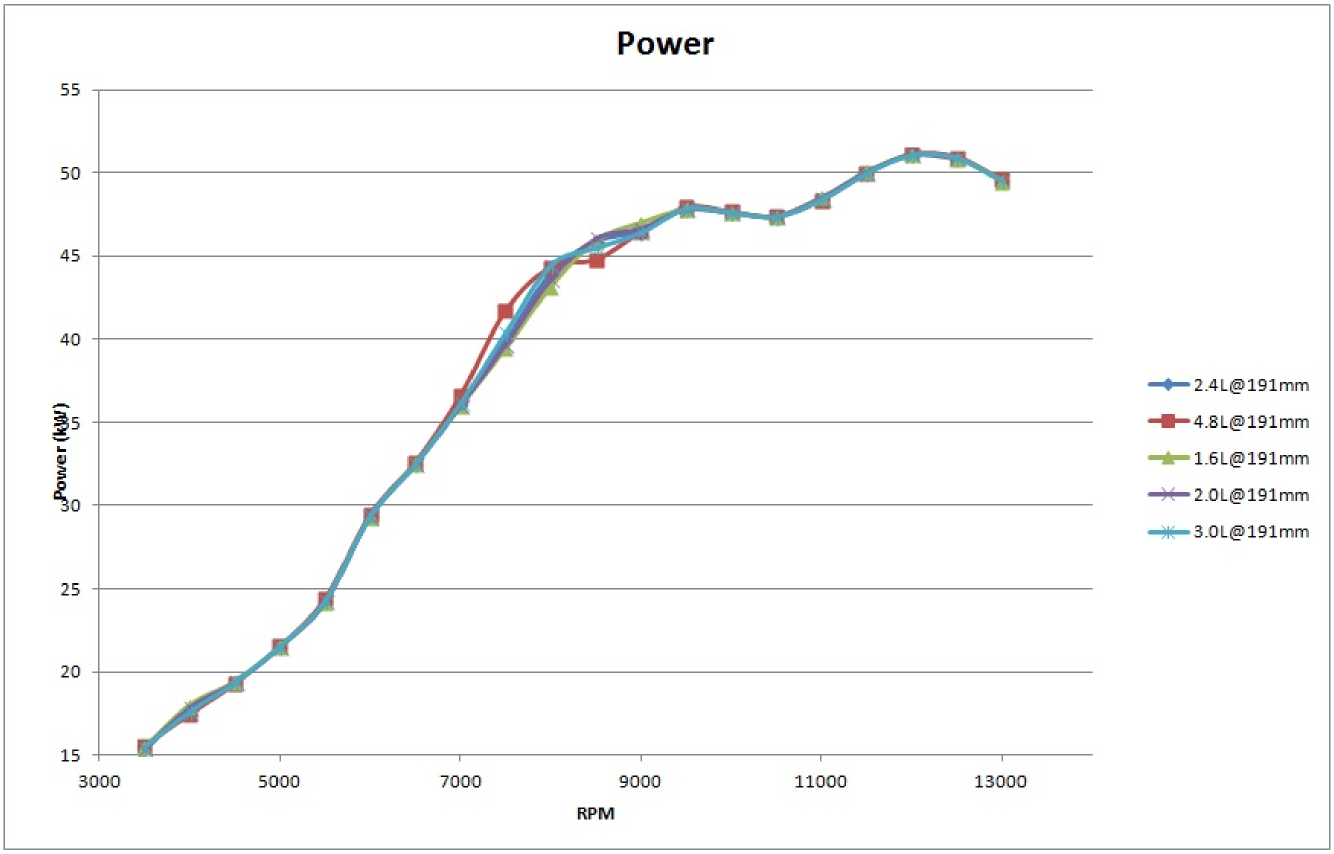


Figure 73: GT Power Results for Power with Varying Plenum Volume

The torque and power curves are shown in figures 72 and 73. A larger plenum volume seems to be only effecting the torque and power putput between 6500 - 9000 RPM. This result is surprising since a higher mass flow rate is required at higher RPM. A larger plenum volume should be at more of an advantage at higher RPM given that more air is accessible to the engine. A coupled simulation needs to be conducted and then tested to ensure accuracy. We can see that ideally we would require a plenum volume between 2.4-3.0L in order to achieve maximum performance between 6000-9000 RPM. A 2.4L plenum has a lower peak torque, power and VE as well as being effective at a higher RPM range. The inverse is true for a larger volume. Testing on various plenum volumes needs to be tested and verified to ensure to correct one is selected for our use.

C.7 Fluent/CFX Simulations

ANSYS CFX is used to obtain flow characteristics through the intake manifold. In order to ensure validity of the simulation, an understanding of the theory and equations used is required, especially information regarding turbulence and heat transfer models.

With the aid of ANSYS documentation (31), we were able to understand the various turbulent models. Below is a summary of various models.

One of the most prominent turbulence models, the k-epsilon model, has been implemented in most general purpose CFD codes and is considered the industry standard model. It has proven to be stable and numerically robust and is established as a dominant model for its capability . For general purpose simulations, the κ - ϵ model offers a good compromise in terms of accuracy and robustness.

Within CFX, the k-e turbulence model uses the scalable wall-function approach to improve robustness and accuracy when the mesh around the wall is extremely small. The scalable wall functions enable solutions on arbitrarily fine near-wall grids. This a significant advancement over standard wall functions both computationally and accuracy. While standard two-equation models, such as the k-e model, provide good predictions for many engineering applications, there are applications for which these models may not be suitable. Among these are:

- Flows with boundary layer separation
- Flows with sudden changes in the mean strain rate
- Flows in rotating fluids
- Flows over curved surfaces

A Reynolds Stress model may be more appropriate for flows with sudden changes in strain rate or rotating flows, while the SST model may be more appropriate for separated flows. This is a key factor for the simulation of the intake. Flow separation introduces pressure losses as shown in is section 7.5. These factors are a major contributor to flow distribution, efficiency and volumetric efficiency imbalance.

In the first simulation we used the Shear Stress Transport (SST) Model. It is based upon the κ-ω model. The BSL model and the κ - ϵ model do not account for the transport of the turbulent shear stress and results in an over prediction of the eddy-viscosity. SST contains two blending functions which restricts the limiter to the wall boundary layer

The Reynolds stress model is based upon all components of the Reynolds stress tensor and the dissipation rate. These models do not use the eddy viscosity hypothesis. It solves an equation for the transport of Reynolds stresses in the fluid. In particular the BSL Reynolds stress Model has various coefficients blended which correspond to the ω & ϵ based models.

Based upon the above information the SST model and the BSL and SSG models seem to be the most beneficial. The BSL is better for lower Reynolds number. However the SST model was selected for the CFX and fluent simulations. Not only is it recommended by the software package but is ideal for its solver and captures the most information for the design parameters available.

C.8 Restrictor Simulation

ANSYS Fluent has been used to solve the fluid equations for this simulation. ANSYS incorporates its own meshing program and its results are shown in figure 74 and Tables 22 and 23. The key feature of this type of simulation is that it is well documented and can be described by various fluid mechanic and thermodynamic principles.

Figure 75 shows the velocity profile through the restrictor. It is in essence a converging-diverging pipe with laminar to turbulent characteristics. Viewing the throat area as the entrance region we can see the development of the velocity profile. We see the initial turbulence that is created which is affirmed in figures 77 and 78. They show an increase in both turbulent kinetic energy and turbulent viscosity at the throat which prevents the profile from being fully developed. As the air continues through the diverging pipe the total pressure drop decreases due to decreasing wall shear stress as shown in figure 76 and in figure 80 we see the total pressure gradient decreasing. This decrease allows the velocity profile to fully develop.

It is the goal of these simulations to see the effect of the total pressure loss in the system with different geometric configurations. The results overwhelmingly support the idea that down stream conditions have drastic effects on the losses on the restrcitor.

There are numerous papers that analyse the performance of restrictors. However we decided to complete our own model as a learning tool as well as playing an essential role in model verification, calculating losses and determining the overall effectiveness of our system. Table 3 shows the effect of varying diffuser diameters and lengths with and without the inclusion of slots to trip boundary layers.

A key feature of the pressure loss is the effect of Reynolds number and angles of the restrictor. This effect is described in section C.9 as these parameters are adjustable for the diffuser through the use of rapid prototyping.

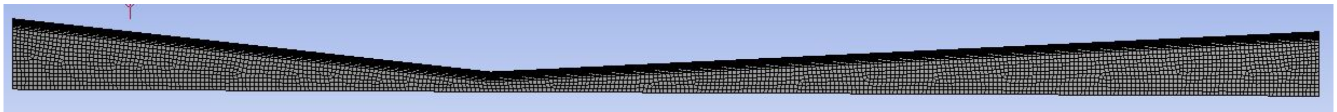


Figure 74: Mesh Visualisation of Restrictor

Advanced Size function	Curvature
Relevance Centre	Fine
Smoothing	Medium
Min Size	0.091556 mm
Max Face Size	2.0 mm
Max Size	5.0 mm
Inflation Layers	17
Inflation Max Thickness	5.0mm
Nodes	8296
Elements	7950

Table 22: Mesh Set Up

Orthogonal Quality	
Min	0.82888
Max	0.99999
AVG	0.9971
Standard Diviation	8.5727 e-03

Table 23: Mesh Statistics

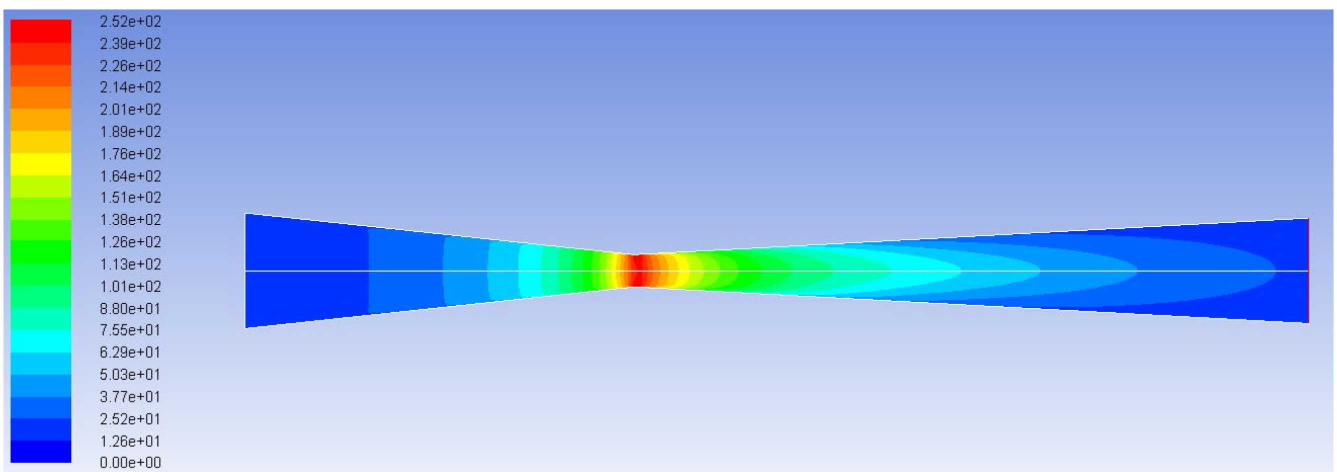


Figure 75: Velocity Magnitude Visualisation

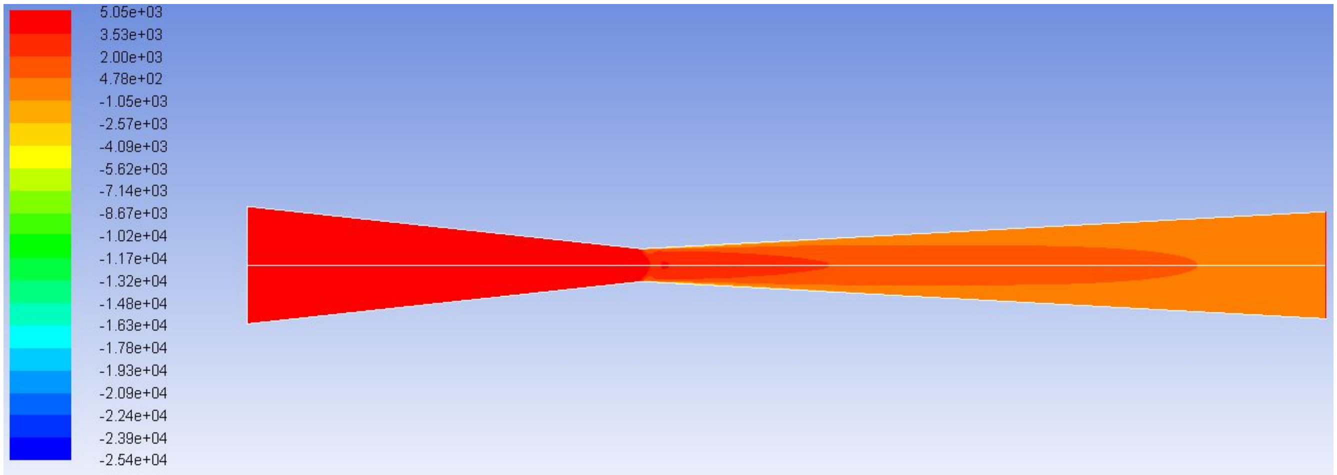


Figure 76: Total Pressure Visualisation

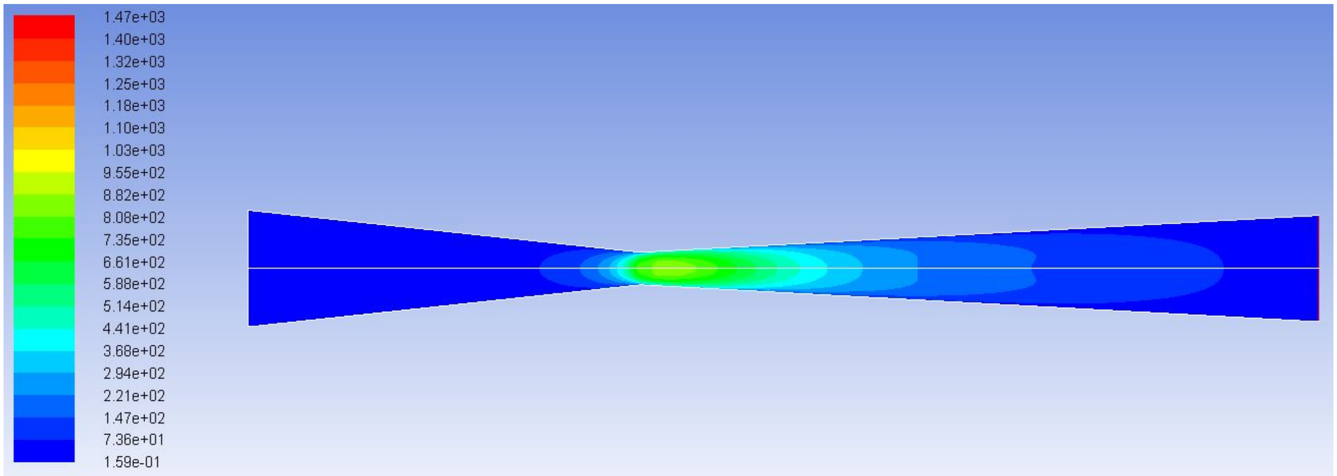


Figure 77: Turbulent Kinetic Energy Visualisation

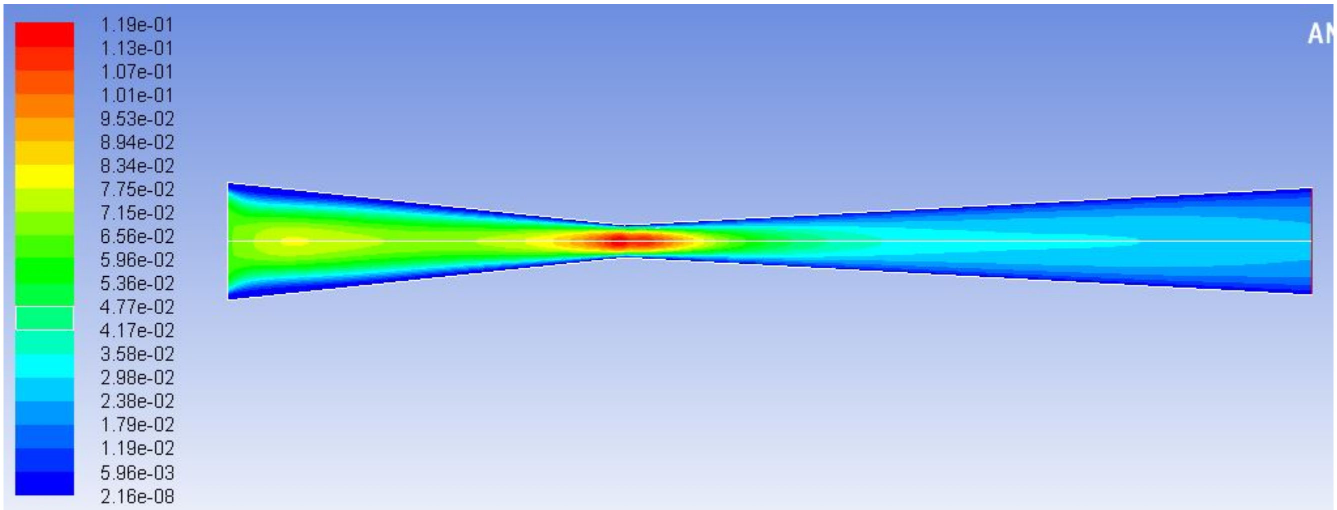


Figure 78: Turbulent Kinetic Energy Visualisation

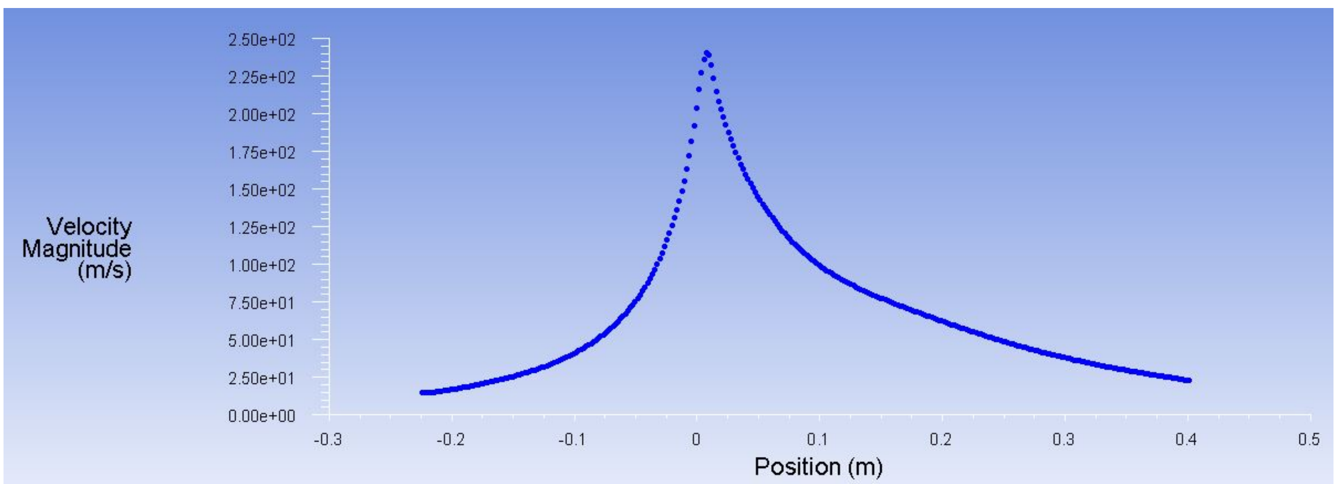


Figure 79: Velocity Magnitude Plot

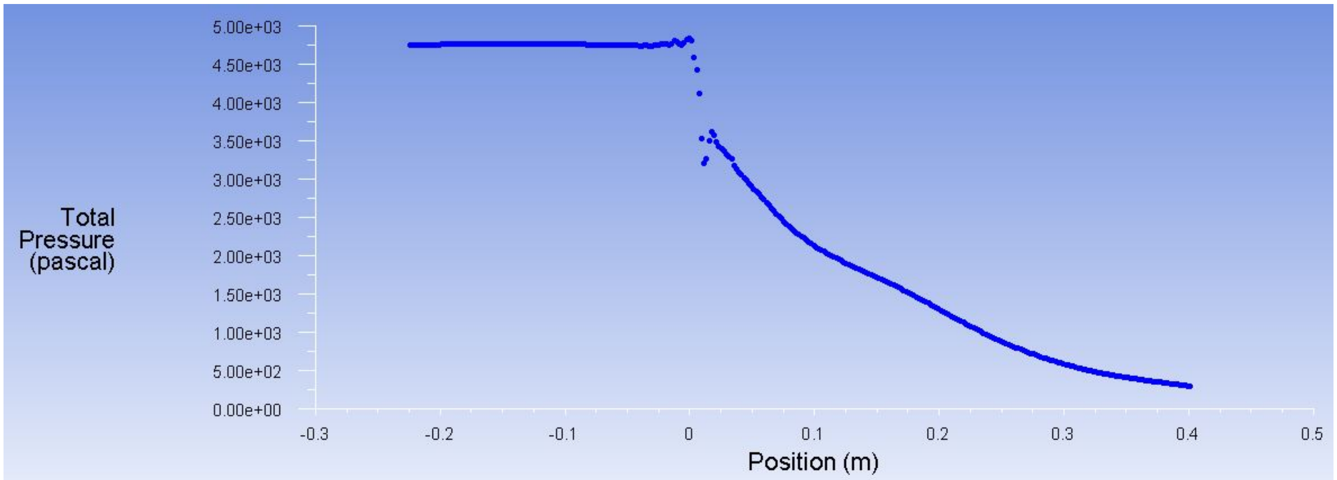


Figure 80: Total Pressure Plot

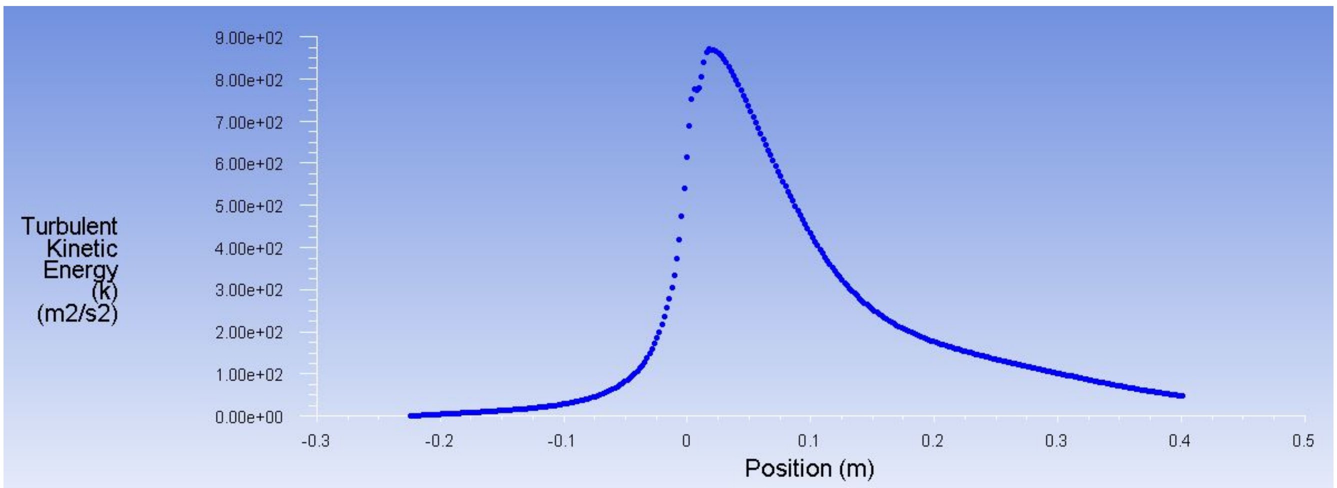


Figure 81: Turbulent Kinetic Energy Plot

C.9 Diffuser Theory and Simulations

Flow separation is a major cause of pressure drop in a diverging duct. The purpose of a diffuser is to reduce the pressure and provide laminar flow at a high velocities to the manifold. It is for these reasons that the diffuser has been a key focus for MUR2014.

Fluid flow for internal pipe flow is essential in understanding how flow separation can be avoided. As fluid enters a pipe there are two mechanisms that cause fluid near the top of the boundary layer to drag . Viscous forces maybe large enough to effect the fluid near the solid boundary and cause drag effects. These effects are important especially in the presence of a thin boundary layer. The velocity gradient normal to the surface becomes large and hence so does the shear stress. Fluid flow becomes turbulent when there is an increase in momentum in the boundary layer. This occurs when the viscous shear stresses and velocity gradient gets smaller and hence the fluid begins to rotate.

Diffusers create a positive pressure gradient in the direction of flow. As the flow passes through the diverging section, fluid at the edges and outside of the boundary layer will reach a point where it generates enough momentum to overcome the pressure trying to push it. This reduces the momentum within the boundary layer and may cause

it to stop or cause revered flow. If reversed flow occurs, the boundary layer shifts away from the contours of the surface and results in boundary layer separation.
The results show that a decrease in diffuser diameter increases the Reynolds number and allows the flow to stick to the contours of the diffuser.

Further analysis on diffuser design can be completed by utilising Bernoulli's equation:

$$\rho\frac{c_1^2}{2} - \rho\frac{c_2^2}{2} - \Delta p = p_2 - p_1 \tag{C.16}$$

p_1, p_2, c_1 and c_2 are measured values and the pressure loss due to viscous and wall friction is described by Δp.

The loss coefficient is defined as:

$$K_d = \frac{\Delta p}{\rho\frac{c_1^2}{2}} \tag{C.17}$$

However, for diffuser design, the key constraint is the pressure difference, more specifically the pressure recovery coefficient

$$C_p = \frac{p_2 - p_1}{\rho\frac{c_1^2}{2}} \tag{C.18}$$

Dividing by $\rho\frac{c_1^2}{2}$ we get:

$$1 - \frac{c_2^2}{c_1^2} - K_d = C_p \tag{C.19}$$

Using the continuity equation $c_1\ A_1 = c_2\ A_2$:

$$1 - \frac{A_1^2}{A_2^2} - K_d = C_p or \frac{1}{AR^2} - K_d = C_p \tag{C.20}$$

AR is defined as the area ratio of the inlet to outlet. Given that the diffuser is connected to a plenum we can assume c_2=0 hence reduces our equation to:

$$1 - K_d = C_p or K_d + C_p = 1 \tag{C.21}$$

Finally we can define the diffuser efficiency as a ratio:

$$\eta = \frac{actual static pressure recovery}{ideal static pressure recovery} = \frac{p_2 - p_1}{\frac{\rho}{2}(c_1^2 - C_2^2)} = \frac{p_2 - p_1}{\frac{\rho}{2}c_1^2(1 - \frac{c_2^2}{c_1^2})} = \frac{p_2 - p_1}{\frac{\rho}{2}c_1^2(1 - \frac{A_1^2}{A_2^2})} = \frac{C_p}{C_p i} \tag{C.22}$$

Given that the diffuser connects to a plenum we let c_2= 0

$$\eta = C_p \tag{C.23}$$

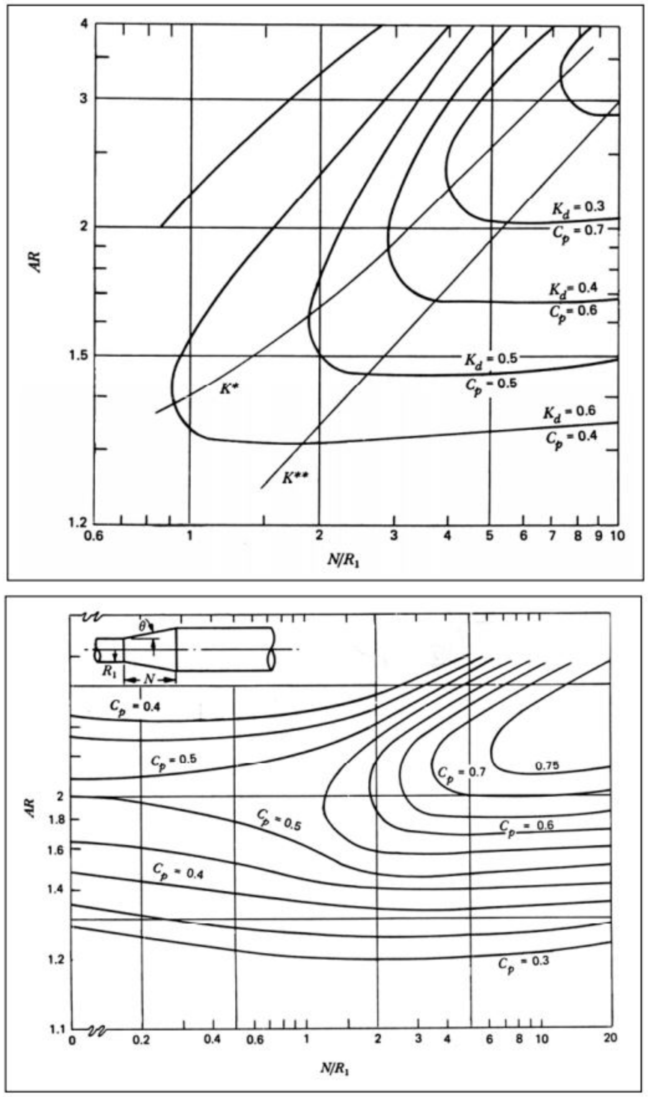


Figure 82: Pressure Loss Graphs

These charts will give a value for the pressure loss or pressure recovery coefficients depending on the length and radius ratio and the Aarea ratio. On the first chart, for a free-discharge diffuser, are two lines K_d^* and K_d^**. The K_d^* line defines the area ratio producing the minimum pressure loss for a specified N/R1 value. In comparison the K_d^** line defines the N/R1 ratio producing the minimum total pressure loss from the system given a specified area ratio.

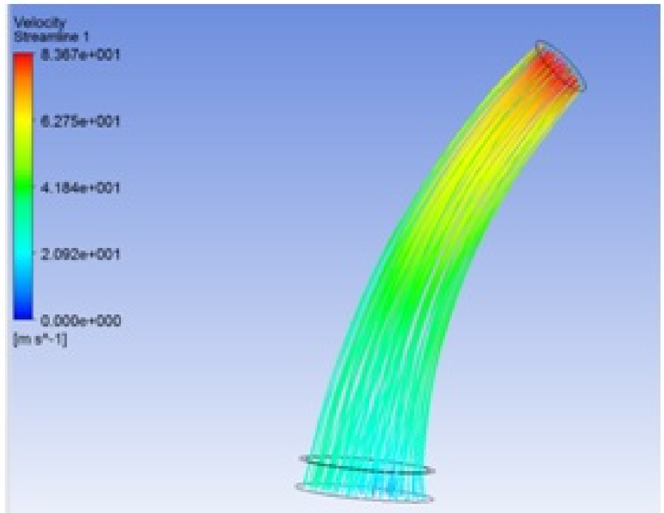


Figure 83: 2013 Diffuser Velocity Distribution

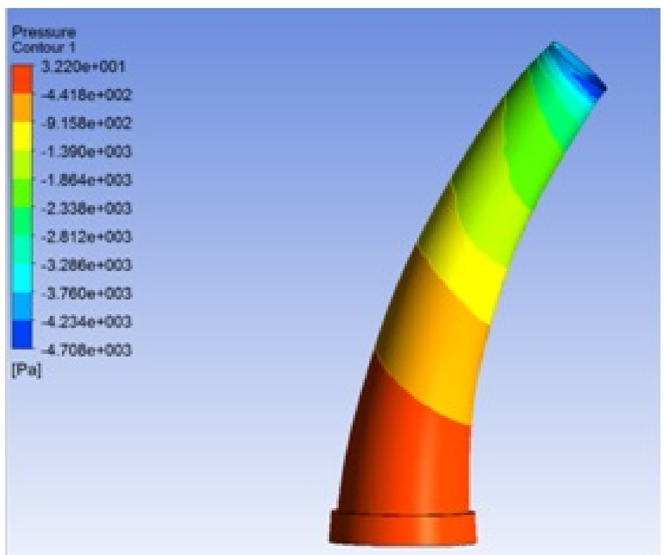


Figure 84: 2013 Diffuser Pressure Distribution

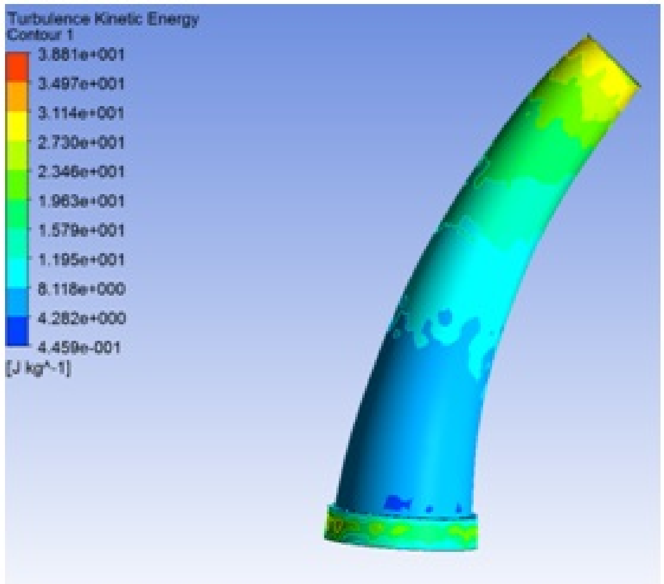


Figure 85: 2013 Diffuser Turbulent Kinetic Energy Distribution

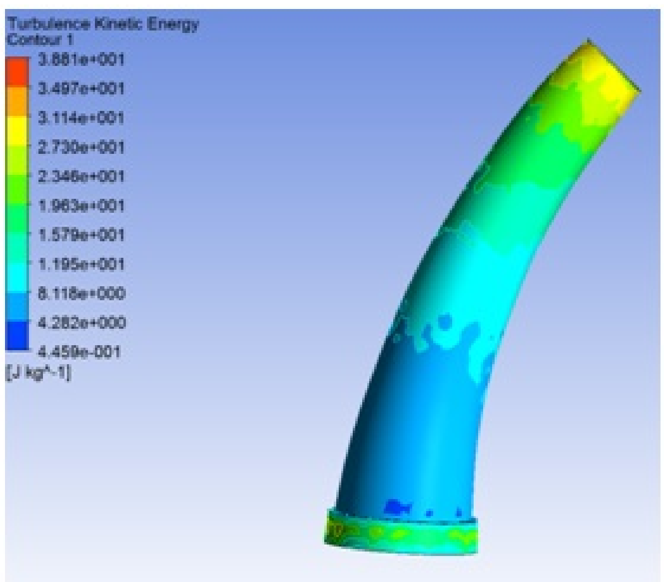


Figure 86: 2013 Diffuser Turbulent Kinetic Energy Distribution

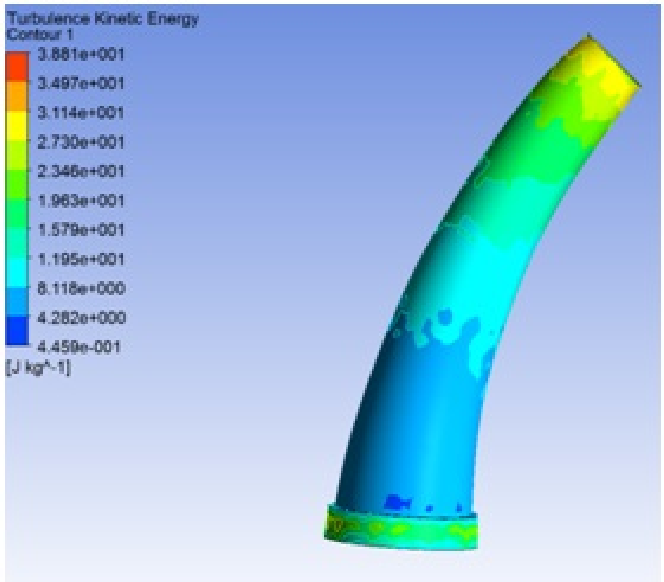


Figure 87: 2013 Diffuser Turbulent Kinetic Energy Distribution

Figure 88: Slot placed in diffuser for 1D/3D coupling

C.10 Steady State Intake Simulations

To ahceive the team goal

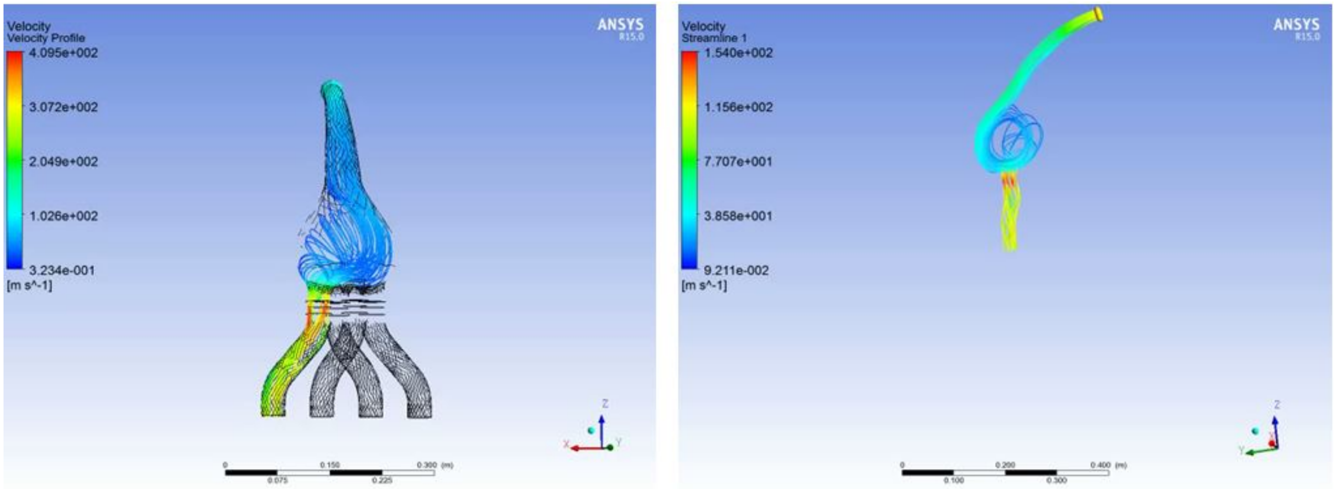


Figure 89: Steady-state flow through Runner 1

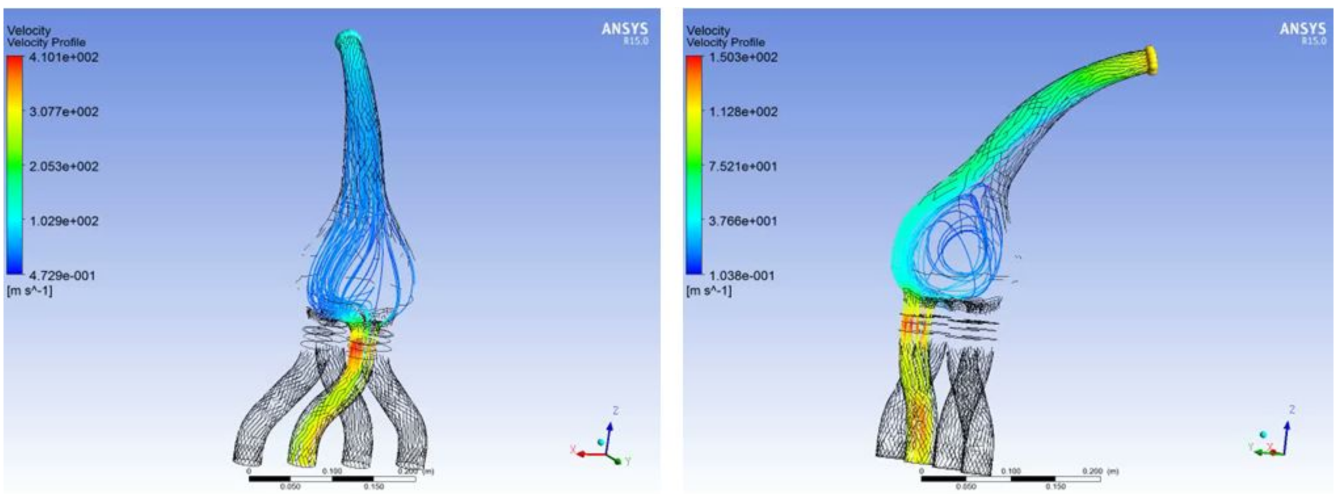


Figure 90: Steady-state flow through Runner 2

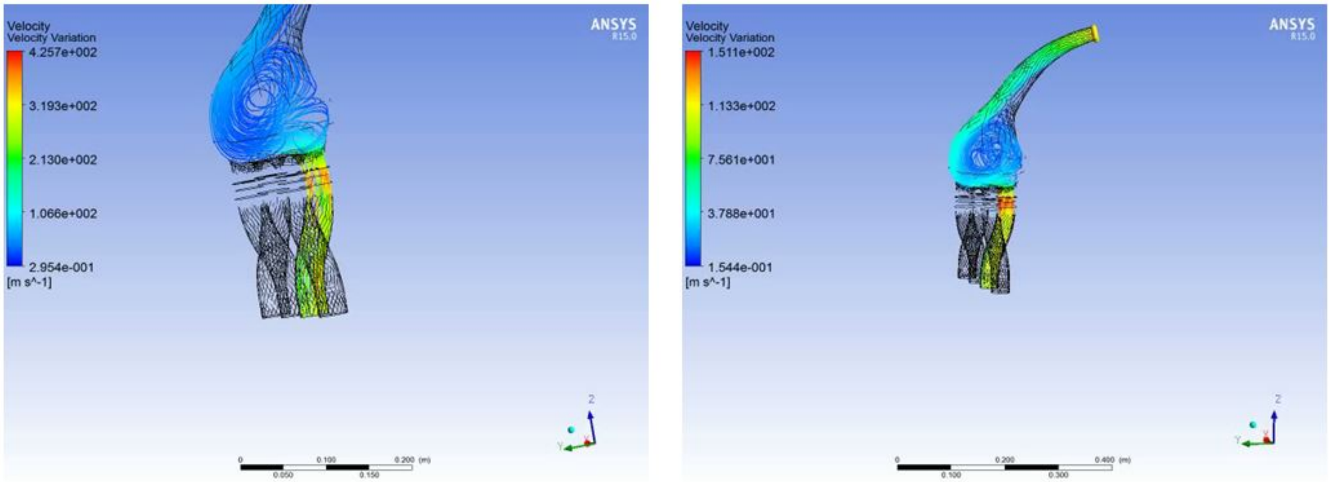


Figure 91: Steady-state flow through Runner 3

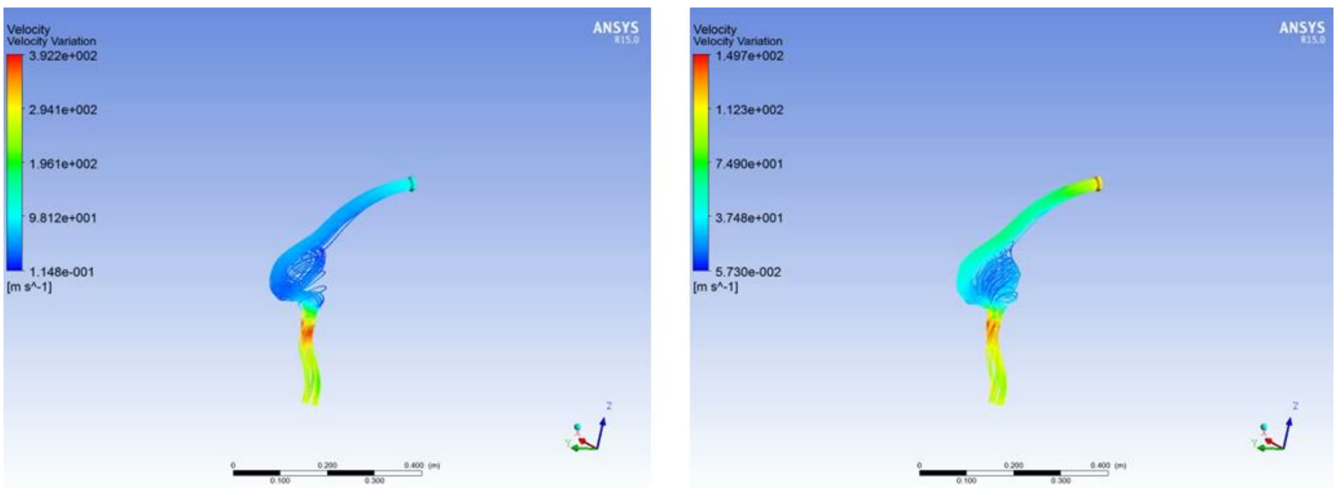


Figure 92: Steady-state flow through Runner 4

C.11 Coupled Simulation

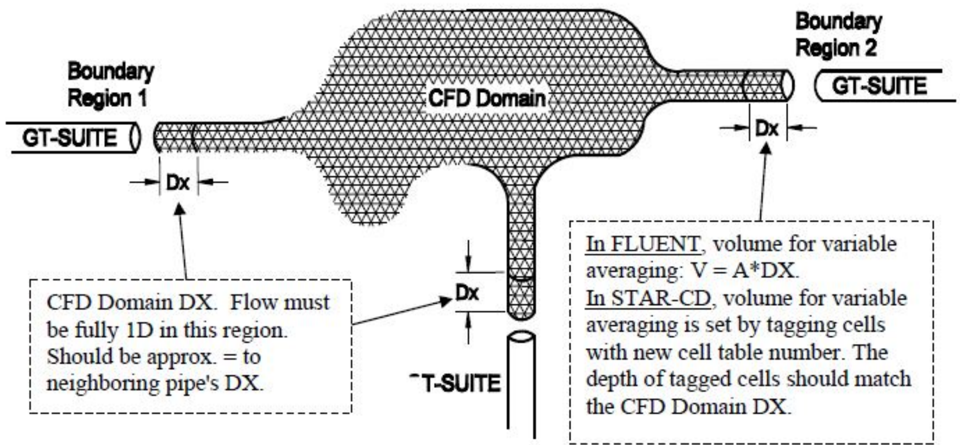


Figure 93: Relationship Between Flow Domains

Coupling both the 1D and 3D domains allows the designer to obtain critical information in 3D whilst modelling a complex system in 1D, such as the engine. Moreover, it eradicates the need to approximate boundary conditions and hence can yield very accurate results. GT Power is used to approximate key design parameters that affect performance. Runner and diffuser geometry are the two key characteristics that are varied to obtain performance characteristics such as IMEP, pressure distribution, BSFC, torque and power.

The method is first initialised in GT Power where majority of the work is done. An implicit discretisation scheme is used to solve the fluid equations. GT Power is run for a predetermined number of "pre-cycles', independent from fluent. These pre-cycles act as initial conditions which are fed into Fluent. Volume averaging of the CFD solution variables is required in order to allow both domains to interact. The length is determined by the targeted discretisation length attribute. The sub-volumes contain both a cell centre and face. A staggered grid is used to solve these solution variables, whereby vectors are solved on cell faces and scalars are solved at cell centres. Coupling occurs at the boundary interfaces of both codes. Flow entering and leaving the boundary regions is fully one dimensional.

C.11.1 Methodology

GT Power provides a tutorial on creating a socket connection between the two software packages. Firstly, the GT Power model needs to be adapted so that all the components that are in Fluent are replaced by a CFD Component. Figures 94 and 95 shows how the manifold is changed and ports are created to connect the throttle body to the diffuser and the plenum ports to the runners.

After the GT Power model is set up, Fluent must be linked to GT Power. A .dat file is created, and a definition is made to connect them. For successful coupling, all the species that are used in GT Power must be defined in Fluent. This involves making a mixture template where specific properties such as heat of evaporation, molecular weight and density must be defined. This allows fluent to utilise these properties if EGR is utilised in the system. However, given that only air from the atmosphere is used, these properties are included for completeness.

A major source of divergence in coupling is a result of an inadequate mesh. Initial runs of the simulation showed major divergence in the AMG temperature solver. To verify that the mesh is adequate, we can generate graphs of the orthogonal skewness within the mesh module, this is shown in figures 96, 98 and 101 Mesh statistics as shown in figures 97, 100 and 102 show the settings utilised to ensure the orthogonal skewness is kept as close to 1.0.

Orthogonal quality is an excellent way of measuring the quality of the mesh. It is calculated from the normalised dot product of the area vector of a face and a vector from the centroid of the cell to to the centroid of that face. Then it again calculates the normalised dot product of the area vector of a face but utilises the vector from the centroid of a cell to the centroid of the adjacent cell that shares that face (31).

The purpose of these simulations is to capture effects near the boundary; moreover the effect of flow separation. Accurate capturing of this phenomenon is crucial for the design of the diffuser. To include such effects in the simulation, inflation layers were used. Global and local inflation layers are used for the models. This is essential as local inflation layers were applied to the inlet, and connecting ports. As mentioned previously (see section C.11) that these connections need to be preferably 1D to get accurate results and allow successful coupling. The inflation layers create layers of meshed regions near boundaries and hence increase the resolution in those areas.

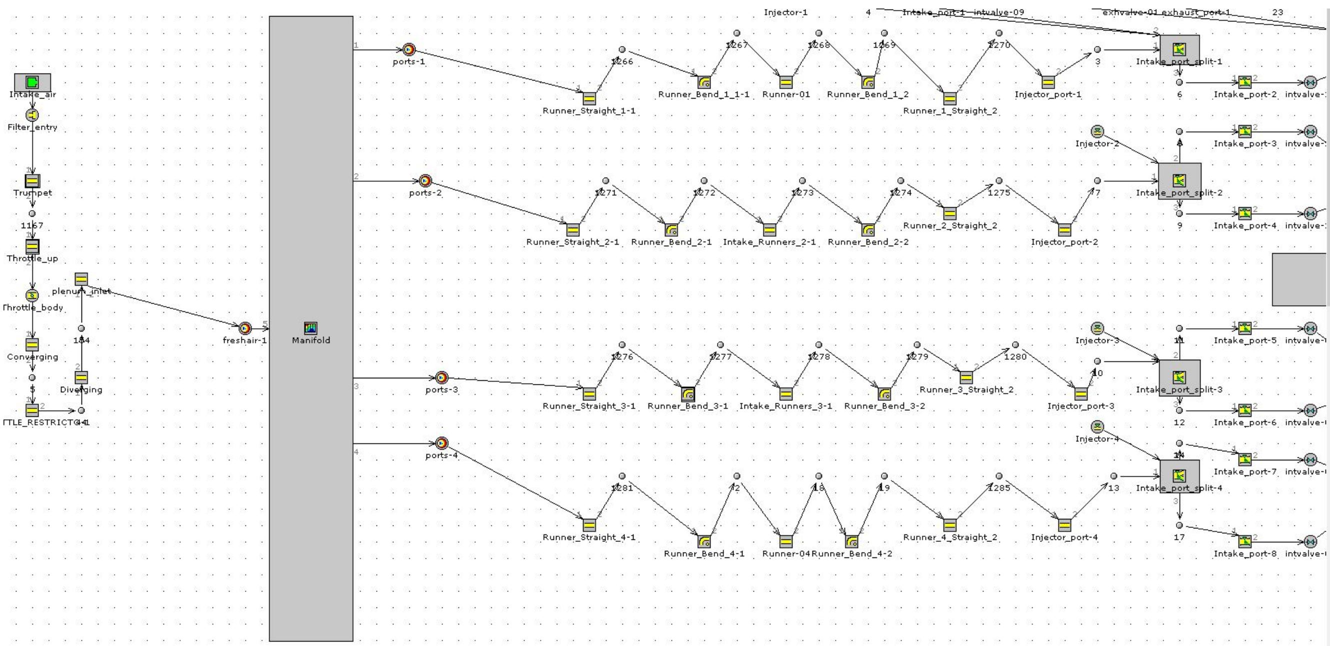


Figure 94: Altered GT Power Model

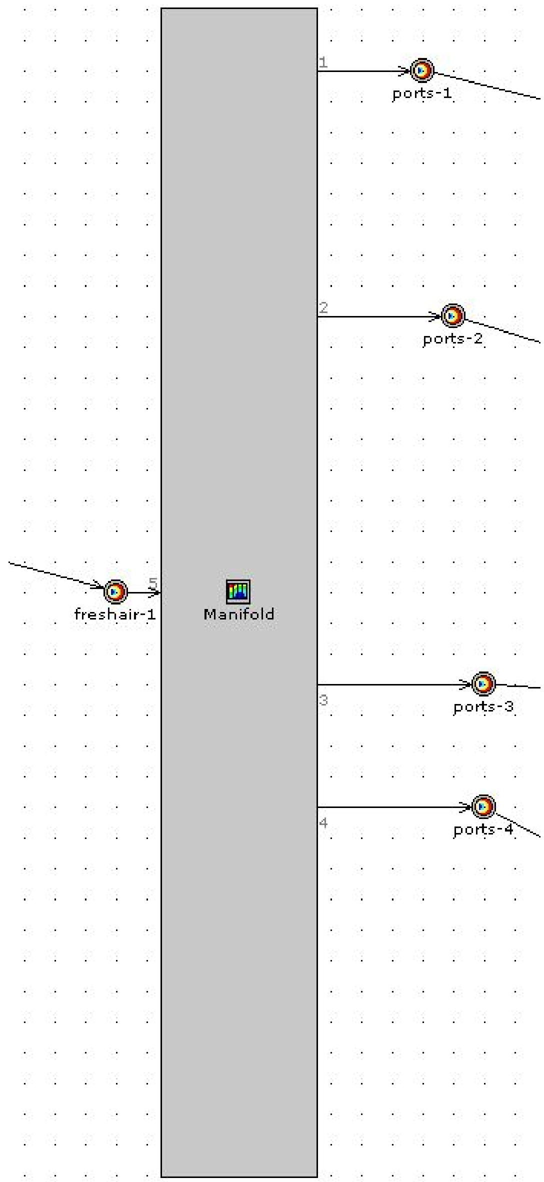


Figure 95: Close up of CFD Components

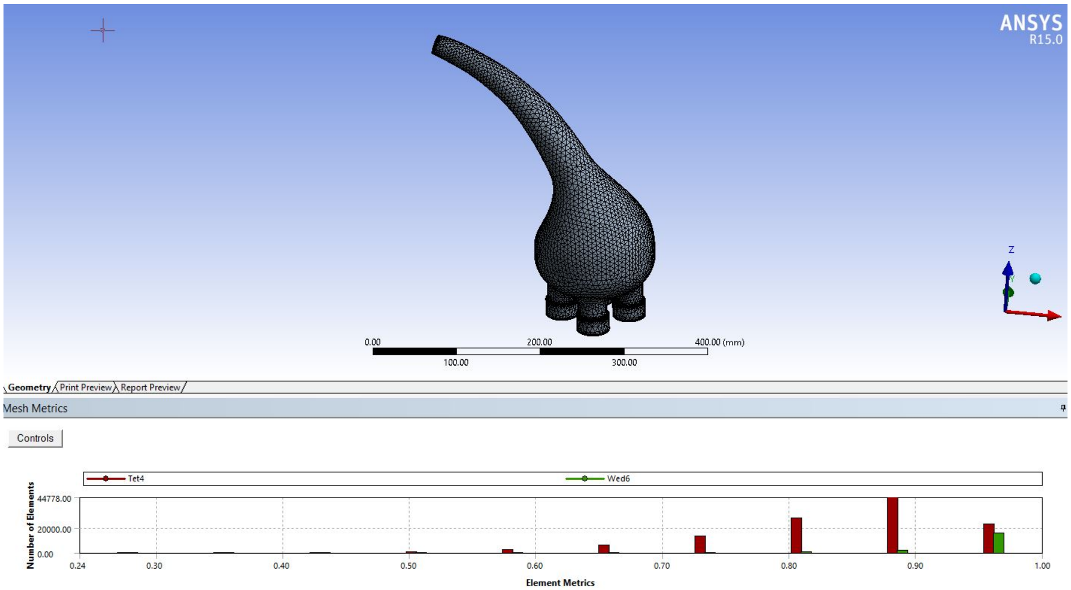


Figure 96: Mesh and Mesh Statistics Graph

Details of "Mesh"

Defaults	
Physics Preference	CFD
Solver Preference	Fluent
Relevance	0
Sizing	
Use Advanced Size Function	On: Proximity and Curvature
Relevance Center	Coarse
Initial Size Seed	Active Assembly
Smoothing	Medium
Transition	Slow
Span Angle Center	Fine
Curvature Normal Angle	Default (18.0 °)
Num Cells Across Gap	Default (3)
Min Size	1.e-002 mm
Proximity Min Size	Default (0.229440 mm)
Max Face Size	1.0 mm
Max Size	5.0 mm
Growth Rate	Default (1.20)
Minimum Edge Length	0.30270 mm
Inflation	
Assembly Meshing	
Patch Conforming Options	
Patch Independent Options	
Advanced	
Defeaturing	
Statistics	
Nodes	33847
Elements	135786
Mesh Metric	Orthogonal Quality
Min	0.239267388634496
Max	0.999999999999347
Average	0.86070482253348
Standard Deviation	9.85256071640994E-02

Figure 97: Mesh Settings and Mesh Statistics

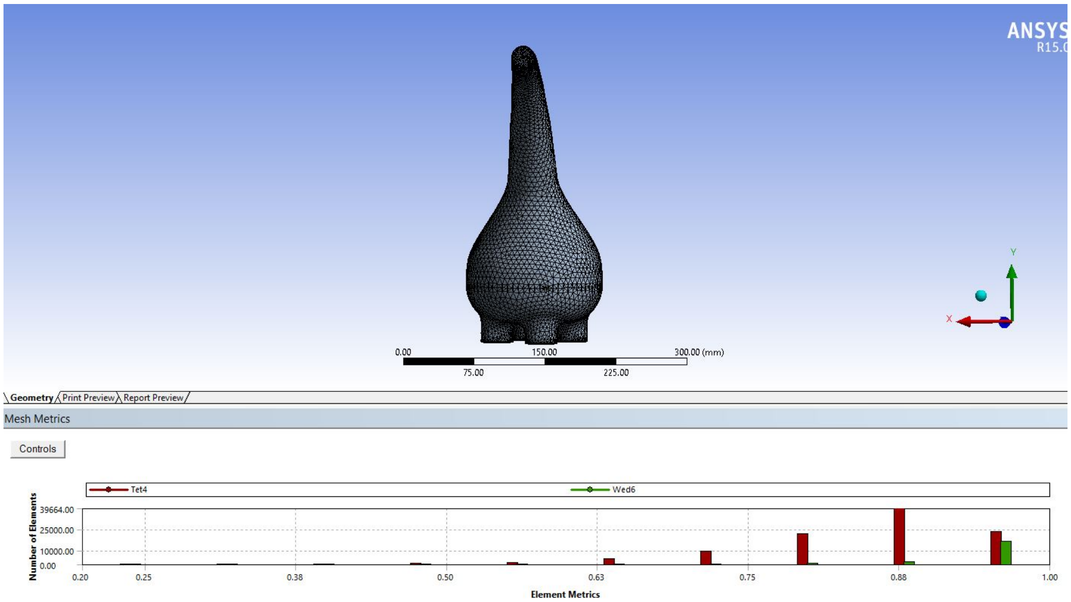


Figure 98: Mesh and Mesh Statistics Graph

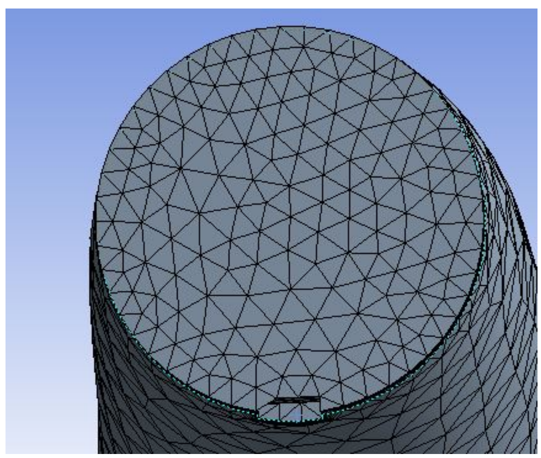
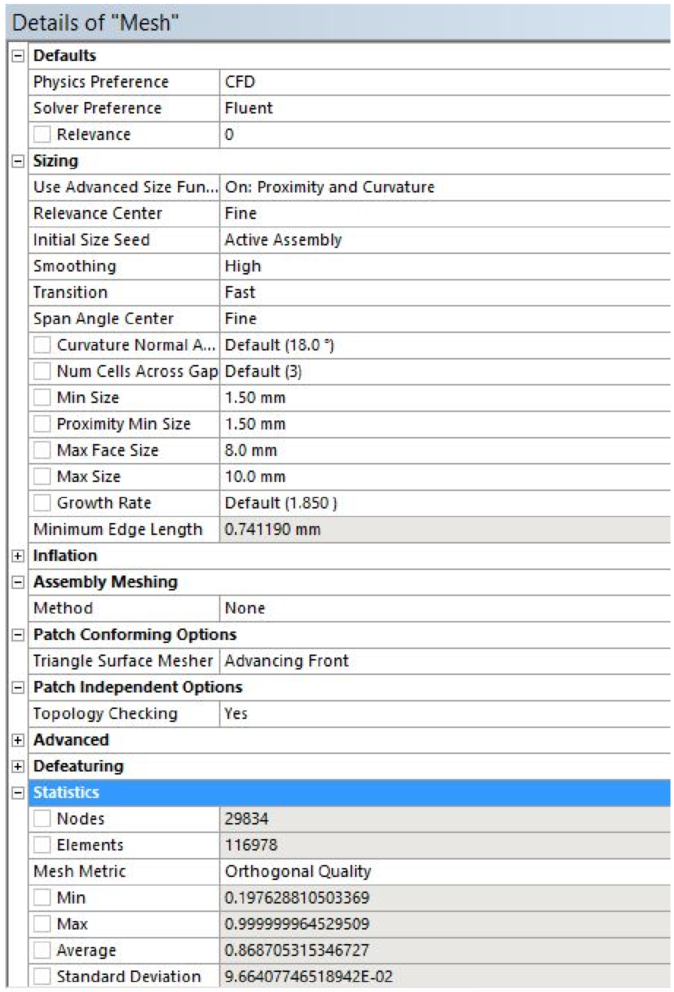


Figure 99: Close up of the Dall Slot Mesh region



Details of "Mesh"

Defaults	
Physics Preference	CFD
Solver Preference	Fluent
Relevance	0
Sizing	
Use Advanced Size Fun...	On: Proximity and Curvature
Relevance Center	Fine
Initial Size Seed	Active Assembly
Smoothing	High
Transition	Fast
Span Angle Center	Fine
Curvature Normal A...	Default (18.0 °)
Num Cells Across Gap	Default (3)
Min Size	1.50 mm
Proximity Min Size	1.50 mm
Max Face Size	8.0 mm
Max Size	10.0 mm
Growth Rate	Default (1.850)
Minimum Edge Length	0.741190 mm
Inflation	
Assembly Meshing	
Method	None
Patch Conforming Options	
Triangle Surface Mesher	Advancing Front
Patch Independent Options	
Topology Checking	Yes
Advanced	
Defeaturing	
Statistics	
Nodes	29834
Elements	116978
Mesh Metric	Orthogonal Quality
Min	0.197628810503369
Max	0.999999964529509
Average	0.868705315346727
Standard Deviation	9.66407746518942E-02

Figure 100: Mesh Settings and Mesh Statistics

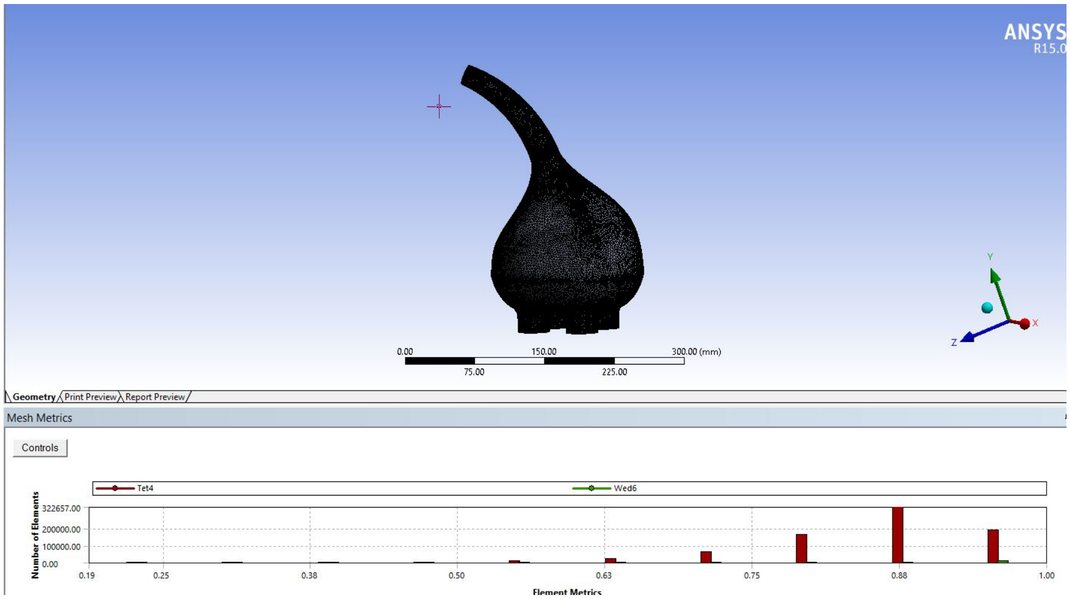


Figure 101: Mesh and Mesh Statistics Graph

Details of "Mesh"	
Defaults	
Physics Preference	CFD
Solver Preference	Fluent
Relevance	0
Sizing	
Use Advanced Size Function	On: Proximity and Curvature
Relevance Center	Coarse
Initial Size Seed	Active Assembly
Smoothing	Medium
Transition	Slow
Span Angle Center	Fine
Curvature Normal Angle	Default (18.0 °)
Num Cells Across Gap	Default (3)
Min Size	1.e-002 mm
Proximity Min Size	Default (0.199250 mm)
Max Face Size	1.0 mm
Max Size	5.0 mm
Growth Rate	Default (1.20)
Minimum Edge Length	0.30270 mm
Inflation	
Assembly Meshing	
Patch Conforming Options	
Patch Independent Options	
Advanced	
Defeaturing	
Statistics	
Nodes	148472
Elements	765972
Mesh Metric	Orthogonal Quality
Min	0.186822813188256
Max	0.99971259370163
Average	0.856851323084149
Standard Deviation	8.47963095236642E-02

Figure 102: Mesh Settings and Mesh Statistics

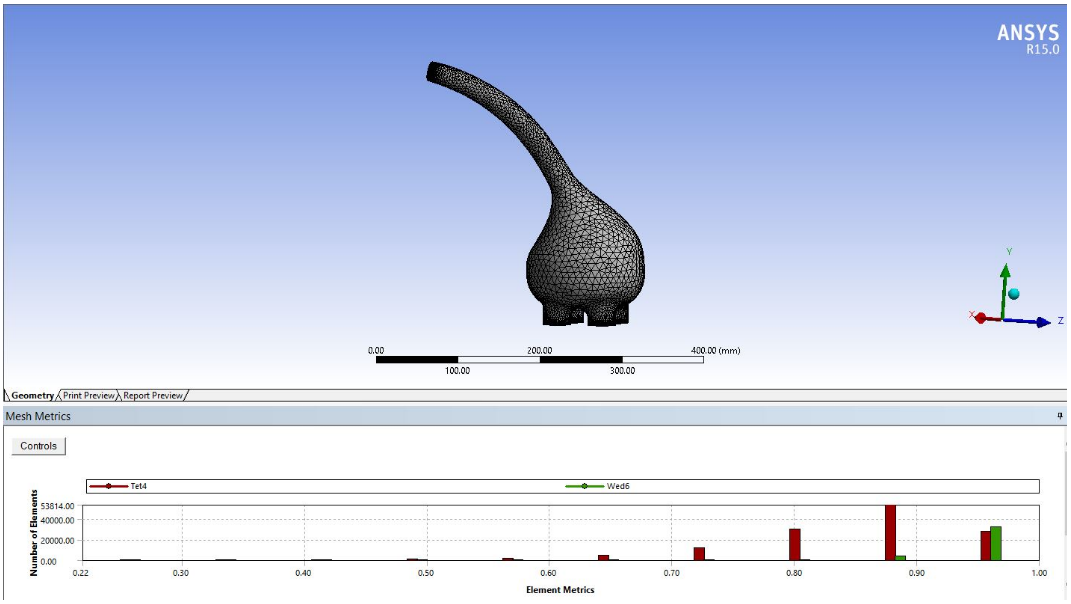
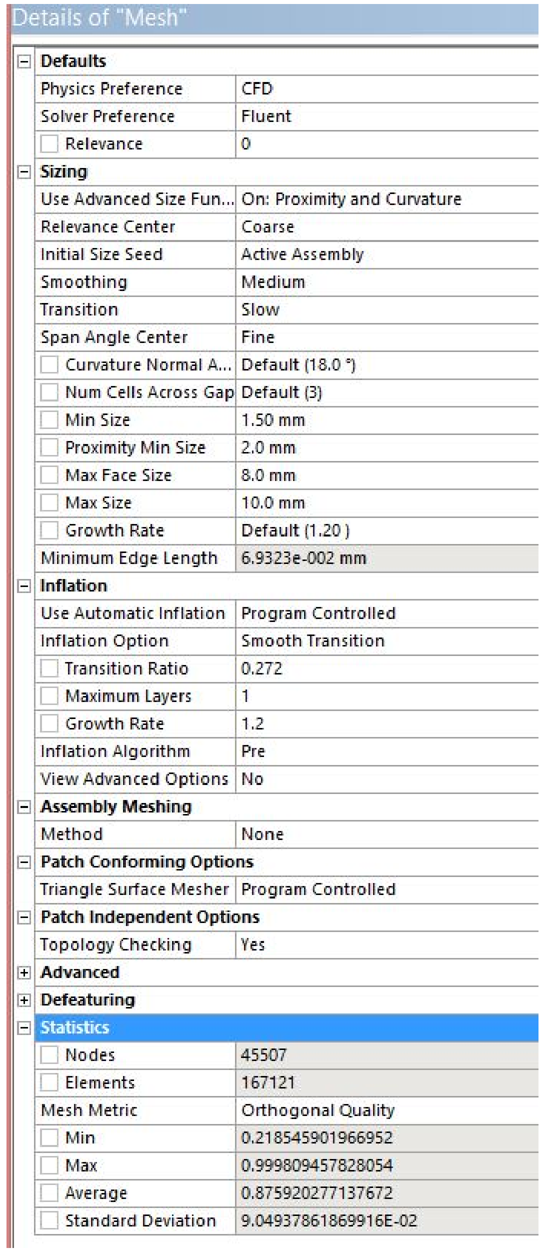


Figure 103: Mesh Settings and Mesh Statistics



Details of "Mesh"

Defaults	
Physics Preference	CFD
Solver Preference	Fluent
Relevance	0
Sizing	
Use Advanced Size Fun...	On: Proximity and Curvature
Relevance Center	Coarse
Initial Size Seed	Active Assembly
Smoothing	Medium
Transition	Slow
Span Angle Center	Fine
Curvature Normal A...	Default (18.0 °)
Num Cells Across Gap	Default (3)
Min Size	1.50 mm
Proximity Min Size	2.0 mm
Max Face Size	8.0 mm
Max Size	10.0 mm
Growth Rate	Default (1.20)
Minimum Edge Length	6.9323e-002 mm
Inflation	
Use Automatic Inflation	Program Controlled
Inflation Option	Smooth Transition
Transition Ratio	0.272
Maximum Layers	1
Growth Rate	1.2
Inflation Algorithm	Pre
View Advanced Options	No
Assembly Meshing	
Method	None
Patch Conforming Options	
Triangle Surface Mesher	Program Controlled
Patch Independent Options	
Topology Checking	Yes
Advanced	
Defeaturing	
Statistics	
Nodes	45507
Elements	167121
Mesh Metric	Orthogonal Quality
Min	0.218545901966952
Max	0.999809457828054
Average	0.875920277137672
Standard Deviation	9.04937861869916E-02

Figure 104: Mesh Settings and Mesh Statistics

C.12 Rapid Prototyping/ 3D Printing

Rapid prototyping of 3D Printing involves extruding a heated material through a nozzle building layers to eventually construct a part. MUR2014 has continued to utilise the 3D printers located in the Engineering Workshop. With after hours granted, this allowed MUR2014 to practice their skills to print the required components. The 3D printers in the Engineering Workshop use a process called Fused Deposition Modelling (FDM), in which a plastic filament is melted and pushed through a small extruder nozzle onto a build plate. The nozzle moves relative to the plate depositing lines of solidified plastic. The process then deposits layer on layer until the part is fully built. The layer size is typically adjustable between 0.15 and 0.4 mm.

The 3D printing facilities at the university has allowed a means to easily manufacture complex geometric components. These parts a light in weight and allows MUR2014 to print many iterations and test them on a dyno. 3D printing can either be done in ABS (acrylonitrile butadiene styrene) or PLA (polylactic acid). A Reel of Nylon has also been supplied by the Engineering workshop for MUR2014 to test. PLA is a very brittle material, so ABS becomes the only usable material. The Properties of ABS are shown in table 24

The key aspect of the intake system design in 2014 has been utilising CFD to reduce pressure loss. ABS has a rough and porous surface finish. This produces two problems, the first is an increase in surface friction and it may cause excessive backfire through increased porosity. The friction leads creating a thicker boundary layer which creates turbulence and hence causes flow separation. To solve this issue acetone was used to smooth the surface (see section C.12.1)

Density (kg/m$\hat{3}$)	1040
Yield Strength (Mpa)	44
Elastic Modulus (Gpa)	2.1
Glass Transition Temperature (^{o}C)	105

Table 24: Mechanical Properties of ABS

Majority of the components have been printed in house, however with a malfunctioning printer, the larger printer has now been decommissioned. We are left with only the smaller printers. Two new sponsors have been procured to help with manufacturing. Rapid Dimensions have already printed out the top half of the plenum whilst Advanced Manufacturing Services have offered to print the plenum in Nylon. The material properties of Nylon are shown in table 25. These properties allow thinner walls to be used as well as being more structurally rigid. The main concern is a rough surface finish. It can be coated in a resin which will be investigated as an alternative means of manufacturing.

3D Data	**Value**	**Units**
Tensile Modulus (X Direction)	1650	Mpa
Tensile Modulus (Y Direction)	1650	Mpa
Tensile Modulus (Z Direction)	1650	Mpa
Tensile Strength (X Direction)	48	Mpa
Tensile Strength (Y Direction)	48	Mpa
Tensile Strength (Z Direction)	42	Mpa
Strain at break (X Direction)	18	%
Strain at break (Y Direction)	18	%
Strain at break (Z Direction)	4	%
Charpy impact strength (+23°C, X Direction)	53	kJ/m$\hat{2}$
Charpy notched impact strength (+23°C, X Direction)	4.8	kJ/m$\hat{2}$
Flexural Modulus (23°C, X Direction)	1500	MPa

Table 25: Mechanical Properties of Nylon (PA 2200 Balance 1.0)

C.12.1 Acetone Vapour Bath

In order to smooth out the surface of the intake, acetone vapour was used. Using many online resources and videos an apparatus was set up. The items required were a variable electronic heating plate, an aluminium bucket and lid and thread. A 1 cm layer of acetone was poured into the bucket and then heated until the vapour began to boil. A key feature of acetone is that it is heavier than air which means the right temperature needs to be reached to allow the vapour to sit in the bucket to a height of about 2-5 cm below the top of the bucket. The vapour height can be seen by looking and the discrepancies in colour of the aluminium bucket.

After many attempts it was noted that tying string through bolt holes to suspend the part at the right height was most effective. The string did slightly damage the bolt surface but this result was much better than the surface finish created when placed flat on various materials. Once in, it was left for one minute to achieve a good surface finish. If left in for too long, the part does severely degrade dimensions. Some components do have tight tolerances so

After acetoning, the part had to be left to dry for about 30 minutes to allow it to solidify. If touched during this drying phase, finger print marks are left on the surface. Examining the component, shows a reduction in the porosity of the material and an increase in rigidity was achieved.

It is worthwhile noting that a small fillet was produced on any sharp corners. Drip points may also develop on any unsupported surfaces which may introduce bumps and more turbulent effects.

C.13 CAD

1^{st} Design Iteration

Figure 105: Initial Design Concept: increase flow distribution

2^{nd} Design Iteration

Figure 106: Selected Design: Iteration 1

3^{rd} Design Iteration

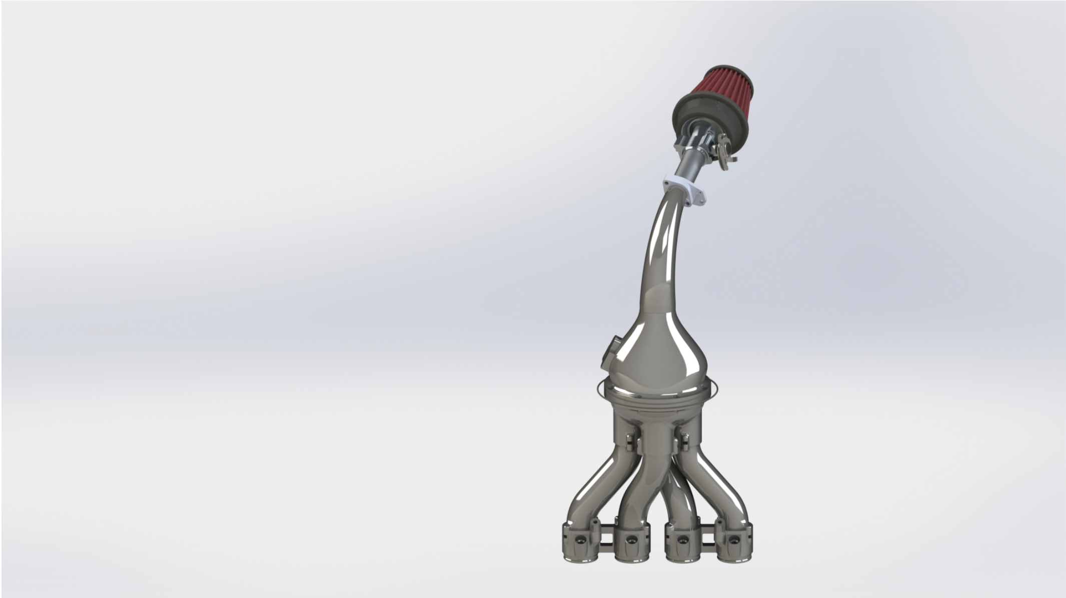


Figure 107: Selected Design: Iteration 2

4^{th} Design Iteration

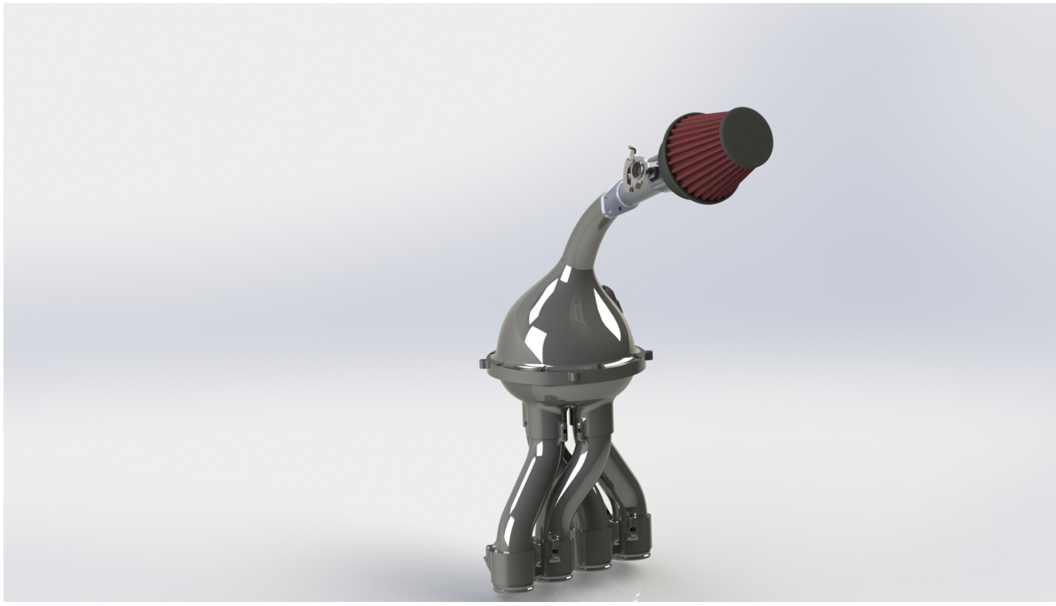
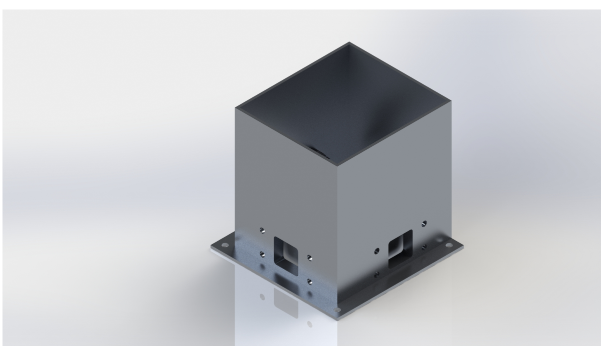
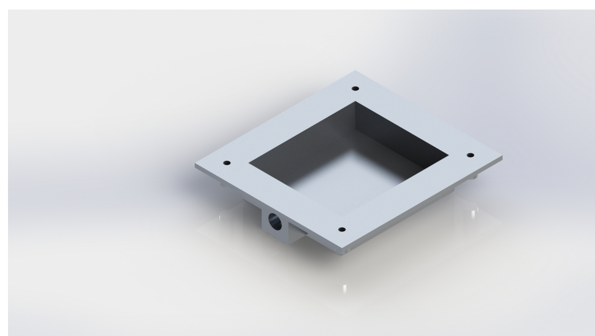


Figure 108: Selected Design: Iteration 3

D Fuel



(a) Surge Tank Render



(b) Fuel Pickup Render

D.1 Packaging Restrictions

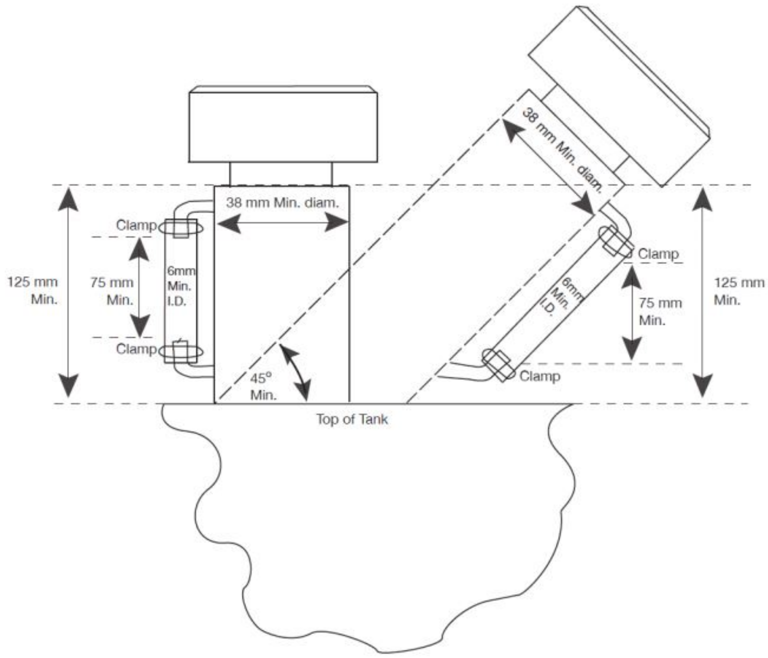


Figure 110: Fuel Filler Neck Restriction

D.2 Fuel Calculations

The equations used to calculate the pulse width, fuel flow rate and air flow rate are given below. Equation D.1 calculates the mass flow rate of air delivered into one cylinder. It assumes lambda of 0.85 and loss coefficient of 0.7. The density is also assumed to be constant at 1.181 $\frac{kg}{m^3}$. Equation D.2 calculates the mass flow rate of fuel into the cylinders. From this we can calculate the mass of fuel required which can be determined from equation D.3. Finally the time of injection/pulse width is given by equation D.4

$$\dot{m}_a(i) = \frac{(\rho_a V_{d1} \frac{N(i)}{60} n_v)}{n} \tag{D.1}$$

$$\dot{m}_f(i) = \frac{\dot{m}_a(i)}{AF} \tag{D.2}$$

$$\dot{m}_f 1(i) = \frac{\dot{m}_f(i) n}{\frac{N(i)}{60}} \tag{D.3}$$

$$t(i) = \frac{m_{f1}(i)}{\dot{m}_f(i)}; \tag{D.4}$$

```
clear all
close all
clc

V_d1=(0.6/1000)/4; % swept volume of each cyilnder m^3 for one cyilinder
V_d=(0.6/1000); % swept volume of each cyilnder m^3 for whole system
rho_a=1.181;
N=1000:100:13500;
n_v=0.95;
n=4;
```

```
lambda=0.85;
phi=1/lambda;
AF_st=9.85;
AF_actual=AF_st*lambda;
%AF=AF_st*phi;
AF=AF_st/phi;

C_D=0.7;
An=(pi*(7.95e-3)^2)/4;
rho_f=783.7; %kg.m^-3
P_rail=350000; %Pa
MAP=95000; %Pa

m_dot_a=zeros(1,length(N));
m_dot_f=zeros(1,length(N));
m_f1=zeros(1,length(N));
t=zeros(1,length(N));
omega=zeros(1,length(N));
td=zeros(1,length(N));

tic

for i=1:length(N)
    %air flow rate into one cyilinder at WOT
    m_dot_a(i)= (rho_a * V_d1  * (N(i)/60) *n_v)/n;

    %fuel flow rate into one cylinder .: the flow rate through the injector
    m_dot_f(i)=m_dot_a(i)/AF;

    %mass of fuel into one cylinder for one cycle
    m_f1(i)=(m_dot_f(i)*n)/(N(i)/60);     %kg

    %time of injection
    t(i)=m_f1(i)/(m_dot_f(i)) ;%(s)

    %rotational speed
    omega(i)=(N(i)/60)*360; %(degrees/sec)

    %time of injection
    td(i)=omega(i)*t(i); %(degrees)


end

figure(1)
plot(N,t)
xlabel('RPM')
ylabel('Pulse Width (s)')
title('RPM vs Pulse Width (Single Cylinder)')

figure(2)
plot(N,m_dot_f)
xlabel('RPM')
ylabel('Fuel Flow rate kg/s')
title('RPM vs Fuel Flow rate (Single Cylinder)')
%%
%Fuel Injector Actual

delta_P=0:255000/125:255000; %delta_p=(P_rail-MAP)

m_dot_ff=zeros(1,length(N));

for i=1:length(N)

    m_dot_ff(i)=C_D*An*sqrt(2*rho_f*(delta_P(i)));

end
```

```
figure(3)
plot(m_dot_ff,delta_P)
xlabel('Mass Flow rate (kg/s)')
ylabel('Pressure Difference (Pa)')
title('Mass Flow rate vs Pressure Change (Single Cylinder) Fuel Injector')

%% Fuel for whole system
m_dot_a_s=zeros(1,length(N));
m_dot_f_s=zeros(1,length(N));
m_f1_s=zeros(1,length(N));
t_s=zeros(1,length(N));
omega_s=zeros(1,length(N));
td_s=zeros(1,length(N));

for i=1:length(N)
    %air flow rate into  whole system at WOT
    m_dot_a_s(i)= (rho_a * V_d * (N(i)/60) *n_v) /n;

    %fuel flow rate  whole system .: the flow rate through the injector
    m_dot_f_s(i)=m_dot_a_s(i)/AF;

    %Mass of fuel in whole system for one cycle
    m_f1_s(i)=(m_dot_f_s(i)*n)/(N(i)/60);     %kg

    %time of injection
    t_s(i)=m_f1_s(i)/(m_dot_f_s(i)) ;%(s)

    %rotational speed  whole system
    omega_s(i)=(N(i)/60)*360; %(degrees/sec)


    %time of injection  whole system
    td_s(i)=omega_s(i)*t_s(i); %(degrees)

end
figure(4)
plot(N,t_s)
xlabel('RPM')
ylabel('Pulse Width (s)')
title('RPM vs Pulse Width (Whole System)')

figure(5)
plot(N,m_dot_f_s)
xlabel('RPM')
ylabel('Fuel Flow rate kg/s')
title('RPM vs Fuel Flow rate (Whole System)')
%%
E85_flow=zeros(1,length(N));
m_dot_ffL=zeros(1,length(N));


for i=1:length(N)

   E85_flow(i)=(m_dot_f_s(i)/rho_f)*4869.2797458;
   m_dot_ffL(i)=(m_dot_ff(i)/rho_f)*4869.2797458;

end

figure(6)
plot(N,m_dot_f_s*4869.2797458)
xlabel('RPM')
ylabel('Fuel Flow rate L/hr')
title('RPM vs Fuel Flow rate L/hr (Whole System)')

figure(7)
plot(N,m_dot_ffL)
xlabel('RPM')
ylabel('Mass Flow Rate Air L/hr')
```

```
title('RPM vs Fuel Flow rate L/hr (Whole System)')

toc


%% SI Injection Deleivery Rate

clear all
close all
clc

VE=1; % Volumetric Efficency (%)
rho=1.16; %Reference densitgy (kg/m^3)
RPM=3000:500:13500; %Engine Speed
Vd= 0.6; %Engine Displacement (L)
FAR=1/9.89; % Fuel air ratio
n=4; % number of cylinders
PW= 245; % Injection Duration
rho_e85=.78365;


mdot_delivery=((VE*rho*RPM*Vd*FAR)*(6/(n*PW))*60*60)/(1000*rho_e85) %(L/hr)
```

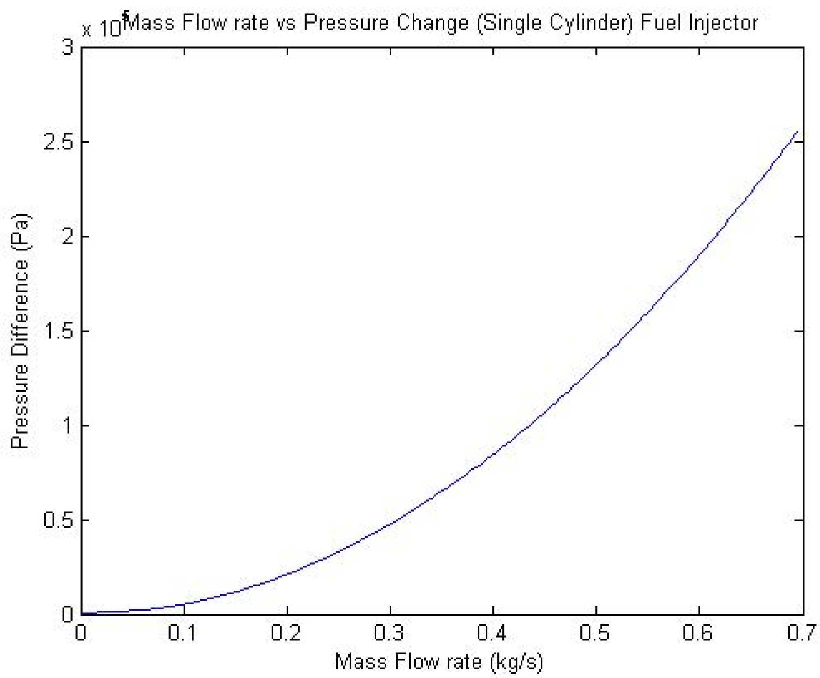


Figure 111: Mass Flow Rate vs Single Cylinder Pressure Change

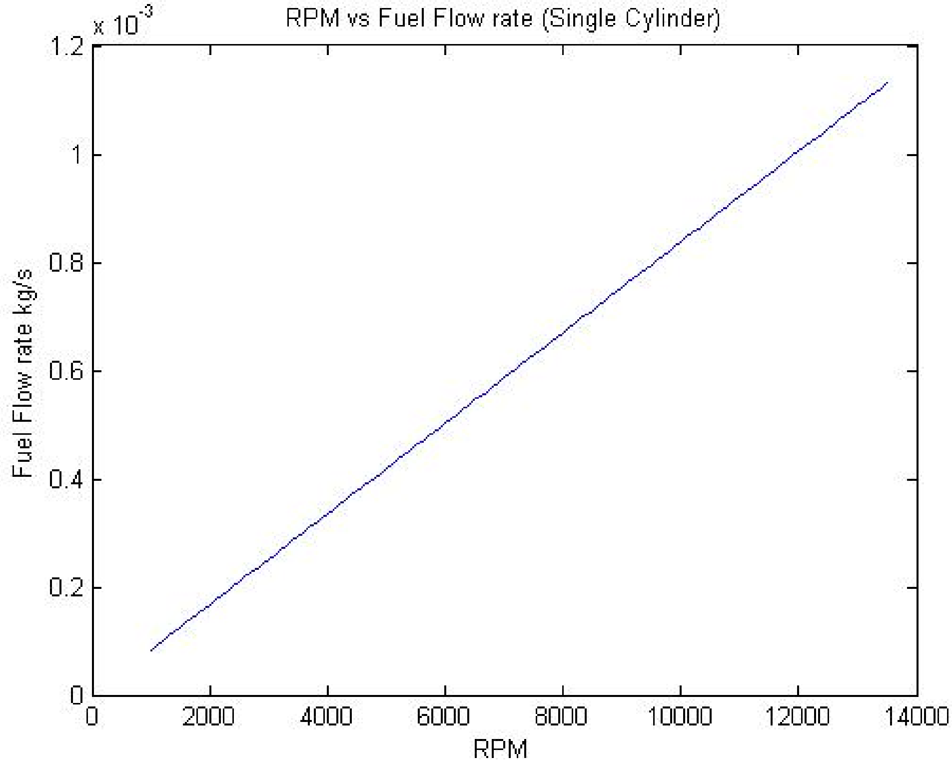


Figure 112: RPM vs Fuel Flow Rate (Single Cylinder)

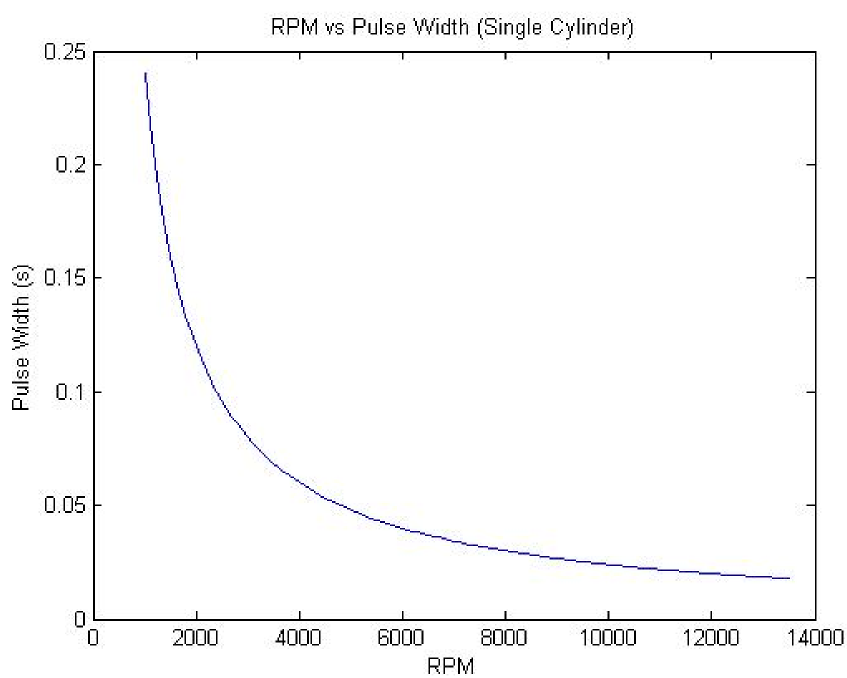


Figure 113: RPM vs Pulse Width Single Cylinder

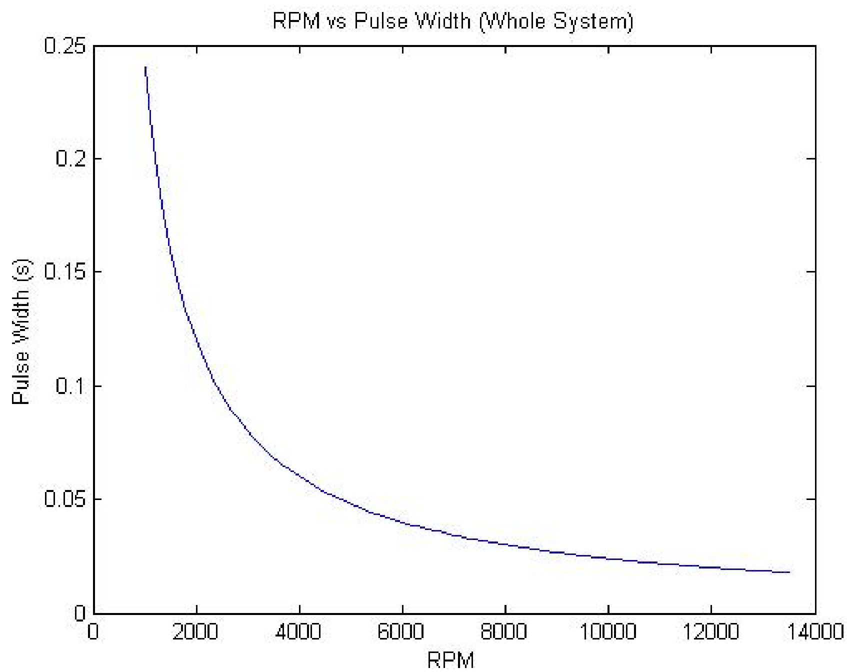


Figure 114: RPM vs Pulse Width (Whole System)

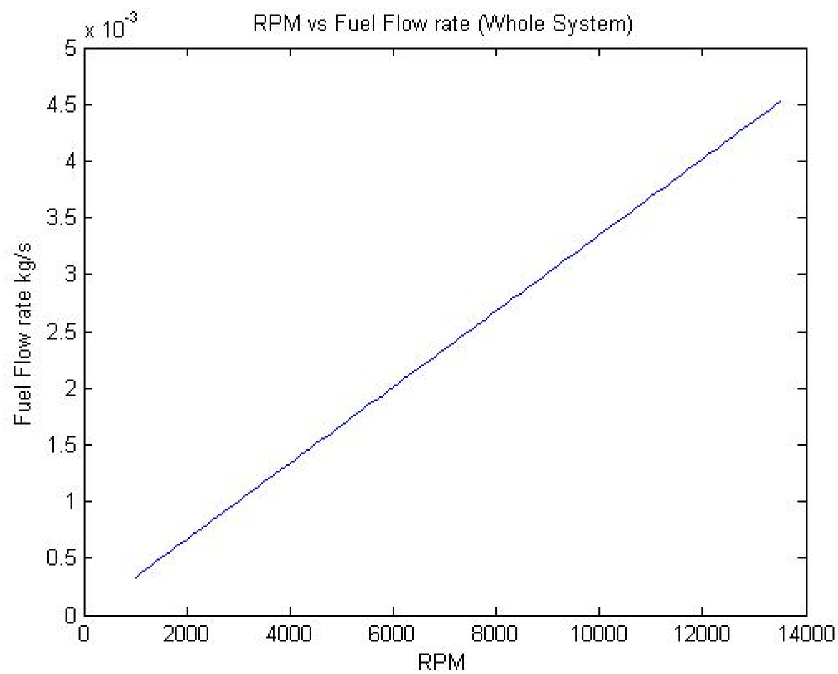


Figure 115: RPM vs Fuel Flow Rate (Whole System)

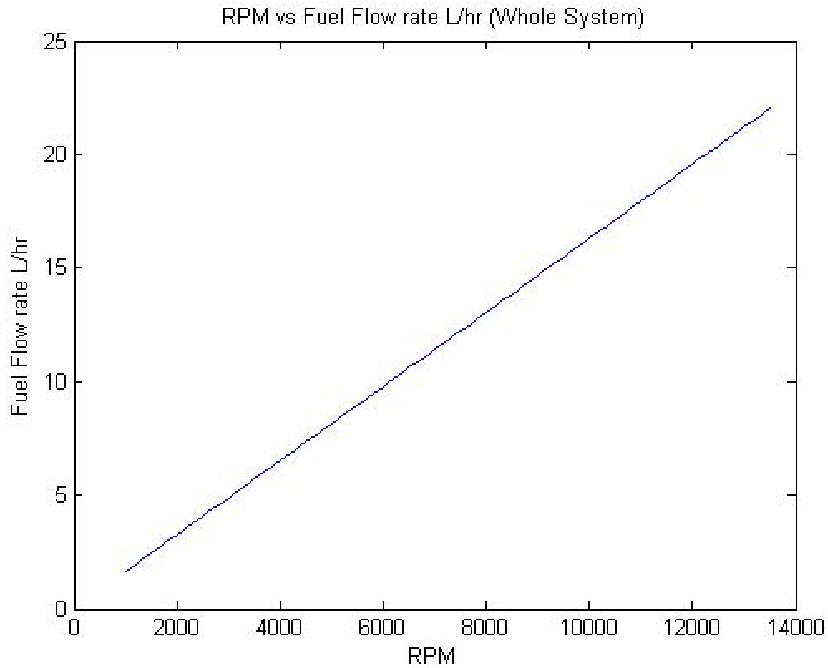


Figure 116: RPM vs Fuel Flow Rate (Whole System)

RPM	**Flow Rate L/hr**
3000	6.947452753
3500	8.105361546
4000	9.263270338
4500	10.42117913
5000	11.57908792
5500	12.73699671
6000	13.89490551
6500	15.0528143
7000	16.21072309
7500	17.36863188
8000	18.52654068
8500	19.68444947
9000	20.84235826
9500	22.00026705
10000	23.15817584
10500	24.31608464
11000	25.47399343
11500	26.63190222
12000	27.78981101
12500	28.94771981
13000	30.1056286
13500	31.26353739

Table 26: SI Flow Rate Calculations

D.3 Fuel Line Velocity

Maximum volume flow (L/h)	100
Maximum volume flow (m3̂/s)	2.78E-05
6AN ID (m)	7.95E-03
6AN Area (m2̂)	4.96E-05
6AN velocity (m/s)	0.5596

Table 27: Velocity through Speedflow lines

```
clear all
close all
clc

max_vol_flow=100; %L/hr
max_vol_flow1=max_vol_flow/(1000*3600); %m^3/s

sixAN_ID= 0.00795; %m
sixAN_area=pi*(sixAN_ID/2)^2; %m^2

sixAN_vel=max_vol_flow1/sixAN_area;
```

D.4 Power Calculations

```
clear all
close all
clc

rho_gas=0.7437;
rho_e85=0.7907;

%AFR for Gasoline at Lambda =1 (complete stoichiometric combustion)
AFR_gas=14.64;

O2=31.9988; %g/mol O2 weight

fprintf(' C2H6O + 3 O2 = 2 CO2 + 3 H2O \n ') % 100% ethanol reaction

ethanol_weight=46.06844; %g/mol 100% ethanol C2H6O weight

O2_mol=3*O2; %g/mol

ethanal_combust=O2_mol/ethanol_weight; %oxygen/Ethanol for complete combustion, (lambda=1)

O2_MM_Air=28.97; %Oxygen Molecular Mass in Air
O2_in_air=6.704;

%Oxygen contents in air
O2_percentage=(O2_in_air /O2_MM_Air)*100

% theoretical AFR for 100% Ethanol at lambda=1
AFR_ethanol=ethanal_combust/(O2_percentage/100)

 %The theoretical AFR for mixture E85 at lambda=1
AFR_e85=(0.85*AFR_ethanol) + (0.15*AFR_gas)

v=(AFR_gas*rho_gas)/(AFR_e85*rho_e85)

extra_volume_required=(v-1)*100
```

```
combustion_energy_gas=44.53; %kJ/kg
combustion_energy_ethanol=29.76; %kJ/kg
combustion_energy_e85= 0.85*combustion_energy_ethanol +0.15 *combustion_energy_gas %kJ/kg

combustion_energy_diff= ((combustion_energy_gas-combustion_energy_e85)...
    /combustion_energy_gas)*100

Power_Increase=extra_volume_required - combustion_energy_diff
```

Variable	Value	Units
rho_e85	0.7907	kg/m$\hat{3}$
rho_gas	0.7437	kg/m$\hat{3}$
AFR E85	9.849937	-
AFR 100% Ethanol	9.004631	-
AFR RON 98	14.64	-
O2	31.9988	g/mol
Molar Mass of O2 in Air	28.97	g/mol
O2 in air	6.704	g/mol
O2 required for 100% Combustion	95.9964	g/mol
O2 percentage in air	23.14118	%
Ethanol combustion ratio	2.083778	-
100% Ethanol molecular weight	46.06844	g/mol
Volume Ratio	1.397957	-
extra_volume_required	39.79566	%
Combustion energy of E85	31.9755	kJ/kg
Combustion energy of ethanol	29.76	kJ/kg
Combustion_energy of RON	44.53	kJ/kg
Combustion energy difference	28.19335	%
Theoretical Power Increase	11.60231	%

Table 28: Values used and obtained from Power Calculations

D.5 Fuel Injection System

In certain critical areas the specification tolerances of a GDI injector design are more rigorous than that of the port-fuel injector. These areas are:

- Significantly enhanced atomisation level; a smaller value of spray mean drop size
- expanded dynamic range
- combustion sealing capability
- avoidance of needle bounce that creates unwanted secondary injections
- reduced bandwidth tolerance for static flow and flow linearity specifications
- more emphasis on spray penetration control
- more emphasis on the control of the sac volume spray
- enhanced resistance to deposit formation
- smaller flow variability under larger thermal gradients;
- ability to operate at higher injector body and tip temperatures

- leakage resistance at elevated fuel and cylinder pressures
- zero leakage at cold temperature
- more emphasis on packaging constraints
- flexibility in producing off-axis sprays in various inclined axes to meet different combustion system requirements

Zhao (5) and others outlined the basis of fuel injection and its importance for combustion. The primary fuel spray characteristics of a port fuel injector is not as influential as a GDI system. This is mainly due to the integrating fuel effects of the residence time on the closed valve, and due to the secondary atomisation that occurs as the induction air flows through the valve opening.

Fuel atomisation is found to ensure a robust system where combustion is predictable. The swirl component plays an important role in how the spray characteristics develop. The spray shapes at the transition between cone growth and torus formation are quite different with and without fuel swirl. The cone angle achieved with swirl is larger than without, whilst the penetration of the fuel are almost identical.

The swirl type injector does increase the surface roughness of the orifice wall and hence tends to exacerbate the formation of streams of fuel in the fuel sheet exiting the nozzle. This type of phenomena defeats the purpose of changing injectors. It will result in the formation of pockets of locally rich air-fuel mixture which in turn will make combustion unpredictable. To combat this inhomogeneity the swirl channel surface finish and nozzle tip must be of high quality(5).

D.6 Bosh 69608 Fuel Pump Specifications

0 580 ...	... 254 053	... 254 909	... 254 910	... 254 911	... 464 069	... 464 070	... 464 085
1)	175...228	148...198	130...200	95...135	98...148	130...195	95...145
2)	500	500	500	400	400	300	250
3)	13	13	13	12	12	12	12
4)	< 11,4	< 8,7	< 9,8	< 5,2	< 7,0	< 6,5	< 4,5
5)	800...1450	800...1450	800...1450	650...850	480...850	450...800	480...850
6)	180	180	203	199	186	175	175
7)	60	60	60	52	52	52	52
8)	12	12	15	15	12	12	12
9)	M 12 x 1,5	M 12 x 1,5	M 12 x 1,5	M 12 x 1,5	M 12 x 1,5	–	–
10)	–	–	–	–	–	8	8
11)	–	–	–	M 10 x 1	M 14 x 1,5	–	–

1) Delivery quantity with nominal voltage l/h
2) System pressure kPa
3) Nominal voltage V
4) Power consumption A
5) Pressure limit kPa
6) Total length (l) mm
7) Pump Ø (a) mm
8) Suction connection, nominal Ø (b) mm
9) Pressure connection thread (c)
10) Pressure connection, nominal Ø (c) mm
11) Additional tube fittings

Figure 117: Bosh Fuel Pump Specifications

D.7 Bosch 280 500 740/2 Fuel Regulator Specifications

ADJUSTABLE REGULATORS

Part Number	Set Pressure [bar]	Inlet Connection	Outlet Connection	Return Flow Max [l/h]	Type	Figure	Comment
B 280 500 701	1.9 - 5.0	10 mm "O" ring	8 mm Barb	115	DR 4	A	
B 280 500 737	1.9 - 5.0	M14 x 1.5	M14 x 1.5	115	DR 4	B	
B 280 500 740	3.0 - 4.0	M12 x 1.5	8 mm Barb	100	DR 4	B	
B 280 500 741	2.0 - 5.0	2 x M14 x 1.5	8 mm Barb	115	DR 4	C	
B 280 500 743	1.5 - 5.0	M14 x 1.5	8 mm Barb	115	DR 4	B	
B 280 500 799	1.9 - 4.0	10 mm "O" ring	M14 x 1.5	115	DR 4	A	
B 280 550 340	2.2 - 3.5	NA	NA	105	DR 2	D	
B 280 550 341	3.5 - 5.0	NA	NA	105	DR 2	D	

Figure 118: Bosh Fuel Regulator Specifications

E Exhaust

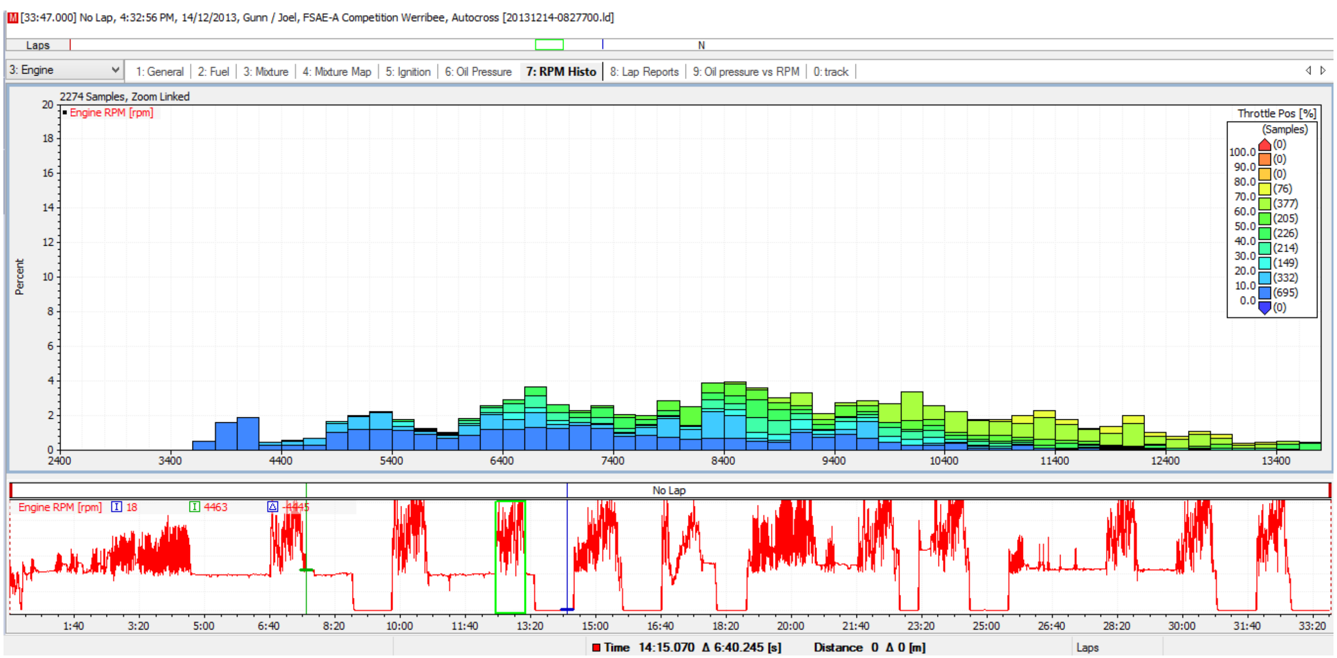


Figure 119: RPM Data from Auto Cross Event 2013

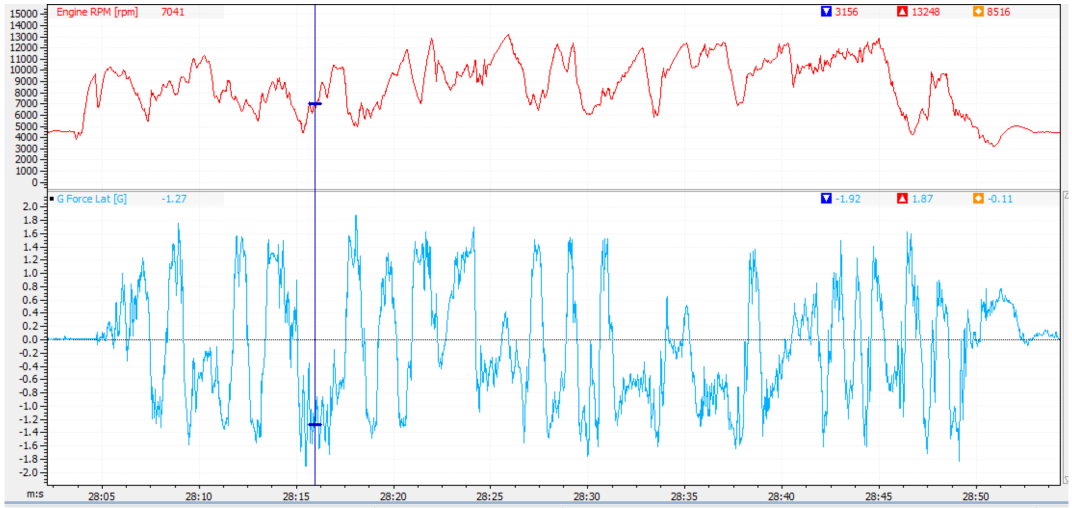


Figure 120: Lat G Force RPM. Used to estimate cornering RPM for 2014 vehicle

RPM RANGE	% TIME
5.4-8.4'000	39.94%
6-9'000	45.75%
6.4-9.4'000	46.20%
7-10'000	45.87%
7.4-10.4'000	45.83%
8-11'000	44.41%
8.4-11.4'000	42.07%
9-12'000	36.09%
9.4-12.4'000	32.51%
10-13'000	26.65%
10.4-13.4'000	21.74%

Figure 121: RPM Ranges Corresponding % of Time in those ranges

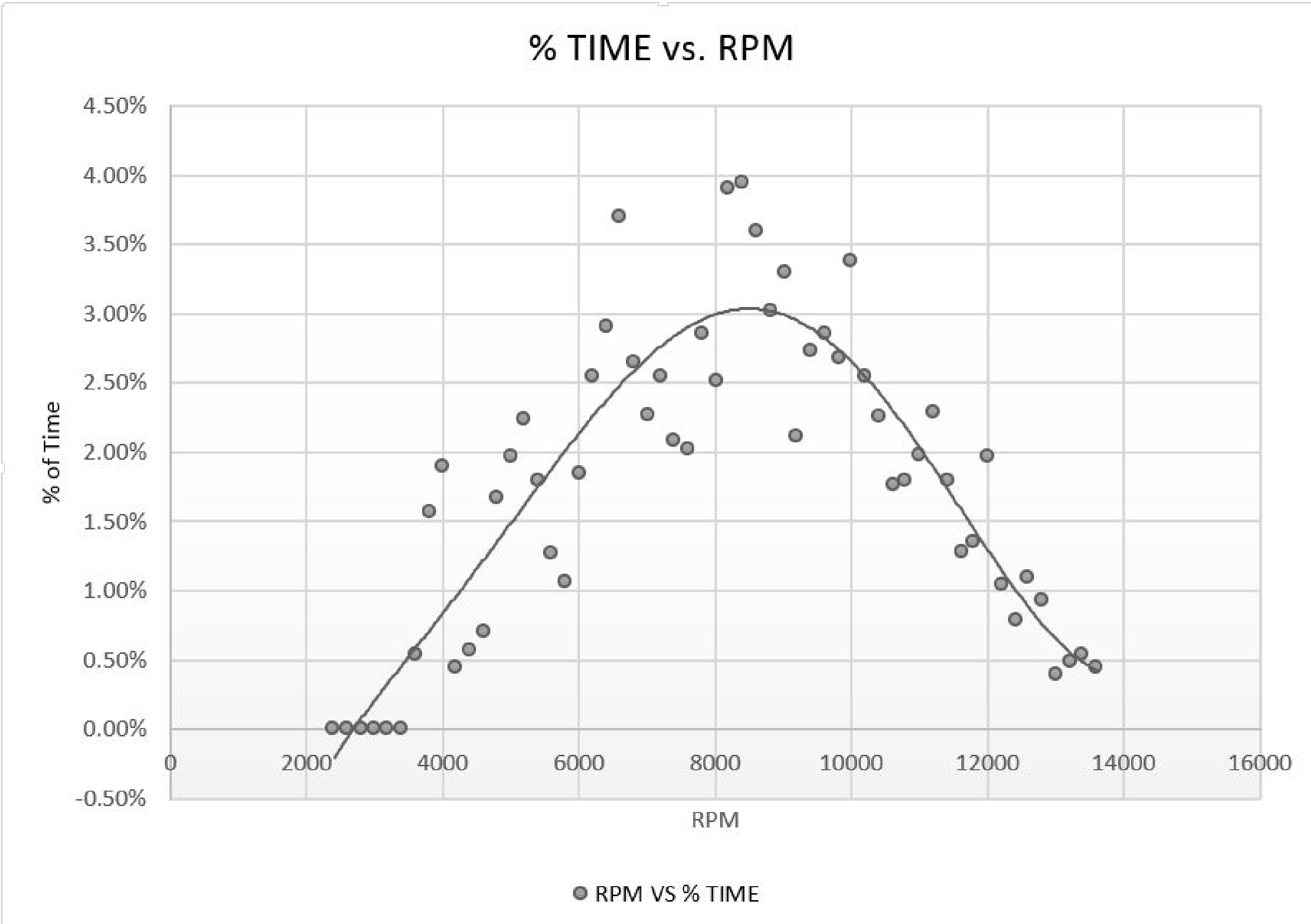


Figure 122: Graphing RPM vs % Time

E.1 Material Justification

What is required: High temperature resistance corrosion resistance are the main priorities.
Potential Material Options

Preliminary Options

- Stainless Steel
- 304
- 316
- 321
- Mild Steel
- Titanium

Stainless Steel
304 Stainless
Density: 8 g/cc
E:193-200GPa
Yield Strength: 215MPa
Melting Point: 1450 C

Advantages: Ductile, excellent drawing, forming and spinning properties.
As this material is appropriate for forming this should will allow for easier in house manufacture.

Disadvantages: If tubing thickness is too thin, they may be too easily deformed and disrupt ideal flow patterns.

321 Stainless
Density: 8 g/cc
E: 193-200GPa
Yield Strength: 240MPa
Melting Point: 1400 C
Advantages: Similar to 304, however, added Ti to the alloy allows to combat Chromium Carbide precipitate due to welding or elevated temps. Known uses include exhaust manifolds

Mild Steel
1008 Carbon Steel
Advantages: Minutely less dense.
Disadvantages: Poor corrosion resistance against exhaust condensate and 'road salt'

Titanium
Whilst it would be lighter, it is more than likely not feasible and so stainless steel seems to be the choice.
Advantages: Gives the same standard of reliability whilst allowing weight reduction. Good design critique.
Disadvantages: Expensive, hence not easily substituted if manufacturing errors occur.

Mild steel has poor corrosion resistance and will suffer from exhaust condensate over time. Of the stainless steel options 304 and 321 have similar characteristics, with the only signification factor separating the pair is the added Titanium to the composition of 321, enabling greater resistance to chromium carbide forming during welding.

E.2 Basic Flow Equations

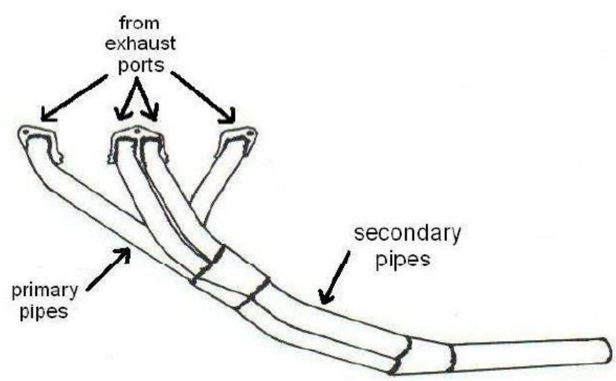


Figure 123: Arrangement of a 4-2-1 System. Source: (4)

$$P = \frac{850 \times ED}{RPM - 3} \tag{E.1}$$

Equation Governing Total Runner Length - Primary and Seconday

$$ID = \sqrt{\frac{cc}{(P+3) \times 25}} \times 2.1 \tag{E.2}$$

Internal Diameter of Primary Tubing

$$IDS = \sqrt{ID^2 \times 2} \times 0.93 \tag{E.3}$$

Internal Diameter of Secondary Tubing

$$P2 = P - P1 \tag{E.4}$$

Secondary Length Calculations

$$CL = \frac{ID2 - ID3}{2} \times CotA \tag{E.5}$$

Collector Length Calculation

$$ID3 = \sqrt{\frac{cc \times 2}{(P+3) \times 25}} \times 2 \tag{E.6}$$

Tail-Pipe Length.

E.3 Collector Considerations

Baffle Collector
-Primary tubes abruptly end and are all tapered and fed together into the tailpipe
-This causes a reflected wave to go back to the exhaust port: now if the wave is negative pressure wave and the exhaust valve is open then it promotes scavenging. Similarly, if a positive pressure wave is produced and the exhaust port is about to close, it will stop over scavenging from happening and force fresh fuel/air to stay in the cylinder.
-However, if the reverse happens then it will be detrimental

Merge Collector
-Primaries blend together into the tail pipe
-More reliant on inertia tuning as fewer pressure waves created without baffles.
-Taper of approximately 10 degrees is ideal, Testing from 9-12 degrees should be conducted to verify angles

Venturi Merge Collector -Primaries merge initially and subsequently flare out throught the collector
-This design ccelerates the flow in the exhaust and promotes scavenging. Usually only used when the tail pipe is a lot large relative to the primaries. I.e. not usually in 4-stroke engines.
-As flow is accelerated it additionally serves to reduce the amplitude of the pressure waves reflected back to the exhaust valve such that the exhaust gas momentum is less reliant on the pressure pulse waves

Split Interference Collector -Was considered due to its versatility of low/mid range power and not knocking too much of high end
-However history and theory suggests that 4 cylinder engines usually don't do well with this arrangement

E.4 Geometry Justification

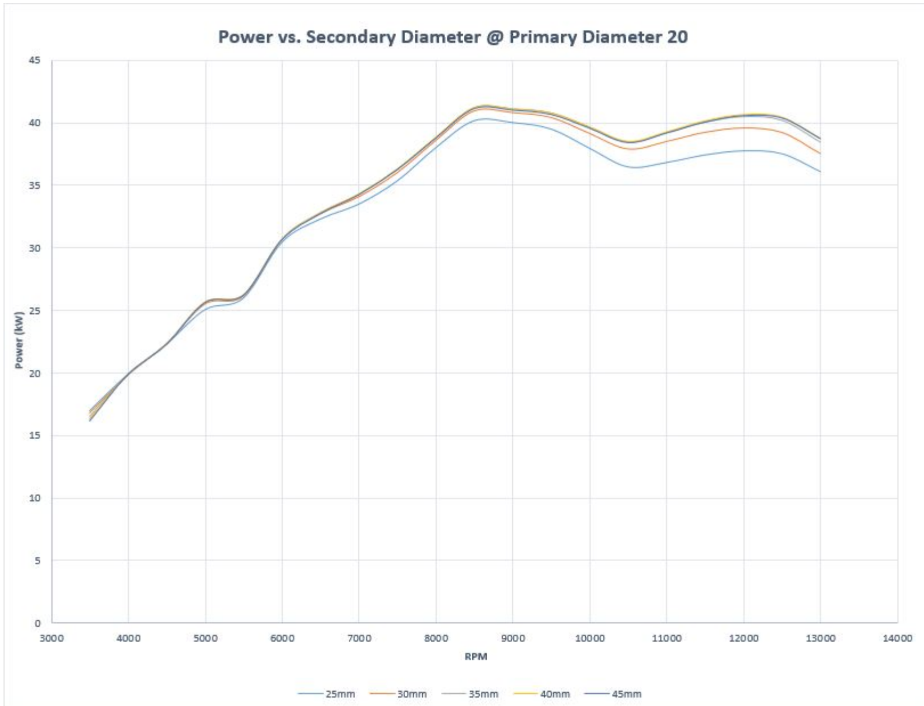


Figure 124: Power outputs with varying Secondary Diameter, with Primary Diameter 20mm

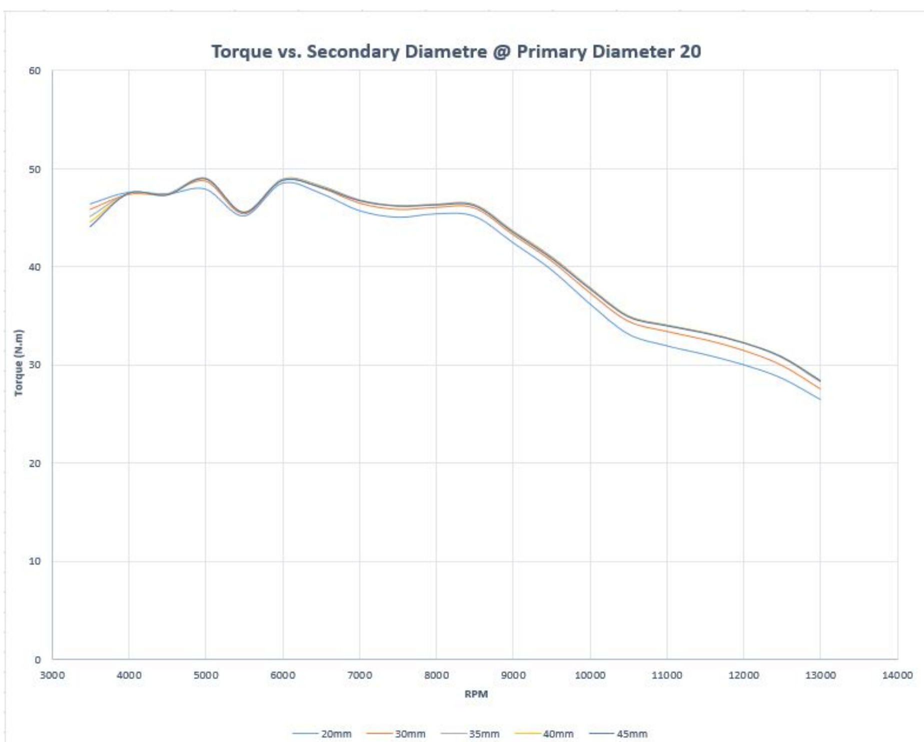


Figure 125: Torque outputs with varying Secondary Diameter, with Primary Diameter 20mm

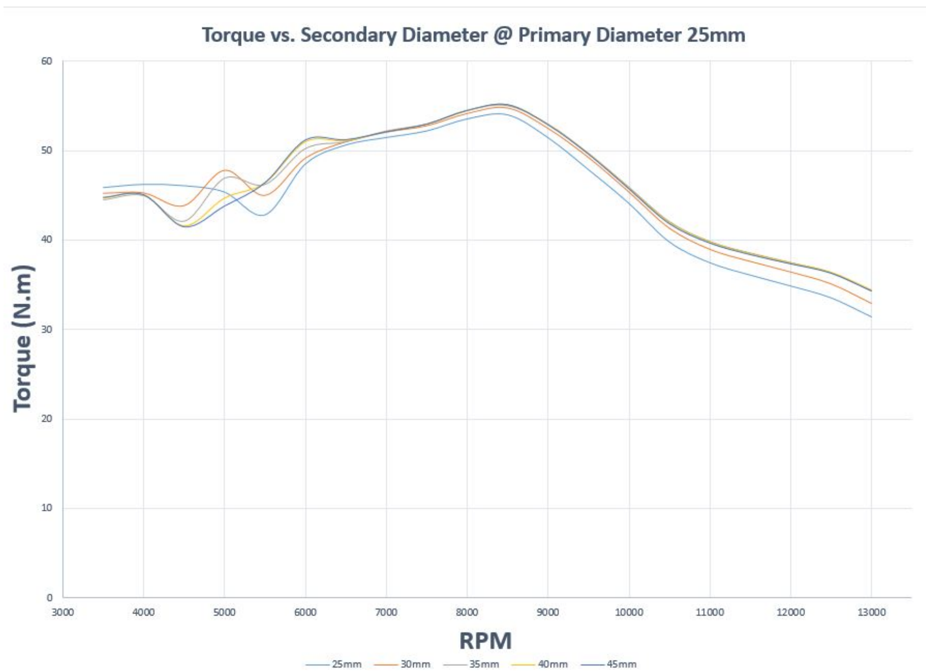


Figure 126: Torque outputs with varying Secondary Diameter, with Primary Diameter 25mm

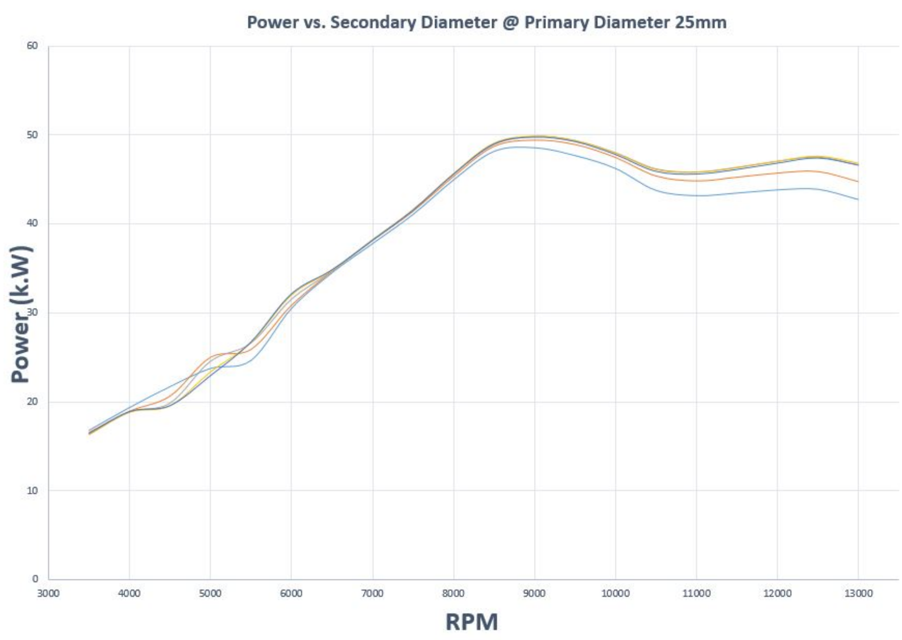


Figure 127: Power outputs with varying Secondary Diameter, with Primary Diameter 25mm

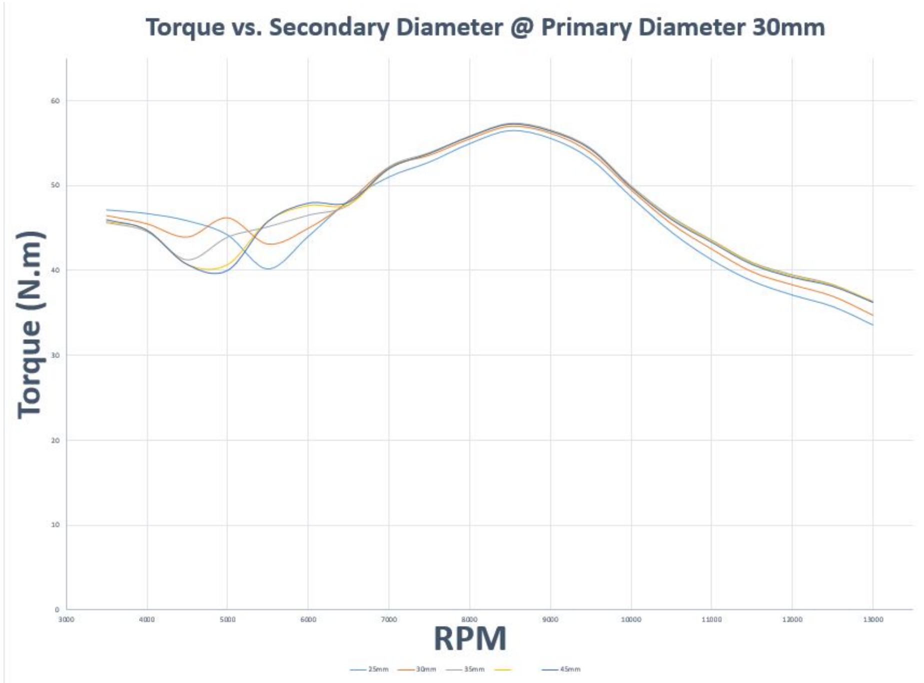


Figure 128: Torque outputs with varying Secondary Diameter, with Primary Diameter 30mm

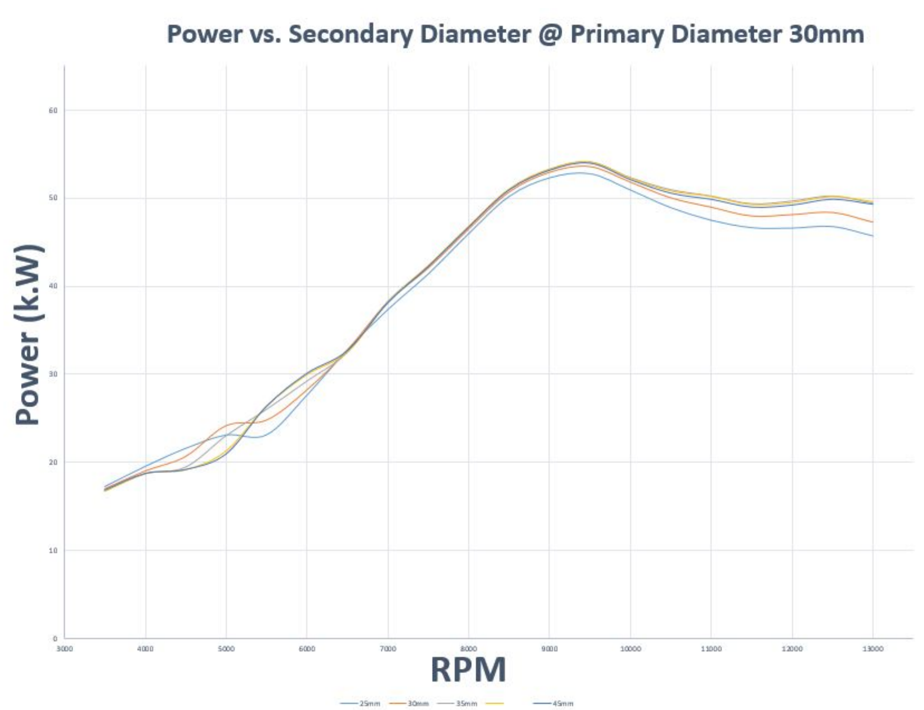


Figure 129: Power outputs with varying Secondary Diameter, with Primary Diameter 30mm

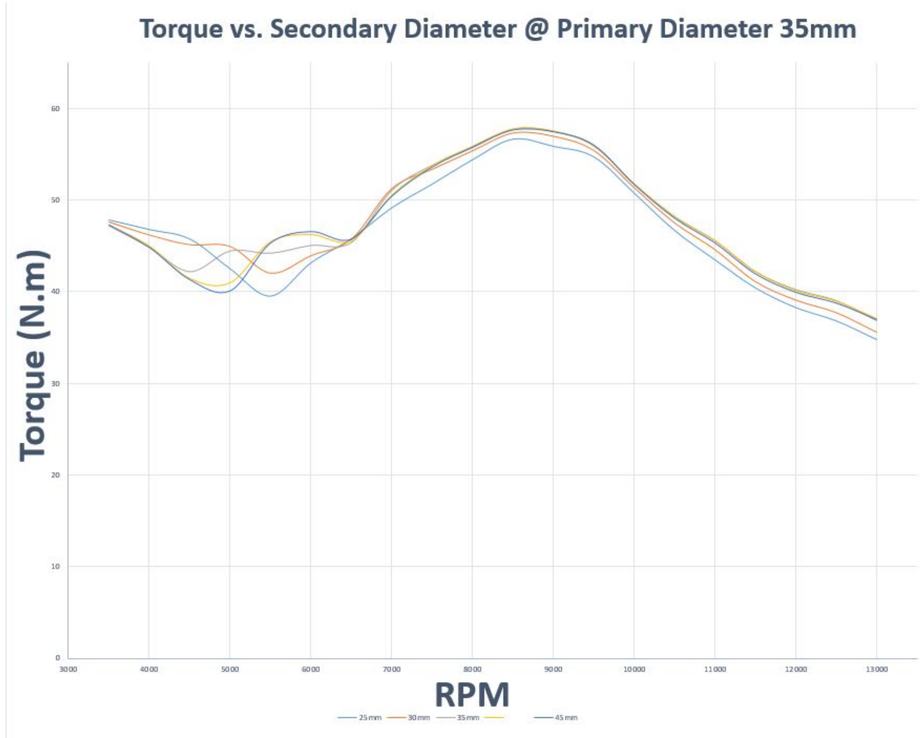


Figure 130: Torque outputs with varying Secondary Diameter, with Primary Diameter 35mm

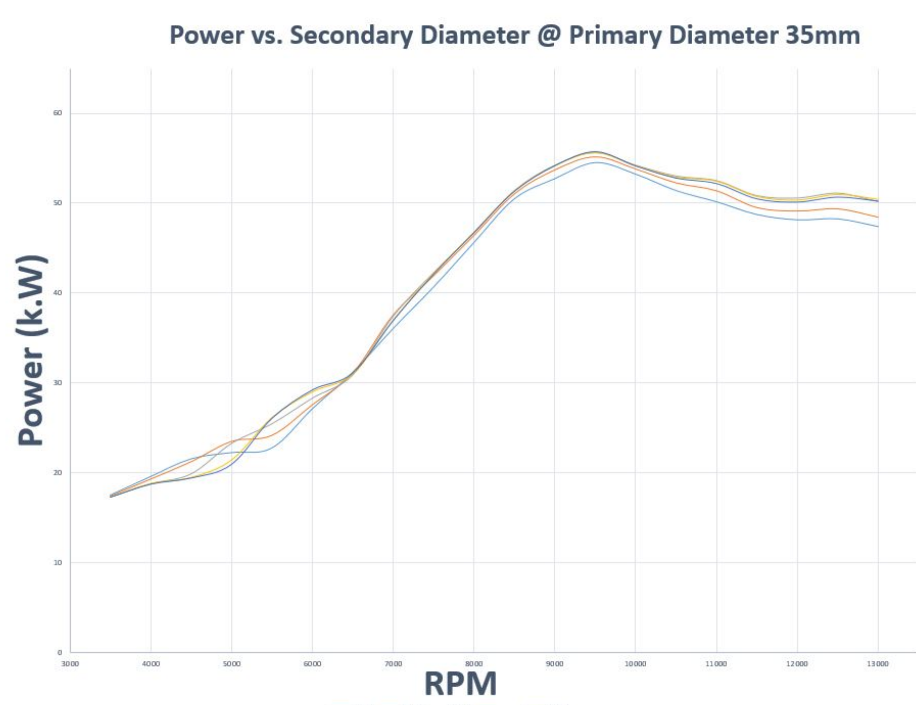


Figure 131: Power outputs with varying Secondary Diameter, with Primary Diameter 35mm

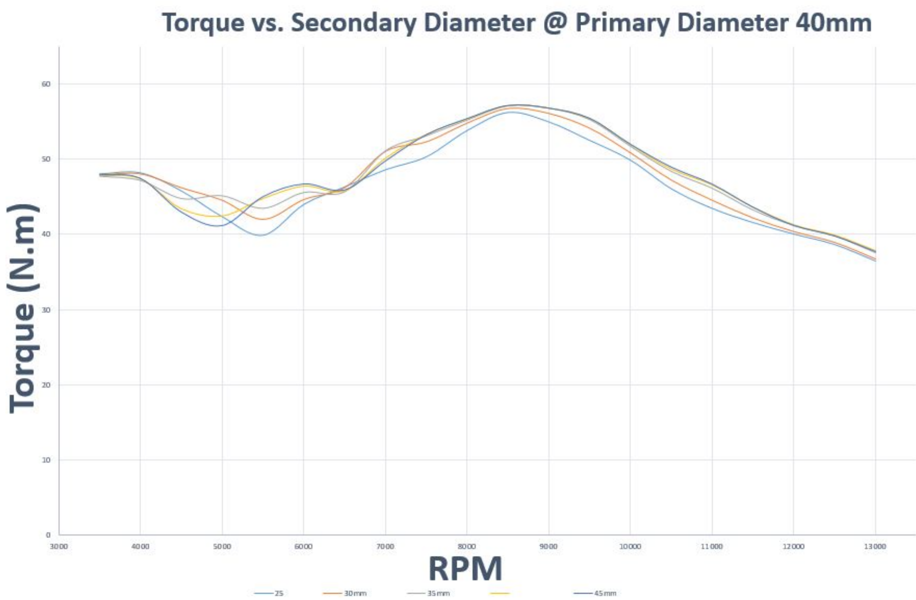


Figure 132: Torque outputs with varying Secondary Diameter, with Primary Diameter 40mm

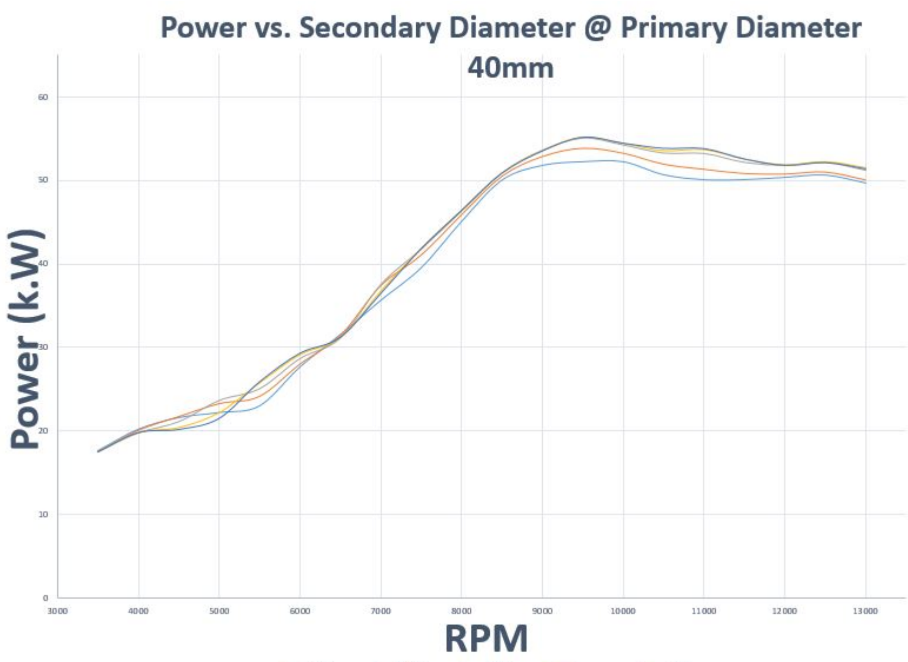


Figure 133: Power outputs with varying Secondary Diameter, with Primary Diameter 40mm

E.5 2014 Rules and Regulations

ARTICLE 3: EXHAUST SYSTEM AND NOISE CONTROL

IC3.1 Exhaust System General

IC3.1.1 Exhaust Outlet The exhaust must be routed so that the driver is not subjected to fumes at any speed considering the draft of the car.

IC3.1.2 The exhaust outlet(s) must not extend more than 45 cm (17.7 inches) behind the centerline of the rear axle,

and shall be no more than 60 cm (23.6 inches) above the ground.
IC3.1.3 Any exhaust components (headers, mufflers, etc.) that protrude from the side of the body in front of the main roll hoop must be shielded to prevent contact by persons approaching the car or a driver exiting the car.
IC3.2 Noise Measuring Procedure
IC3.2.1 The sound level will be measured during a static test. Measurements will be made with a free-field microphone placed free from obstructions at the exhaust outlet level, 0.5m (19.68in) from the end of the exhaust outlet, at an angle of forty-five degrees (45°) with the outlet in the horizontal plane. The test will be run with the gearbox in neutral at the engine speed defined below. Where more than one exhaust outlet is present, the test will be repeated for each exhaust and the highest reading will be used.
IC3.2.2 The car must be compliant at all engine speeds up to the test speed defined below.
IC3.2.3 If the exhaust has any form of movable tuning or throttling device or system, it must be compliant with the device or system in all positions. The position of the device must be visible to the officials for the noise test and must be manually operable by the officials during the noise test.
IC3.2.4 Test Speeds The test speed for a given engine will be the engine speed that corresponds to an average piston speed of 914.4 m/min (3,000 ft/min) for automotive or motorcycle engines, and 731.5 m/min (2,400 ft/min) for "industrial engines". The calculated speed will be rounded to the nearest 500 rpm. The test speeds for typical engines will be published by the organizers. An "industrial engine" is defined as an engine which, according to the manufacturers' specifications and without the required restrictor, is not capable of producing more than 5 hp per 100cc. To have an engine classified as "an industrial engine", approval must be obtained from organizers prior to the Competition.

E.6 Manufacturing

After considering a various selection of material, it was apparent that Stainless Steel 304 was best suited for our application.The process of manufacture was to use pre bent U-tubes, Donuts and straight section tubing to "piece" the exhaust system together, in order to save money on bending. While concurrently scoping out what dimensions of bent radii were readily available with the same material and WT while also making sure the SolidWorks model sections were consistent with what was available to purchase.

Figure 134: Purchased Material

Once our designs were frozen a jig was designed for the welding each all primary and secondary runners, as well as an additional jig to couple with the an engine to put together the entire system and verify it's accuracy. The process involved sectioning the exhaust into parts that only passed through two dimensions where possible, such that they could be welded and constrained more easily when placed into their jigs.

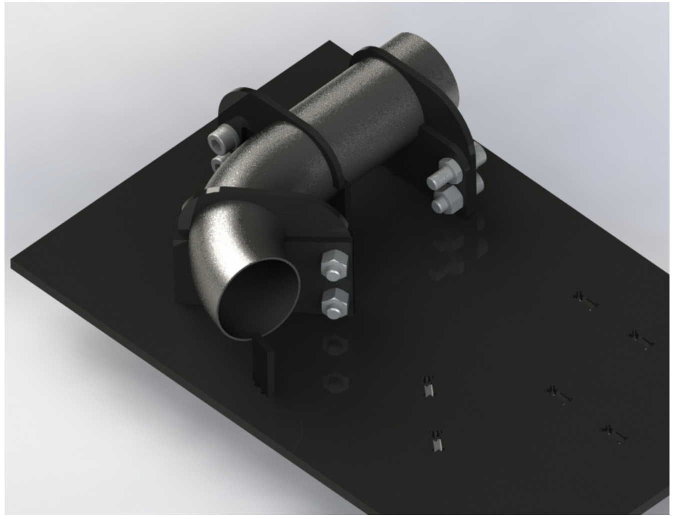


Figure 135: Part of Primary 1 in Jig

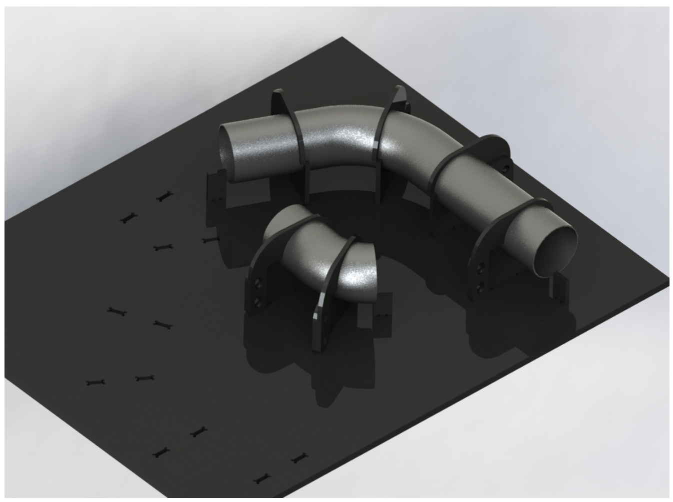


Figure 136: Primary 2 Jig

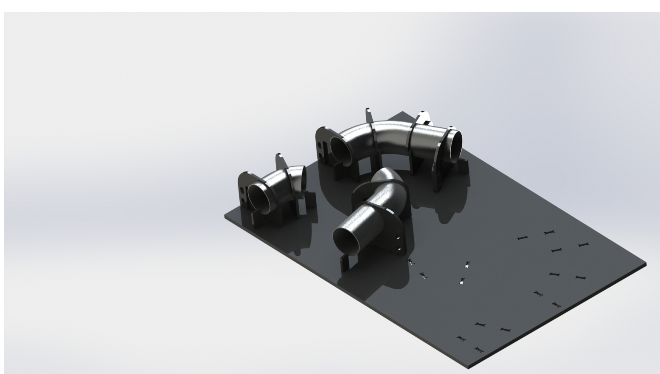


Figure 137: Primary 3 Jig

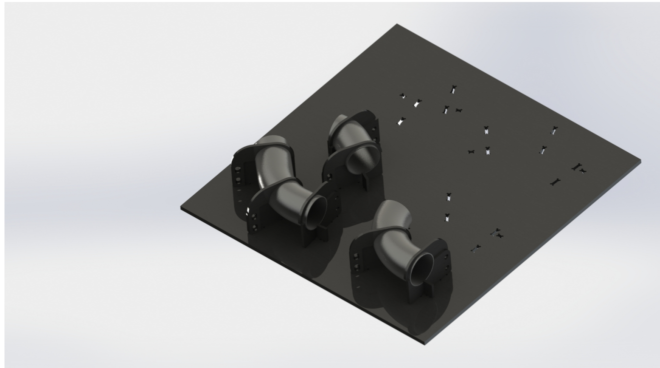


Figure 138: Primary 4 Jig

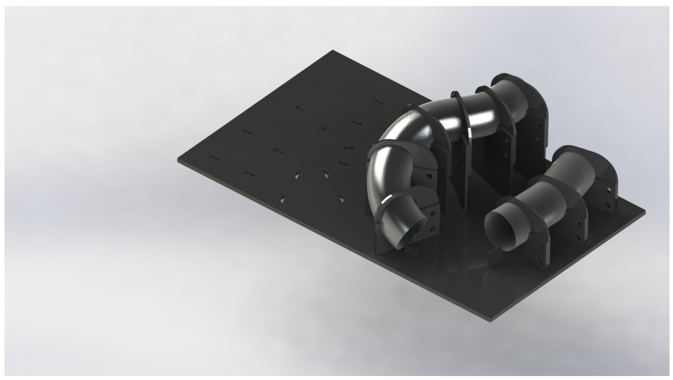


Figure 139: Secondary Jig

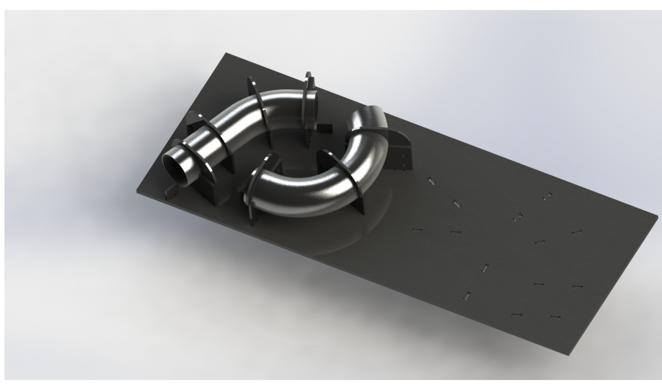


Figure 140: Secondary Jig

After each jig had been designed the tubing for each section of the exhaust, they all had to be measured and cut to size. To keep waste material to a minimum a simple jig from MDF was made for measuring each of the tubes accurately to size for the U-bends as well as the Stainless steel donuts.

Straight Sections = **millimeters** Bent Sections = **degrees**

Primaries

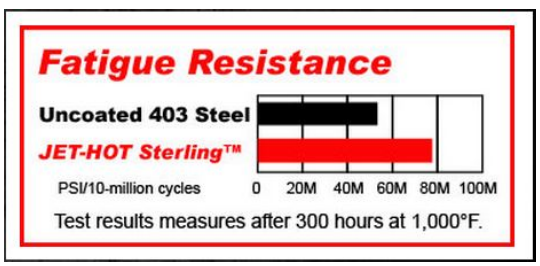
Cyl 1		Cyl 2		Cyl 3		Cyl 4	
STRAIGHT	103.97	STRAIGHT	26	STRAIGHT	34.36	STRAIGHT	29.63
BENT 63.5	75.15	BENT 38	50.52	BENT 38	56.72	BENT 38	36.94
BENT 63.5	62.26	STRAIGHT	63.16	STRAIGHT	63.82	STRAIGHT	16.05
STRAIGHT	49.94	BENT 63.5	62.37	BENT 63.5	64.74	BENT 63.5	75.21
		STRAIGHT	9.82	STRAIGHT	120.14	STRAIGHT	20.35
		BENT 38	76.15			BENT 63.5	48.79
		STRAIGHT	75.05			STRAIGHT	44.23

Secondaries

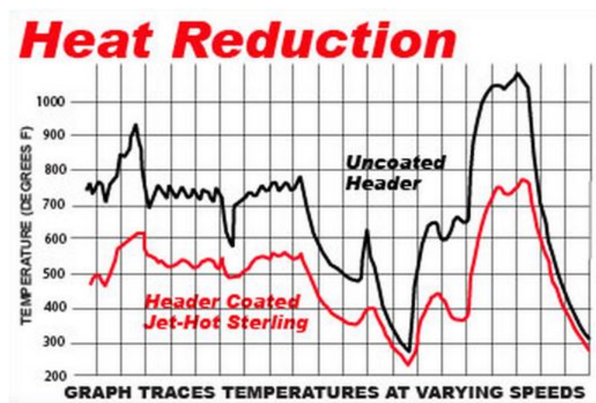
TOP		BOTTOM	
STRAIGHT	51.68	STRAIGHT	21
BENT 76.2	16.54	BENT 76.2	138.39
STRAIGHT	61.39	STRAIGHT	94.22
BENT 76.2	123.32	BENT 76.2	47.31
STRAIGHT	20.6	STRAIGHT	29
BENT 76.2	35.29		
STRAIGHT	20		

Figure 141: Tube Cutting Dimensions

E.7 Ceramic Coating



(a) Comparison of Fatigue Stress.



(b) Graph of Temperature Change.

3Source: $http : //www.jet - hot.com.au/advantages.htmMore_Power$

E.8 GT Power

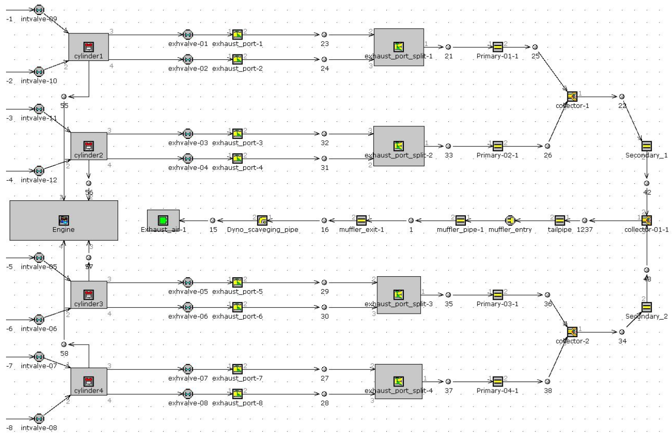


Figure 143: GT Power model of the exhaust - ideal case with no bends

F Cooling

F.1 Heat Dissipation Calculations

To find the amount of heat dissipation required by the system, first the average engine power output was found. This would coincide with the amount of heat to be dissipated as explained in the main text. To justify normalising the data based on the MAP, throttle position versus MAP was plotted to verify the expected near linear relationship. The data was gathered from a team track day.

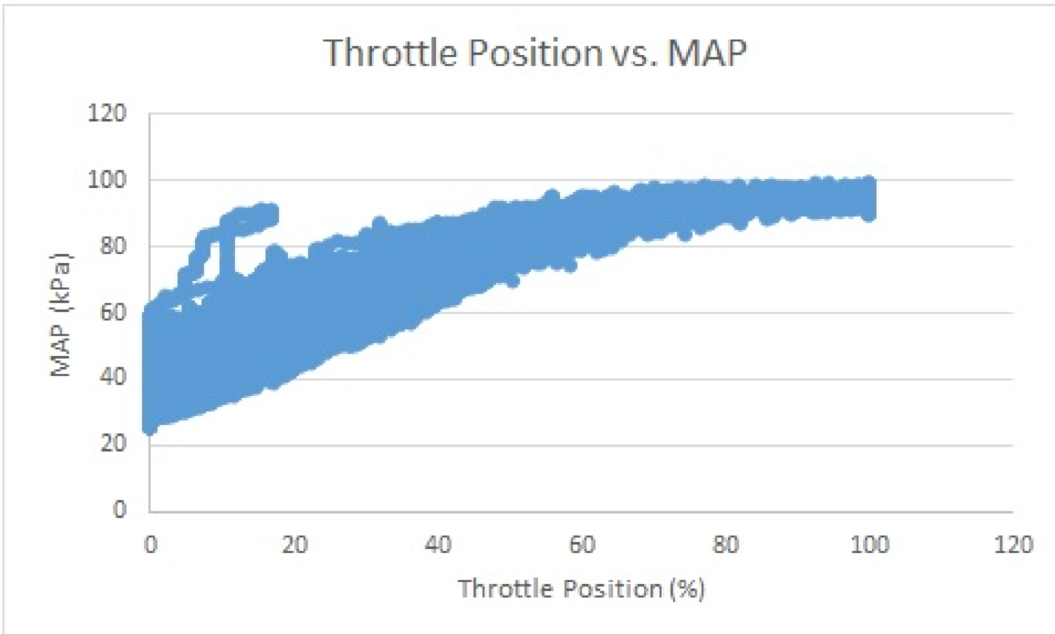


Figure 144: Throttle Position versus MAP

A normalised MAP percentage was computed, with the maximum MAP of 98.4 kPa being treated as the full throttle condition. Using the power curves which are based on full throttle conditions for a given RPM, the power was scaled to reflect the throttle position. The adjusted power at each instant of time was then averaged for all data points (over 50,000 data points) to produce the average power output of the engine. This was found to be 19.884 kW. A safety factor of 1.1 was used to provide a buffer for the designed system. However as power output is relatively well understood for the CBR600 engine, heat dissipation requirements are also relatively accurate. As such, the factor of safety here also reflects the expectation that the system will not be strained far from expectations. The overall heat dissipation requirements of the system are hence, 21.873 kW.

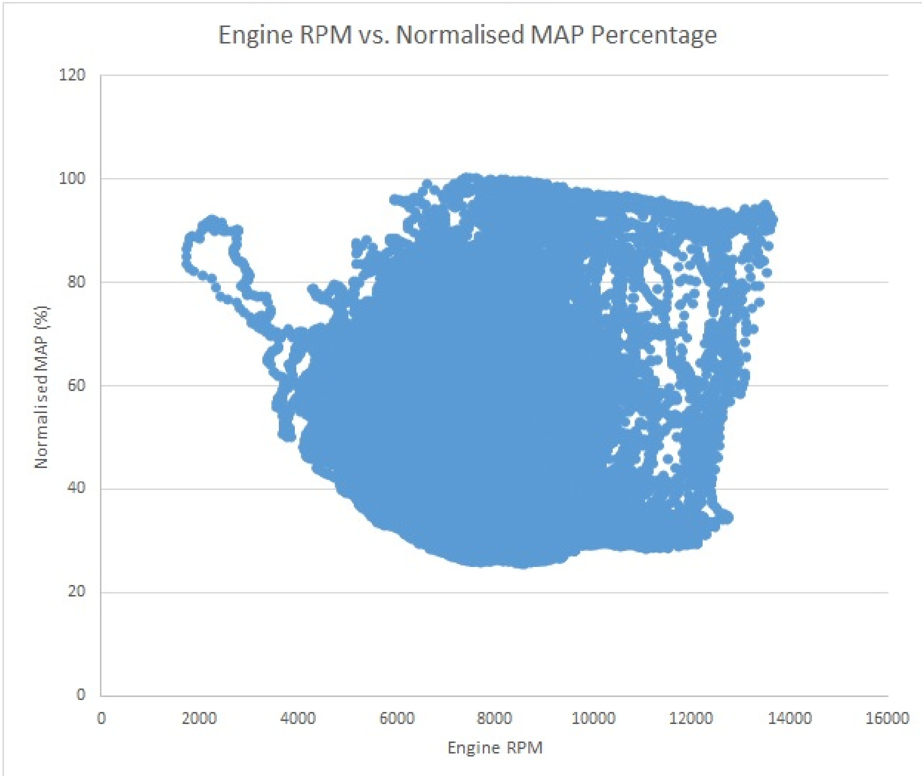


Figure 145: RPM versus Normalised MAP Percentage

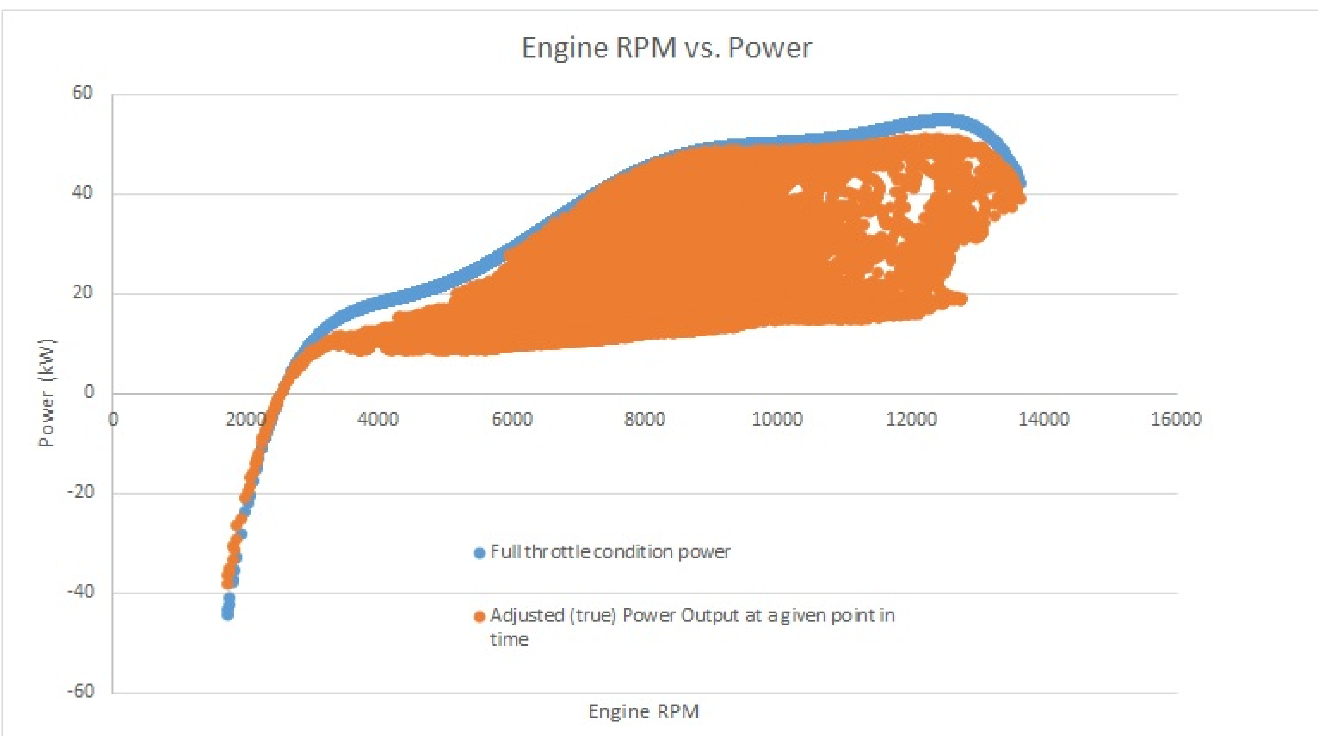


Figure 146: RPM versus Power

F.2 Heat Transfer Calculations

The heat transfer calculations conducted were for the following assumptions:

- 40 °C was the air temperature so as to design for a worst case scenario due to the FSAE competition being held in December.
- 95 °C was set as the water temperature entering the radiator from the engine.
- The velocity of the air through the inlet duct was taken as the speed of the car and then adjusted for the diverging duct to the radiator. The surface are of the inlet duct opening and the inlet duct geometry were based on the 2013 Cooling System.

The Effectiveness-NTU method was used for the heat exchanger calculations as it doesn't require knowledge of the temperatures at the outlets. These temperatures were not available from the gathered data last year.

The mass flow rates for water were computed from the flow rate data provided by the mechanical water pump. The flow rate of water is dependent on engine RPM. The specific heat of water at 95 °C is 4213 J/kg/K. The heat capacity rate of water (the hot fluid) was then computed.

Mass flow rates of air were computed from the adjusted velocity of air explained in the assumptions. The specific heat of air at 40 °C is 1005 J/kg/K. The cross sectional area of the inlet duct used was 0.094068 m^2. The heat capacity rate of air (the cold fluid) was then computed.

The smaller of the two heat capacities is used to calculate the maximum possible heat transfer. The two inlet temperatures are assumed earlier.

$$Q_{max} = C_{min}(T_{hi} - T_{ci}) \tag{F.1}$$

UA refers to the overall heat transfer coefficient multiplied by the heat transfer area. The NTU was calculated by varying values of UA in Matlab according to the equation:

$$NTU = \frac{UA}{C_{min}} \tag{F.2}$$

The effectiveness can then be calculated. The equation used is for a cross-flow geometry with both fluids unmixed.

$$\epsilon = 1 - exp[\frac{exp(-NCn) - 1}{Cn}],\ where\ n = N^{-0.22}\ and\ N\ is\ the\ NTU \tag{F.3}$$

The fluids are unmixed for a vehicle radiator since the plate fins force the air through a particular inter fin spacing and prevent it from moving in a transverse direction.

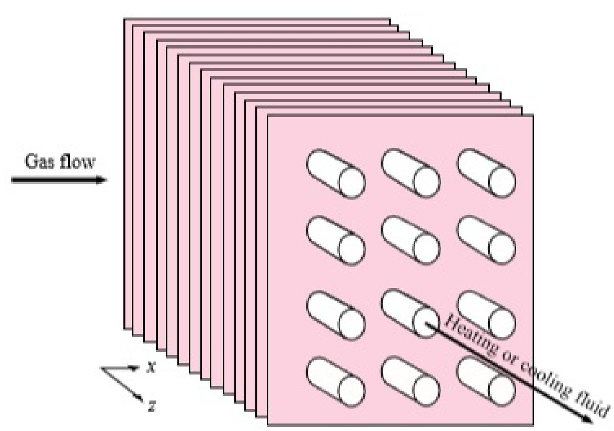


Figure 147: Cross-flow configuration for heat exchanger

The actual heat transfer is then computed:

$$Q = \epsilon Q_{max} \tag{F.4}$$

The Matlab file, Radiator_Calcs is setup such that it only shows solutions that satisfy the heat dissipation requirements at what is considered average conditions. This means using the mechanical water pump flow rates based on average operating RPM and air speeds generated at average velocity conditions. The outputted data took the form of 5 columns. A small section of this table is shown in Table 24. With 21.873 kW of heat to be dissipated, the second row in the table is selected as it meets this requirement and has the smallest value of UA. This solution will provide the lowest value for heat transfer area.

UA (W/K)	Q (W)	ϵ	Pump flow rate (L/min)	Air speed (km/hr)
680	21812	0.5879	36.8	22.8526
685	21901	0.5903	36.8	22.8526
685	21846	0.5930	36.8	22.6935
685	21846	0.5930	36.8	22.6935
690	21990	0.5927	36.8	22.8526

Table 29: Section of Matlab output for heat exchanger calculations

To compute the heat transfer area, the overall heat transfer coefficient must now be found. This requires finding the air and water side heat transfer areas first. The water side heat transfer area is based on internal radiator parameters, some of which are assumed and others are based on values from previous radiators used by the MUR team. The air side heat transfer area is based on external radiator parameters. Some of these are known by the manufacturer and the others are based on previous radiators manufactured for the team.

To calculate the individual heat transfer coefficients, the Nusselt numbers are required. The Nusselt number for water is calculated from the below equation based on tube flow with the conditions for use provided.

$$Nu_d = 0.012(Re_d^{0.87} - 280)Pr^{0.4} \tag{F.5}$$

$$Restrictions: \ 1.5 < Pr < 500, \ 3000 < Re_d < 10^6 \tag{F.6}$$

The Nusselt number for air is calculated from the below equation based on flow over a bank of tubes with the conditions for use provided.

$$Nu_d = CRe_{d,max}^n Pr^{0.36}(\frac{Pr}{Pr_w}) \tag{F.7}$$

$$Restrictions: \ 0.7 < Pr < 500, \ 10 < Re_{d,max} < 10^6 \tag{F.8}$$

The constants in the equation are:

- C=0.27 as the tubes are in-line as well as the Reynolds number for air being withing the range 10^3 and 2×10^5
- n=0.63 for the same reasons as above
- The Prandtl number ratio is assumed to be 0.8 based on the Prandtl numbers of the expected temperatures of air and the wall.

This gives a final equation for the Nusselt number of air as:

$$Nu_d = 0.27Re_{d,max}^{0.63}Pr^{0.36} \times 0.8 \tag{F.9}$$

The heat transfer coefficient is calculated for both air and water:

$$h = \frac{Nu\ k}{D} \tag{F.10}$$

$$\tag{F.11}$$

The overall heat transfer coefficient is calculated from the following equation:

$$U = \frac{1}{\frac{1}{h_o} + \frac{A_o ln(\frac{r_o}{r_i})}{2\pi k L} + \frac{A_o}{A_i}\frac{1}{h_i}} \tag{F.12}$$

The Matlab file Get_U computes the overall heat transfer coefficient, which is found to be 116.756 W/m^2/K.

This means the heat transfer area is calculated to be:

$$A = \frac{685}{116.756} \tag{F.13}$$

$$A = 5.87\ m^2 \tag{F.14}$$

```
%% Radiator_Calcs
clear all
clc
format long
Th1=95;                                                  % Temperature of water from the engine
Tc1=40;                                                  % Temperature of air
mdoth=[];                                                % Mass flow rate of  (given by pump) [kg/s] FOR 6000 RPM
mdotc=[];                                                % Mass flow rate of air  [kg/s]
cph=4213;                                                % Specific heat of water at 95 degrees [J/kgK]
cpc=1005;                                                % Specific heat of air [J/kgK]
UA=[600:5:750];                                          % Varying UA
l=0;
for k=1:length(UA)
    for j=1:length(mdoth)
        Ch=mdoth(j)*cph;                                 % Capacity rate of water
        Cc=mdotc(j)*cpc;                                 % Capacity rate of air
        Cmin=min(Cc,Ch);                                 % Lowest capacity rate
        Cmax=max(Cc,Ch);                                 % Highest capacity rate
        C=Cmin/Cmax;                                     % Ratio of capacity rates
        Qmax=Cmin*(Th1-Tc1);                             % Maximum possible heat transfer
        NTU = UA(k)/Cmin;                                % Number of Transfer Units
        N=NTU^(-0.22);
        epsilon(k)=1-exp((exp(-NTU*C*N)-1)/(C*N));  % Heat exchange effectivness (pg 546)
        Q(k)=epsilon(k)*Qmax;                            % Actual heat transfer [W]
        if Q(k)>21800
            if mdoth(j)>0.4973 && mdoth(j)<0.6704 && mdotc(j)>0.571 && mdotc(j)<0.674
            l=l+1;
            mdotckmhr=(mdotc(j)/(1.124*0.094068))*3.6;
            mdothlmin=mdoth(j)*60;
            U_Q_epsilon_mdoth_mdotc(l,:)=[UA(k) Q(k) epsilon(k) mdothlmin mdotckmhr]
            end
                end

    end
end
```

```
%% Get_U
close all
clear all
clc
%Channels
No_c=28;                                %Number of channels
C_h=0.002;                              %Channel height [m]
C_w=0.04;                               %Channel width [m]
C_t=0.0001;                             %Channel thickness [m]
C_l=0.36;                               %Channel length [m]
W_p_c=0.08;                             %Wetted perimeter of a single channel[m]
T_W_p_c=W_p_c*No_c;                     %Total wetted perimeter of all channels [m]
C_a_c=0.000039;                         %Cross sectional area for a single channel[m^2]
T_C_a_c=C_a_c*No_c;                     %Total cross sectional area of all channels [m^2]
Dh_w=0.00195;                           %Hydraulic diameter [m]
A_w=W_p_c*C_l*No_c;                     %Water side heat transfer area
%Fins
No_f=200;                               %Number of fins per row
No_r=24;                                %Number of rows
f_h=0.002;                              %Fin height [m]
f_w=0.008;                              %Fin width [m]
f_t=0.0002;                             %Fin thickness [m]
f_l=0.036;                              %Fin length [m]
W_p_f=0.0172;                           %Wetted perimeter [m]
T_W_p_f=W_p_f*No_f*No_r;                %Total wetted perimeter [m]
C_a_f=(f_h-2*f_t)*(f_w-f_t);            %Cross sectional area for a single fin [m^2]
T_C_a_f=2*C_a_f*No_f*No_r;              %Total cross sectional area of all fins [m^2]
Dh_a=0.005805;                          %Hydraulic diamter [m]
A_a=W_p_f*f_l*No_f*No_r*2;              %Air side heat transfer area [m^2]
%% Properties of water
cph=4213;                               %Specific heat of water at 95 degrees [J/kgK]
u_w=0.0002985;                          %Dynamic viscosity at 95 degrees [kg/ms]
rho_w=961.85;                           %Density of water at 95 degrees [kg/m^3]
V_w=34.71401159;                        %Volume flow rate of water, average of solutions outputted for UA [L/min]
v_w=(((V_w/1000)/60)/0.00117);          %Speed of water [m/s]
k_w=0.58;                               %Thermal conductivity of water [W/mK]
Re_w=(rho_w*v_w*Dh_w)/u_w;              %Reynolds number of water
Pr_w=(cph*u_w)/k_w;                     %Prandtl number of water
Nu_w=0.012*(Re_w^(0.87)-280)*Pr_w^(0.4);%Nusselt number of water: pg 312 for tube flow
h_w=((Nu_w)*(k_w))/(Dh_w);              %Heat coefficient [W/m^2K]
%% Properties of air
cpc=1008;                               %Specific heat of air at 55 degree (average) [J/kgK]
u_a=0.000020345;                        %Dynamic viscosity of air at 55 degrees (average) [kg/ms]
rho_a=1.082;                            %Density of air [kg/m^3]
v_a=11.5;                               %Speed of air [m/s]
k_a=0.02815;                            %Thermal conductivity of air [W/mK]
Re_a=(rho_a*v_a*Dh_a)/u_a;              %Reynolds number of air
Pr_a=(cpc*u_a)/k_a;                     %Prandtl number of air
Nu_a=0.27*Re_a^(0.63)*Pr_a^(0.36)*0.8;  %Nusselt number of air
h_a=Nu_a*k_a/Dh_a;                      %Heat coefficient of air [W/m^2K]
%% U calculations
k_Al=207;                               %Thermal conductivity of aluminium assumed at 95 degrees [W/mK]
R=A_a/A_w;                              %Ratio of air side heat transfer area to water side heat transfer area
one_on_h_a=1/h_a;
r_over_h_w=R/h_w;
rt_over_k=(A_a*log(Dh_a/Dh_w))/(2*pi*k_Al*C_l*No_c);
U=1/(one_on_h_a+r_over_h_w+rt_over_k)   %Overall heat transfer coefficient [W/m^2K](Uo)
```

F.3 Radiator Design

The size of the radiator was determined by the design space available as well as consideration to how much heat transfer area it would have. A technical drawing can be seen below showing the radiator dimensions, inlets and outlets, and the bungs at the inlet and outlet for temperature sensors to be installed. This radiator has a heat transfer area of approximately 4.23 m^2. This is below the computed heat transfer area shown earlier. Those calculations did not take into account the effects the fan would have and as such provided an overestimate of what would be required with a fan.

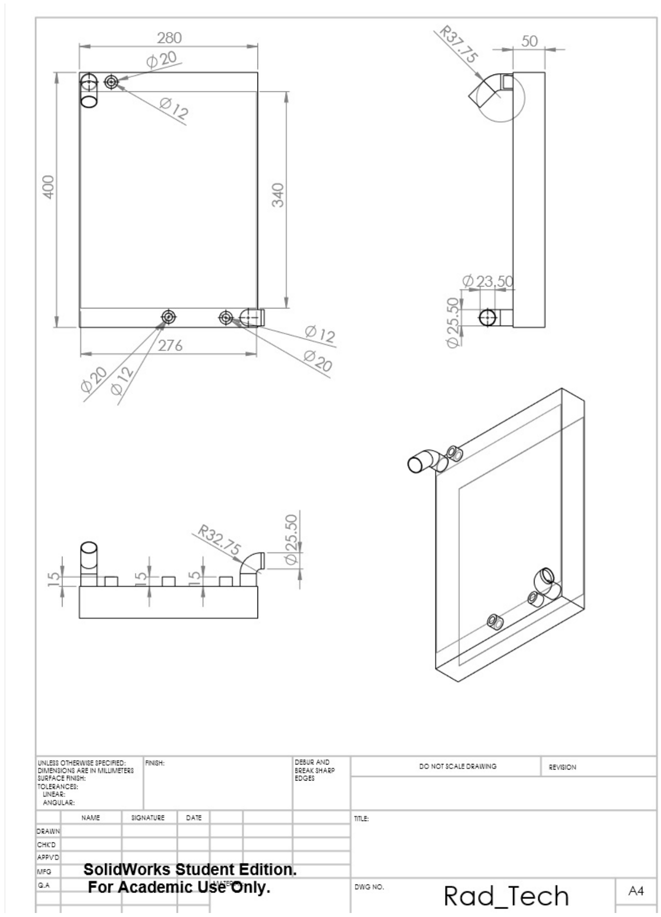


Figure 148: Radiator Technical Drawing

There was a preference to incorporate a dual pass radiator as opposed to the single pass one ultimately used. Dual pass radiators tend to be roughly 7% more efficient than the same size single pass, however the configuration of the radiator, with the vertical side being the major length, meant that for a dual pass to be possible, the core area would have to be substantially reduced. The other way to have incorporated it would have been to have both the inlet and outlet at the top or at the bottom of the radiator. However this would have lead to pump efficiency losses due to the height of which the water would have had to be pumped, essentially nullifying the gains the dual pass may have provided.

F.4 Cooling Fan and Shroud Design

The shroud was designed to be a venturi shape, a converging duct that causes a reduction in pressure. The lower the pressure region is downstream of the radiator, the higher the pressure drop across it and hence the faster the flow of air, aiding heat transfer.

The SPEF3500 straight blade fan from SPAL was re-used this year due to it's sufficient performance in previous years as well as to save money. The below table shows the fan's specifications.

SPEF3500	
Size	9"
Max Flow	596 CFM
Max Current	8.1 A
Weight	1 kg

Table 30: SPEF3500 specifications

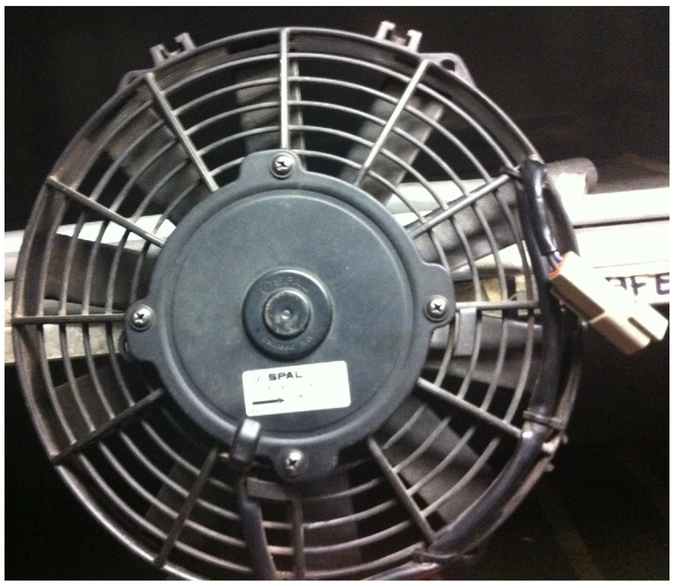
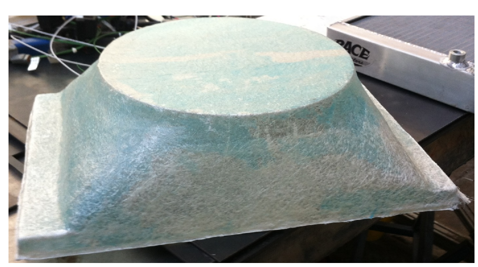


Figure 149: SPAL SPEF3500

The shroud was manufactured out of fibreglass. A mould was first produced into shape and then fibreglass was laid up to produce the structure.

(a) Shroud mould



(b) Final fibreglass shroud

One of the earlier CFD simulations run involved using a closed duct, the outlet being given a type wall boundary condition. At the time of this simulation the length of the duct was unknown, so its dimensions are assumed. This simulation had more assumptions overall that were perhaps not as accurate a reflection of real conditions, but were used mainly to produce an initial result.

The equations solved for this were the k-ϵ and energy equations. The radiator was modelled as a porous zone with the same input values as defined earlier. The velocity inlet boundary condition, as opposed to a pressure inlet one, using an approximation for the velocity based on the chassis team's CFD of the full vehicle was used. The boundary condition for the fan was to use a pressure outlet, with that pressure drop set the same as that for the exhaust fan case explained earlier. The geometry, mesh and pressure contours are shown in figure 151, 152 and 153 below.

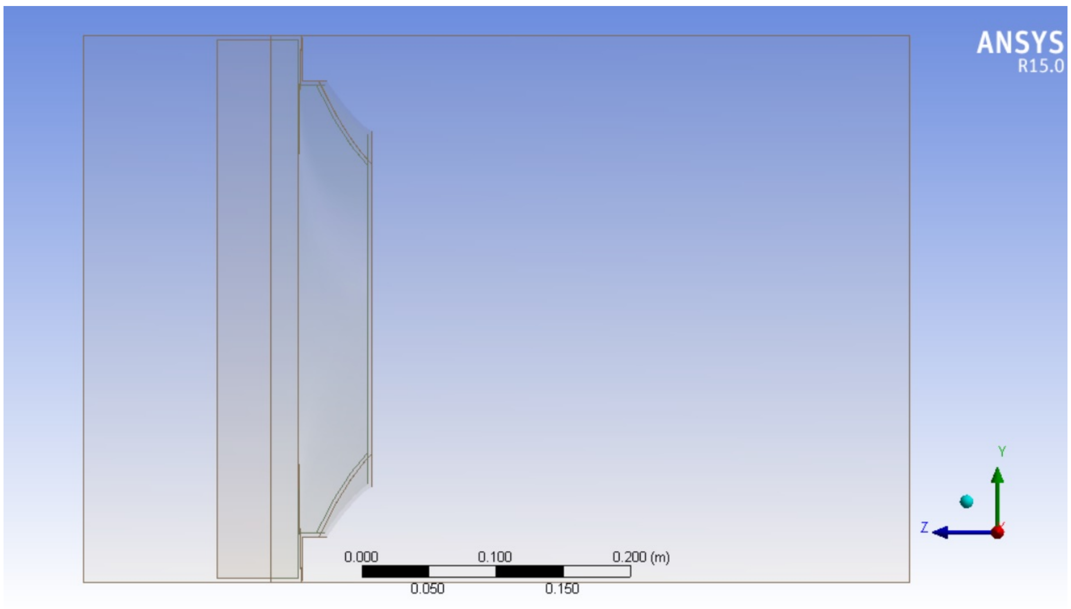


Figure 151: Geometry of radiator, shroud and inlet duct

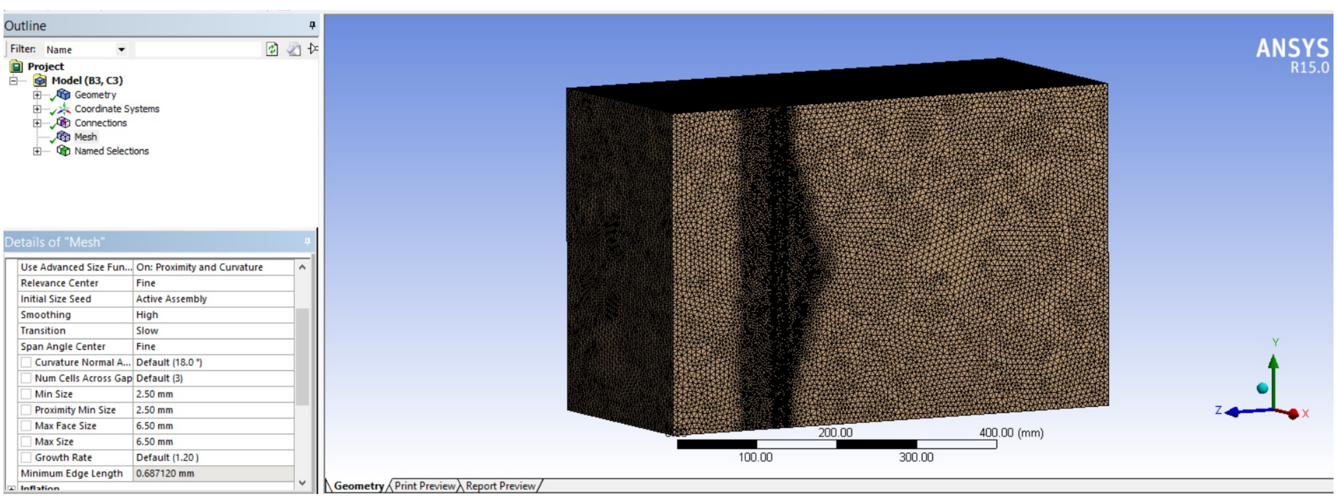


Figure 152: Mesh of the geometry

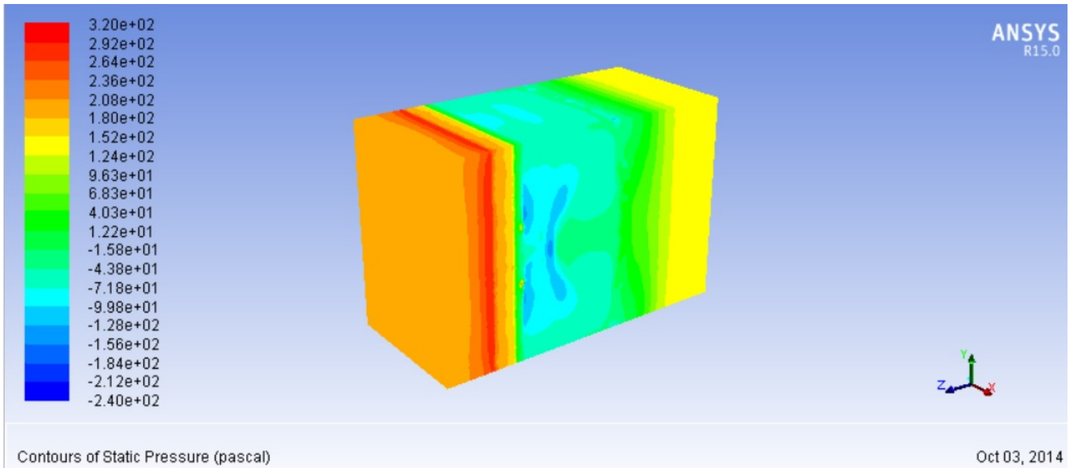


Figure 153: Pressure contours for radiator, shroud and inlet duct with fan modelled as pressure outlet

The result is a pressure drop of 560 Pa. Whilst the value itself seems relatively reasonable, a key reason for changing the inlet condition to a pressure outlet was that the pressure upstream of the radiator of about 300 Pa is significantly higher than the 120 Pa the chassis team's simulation was predicting. The wall condition at the outlet was also changed for the later simulation to reflect real conditions as well as the duct size being adjusted to what was expected to be manufactured for the vehicle.

The geometry for the simulation run in the Cooling Fan and Shroud section and the mesh are shown in figure 154 and 155.

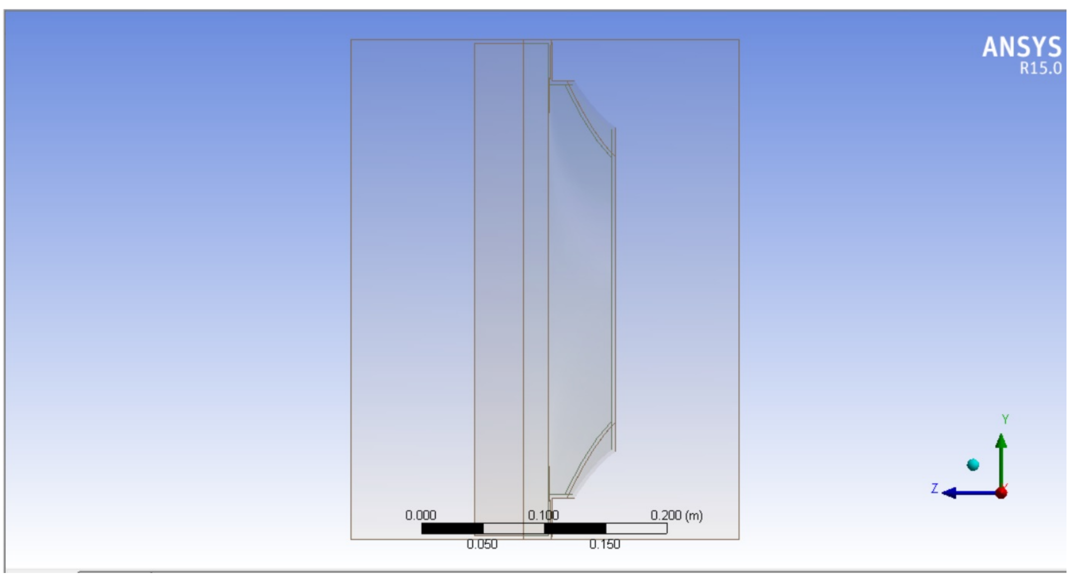


Figure 154: Geometry for the case with the more accurate duct size

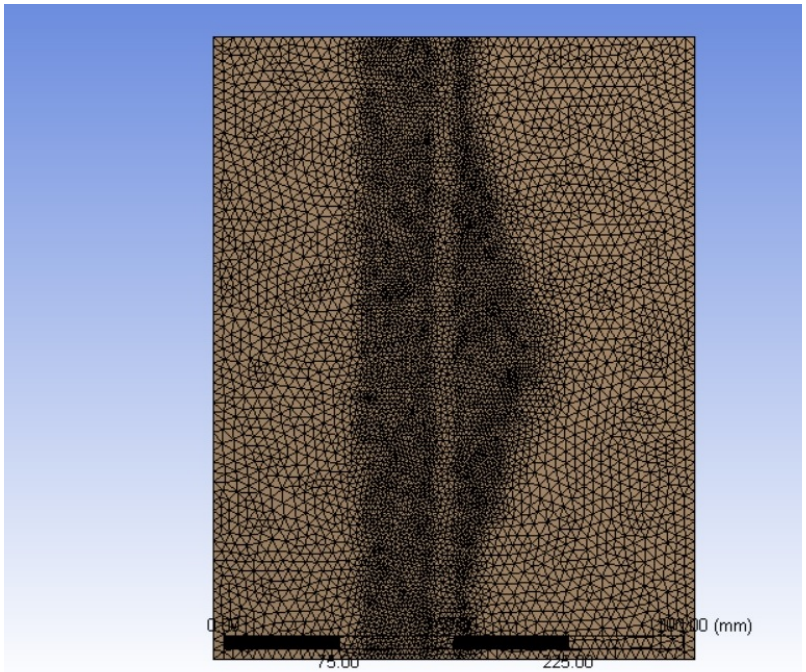


Figure 155: Mesh of the geometry

From the 494 Pa pressure drop, Bernoulli's equation was used to estimate the velocity through the radiator. The equation is simplified by assuming the no change in height, and that the velocity is largely generated by the fan leading to:

$$v = [\frac{2}{\rho} \times \Delta P]^{\frac{1}{2}} \tag{F.15}$$

$$v = 29.65\ m/s \tag{F.16}$$

This velocity is then used in computing the mass flow rate of air through the radiator, with this value being used in the Matlab code Radiator_Calcs. The mass flow rate of air is calculated to be 3.13 kg/s and with this values for UA are found. The lowest UA value is divided by the overall heat transfer coefficient, U, to give the heat transfer area of 4.11 m^2.

F.5 Mechanical and Electrical Water Pumps

The mechanical water pump used is the stock Honda CBR 600RR one. It is driven off the engine crankshaft. This makes the pump flow rates RPM dependent. A test conducted in 2012, plotting RPM versus pump flow rate, is shown in figure 156.

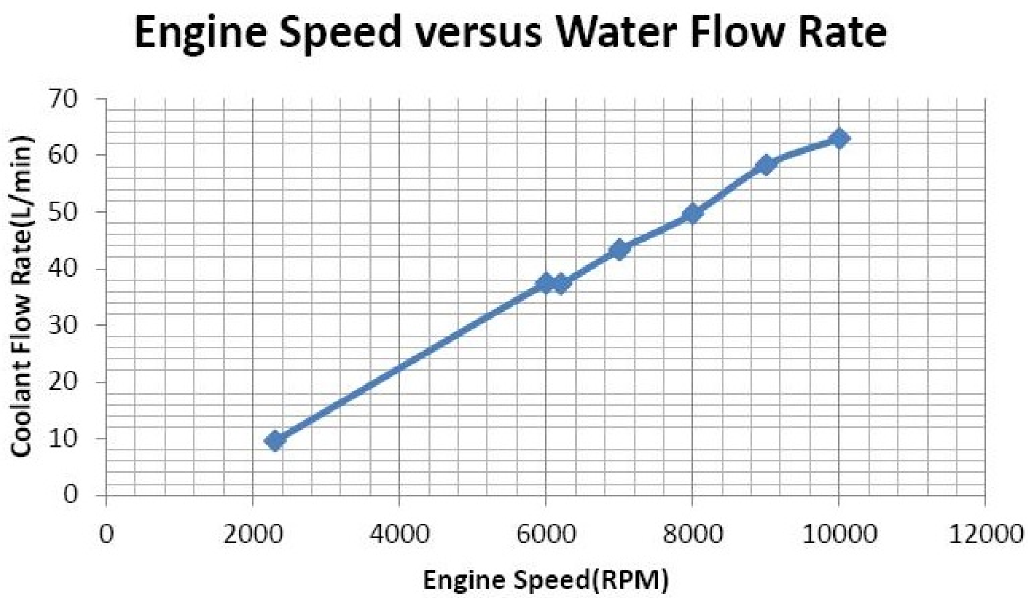


Figure 156: RPM versus flow rate for mechanical water pump

The electrical booster pump used is the Davis Craig EWP80 with the specifications listed in table ??.

Davis Craig EWP80	
Motor	12 V brushless
Operating Voltage	9-15V DC
Max Current	1.3 A
Flow Rate	15 L/min @ 10kPa
Operating Temperature	-40 °C to 120 °C
Pump Design	Recirculating centrifugal
Motor life	15000 hours @ 80 °C
Pump Weight	245 g
Pump Material	Nylon 66, 30% glass filled
Burst Pressure	250 kPa minimum

Table 31: Davis Craig EWP80 specifications

Investigation into using an electrical water pump as the primary pump in the system yielded the result that it wasn't feasible. For one the cost for an electrical water pump is expensive, particularly with the budget constraints the team face. Secondly, the size of the pump make it difficult to package as the rear of the vehicle is very congested. Finally, the flow rates it provides are well below those provided by the mechanical water pump, and the added benefit of the mechanical water pump is it doesn't require any electrical power.

F.6 Header Tank and Catch Can

The header tank is the highest point in the cooling system. It acts as both a water fill point and a pressure relief valve for when the pressure in the cooling system exceeds 20 Psi. The header tank has an inlet pipe that connects a hose from the thermostat and an outlet pipe that has a hose that goes to the inlet of the radiator. Along this same line there is a diverging route to the oil cooler.

Figure 157: Header Tank (yet to be welded)

The catch can is an overflow tank, where water that has been relieved from the cooling system flows to. It is also used for the lubrication system for when the oil pressure in the hydraulic accumulator is relieved. The catch can is a required component by FSAE rules for both the cooling and lubrication systems. Whilst two separate catch cans can be used for each, the pictured design has a separarting wall down the middle so that it can be used for both fluids. This is convenient as it is often difficult to find space at the rear of the vehicle for separate catch cans to be used.

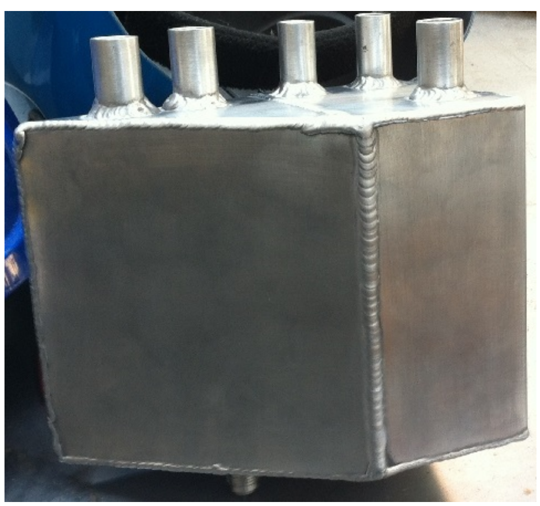
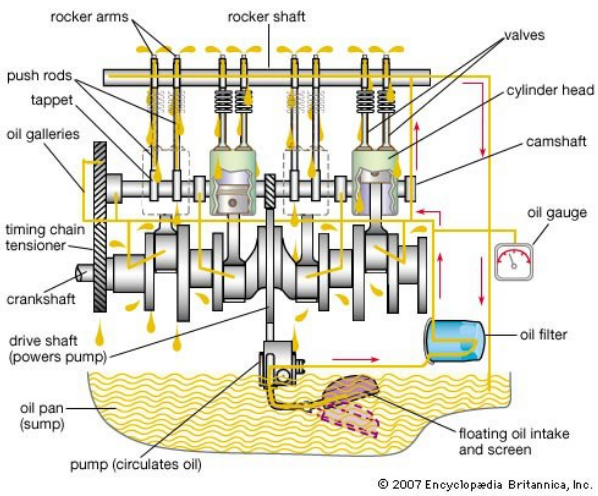


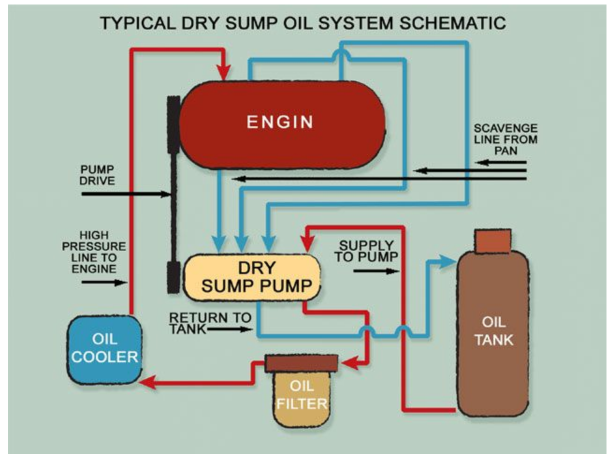
Figure 158: Catch Can

G Lubrication

G.1 Comparison between dry sump and wet sump lubrication system



(a) Wet sump system schematic



(b) Dry sump system schematic

The working principle for the wet sump and dry sump lubrication system is shown in figure 159a and 159b. The wet sump lubrication system uses a relatively deeper sump pan which also acts as a oil container. The advantage of the wet sump lubrication system is the ease of maintenance. The potential risks of implementing a wet sump lubrication system is the oil sloshing under lateral and longitudinal acceleration conditions which may lead to air scavenge issues. Furthermore, a deeper sump pan will increase the COG since the engine would need to be raised to compensate.

For the dry sump lubrication system, an external scavenge pump and an oil tank are added to create an external oil circuit. The positive of implementing this system is a thinner sump pan can be utilised, reducing the COG of the vehicle. The negative is complexity of the routing, vastly increasing the difficulty of system maintenance.

G.2 Design option for wet sump pan

The sump pan is the most significant part for the wet sump lubrication system. Various concepts were investigated. Three of these are proposed, with the advantages and disadvantages of implementing them.

1. The CBR 600 RR stock oil sump

(a) CBR 600 RR stock wet sump

(b) CBR 600 RR stock internal PRV

The stock CBR 600 RR engine sump is shown in figure 160a. It has a 2 litre capacity to contain oil. The sump consists of a relatively larger container body, with a shallow oil containing slot at the bottom. The oil is picked up through a long extended pipe internally. A Honda stock pressure relief valve (shown in figure 160b) is mounted in the sump to regulate the oil pressure inside the engine block.

The shape of the sump means it is reliable against issues caused by oil sloshing. The stock internal pressure relief valve has also simplified packaging. However, the height of the CBR 600 RR stock sump makes it impossible to implement without significantly raising the engine and all subsequent engine mounting points.

2. The 40 mm flat sump
The 40 mm flat sump is implemented by some of the FSAE teams. It has a height that is a lot lower than the stock sump, making it more feasible in meetomg the engine mounting constraints. Also the height of the Honda internal PRV is approximately 35 mm, which means the 40 mm flat sump allows the utilisation of the stock internal PRV. Based on the area of the bottom of the engine block, the volume capacity is approximately 1.3 litres which means the 40 mm flat sump is likely more reliable in maintaining the engine oil pressure and resisting the oil sloshing. The 40 mm flat wet sump concept is shown in figure 161

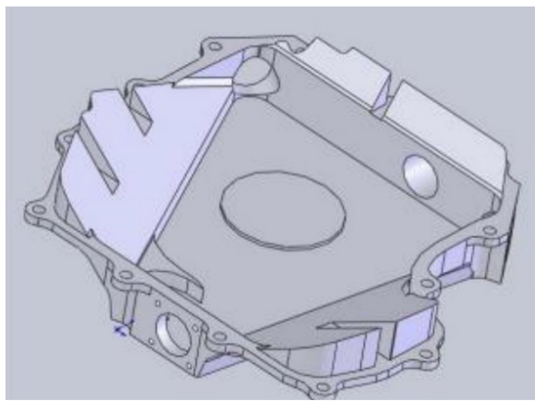


Figure 161: Concept for 40 mm flat wet sump pan

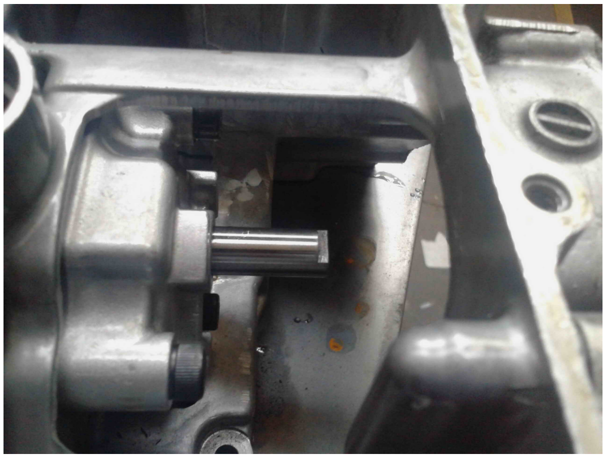
However, compared with the 16 mm dry sump implemented in 2013, a 40 mm flat sump still significantly increases the COG of the vehicle, leading to poorer dynamic performance.

3. Flat sump with calculated height
Compared with the 40 mm wet sump pan, a reduced height sump pan is desirable. The calculation should be based on the oil volume required by the CBR 600 RR engine, besides, further reduction of the sump height may lead to air scavenging issues. The oil pick up port, located at the centre point of the sump, is more susceptible to oil sloshing, hence a supplementary system may be required to prevent air scavenge issues.

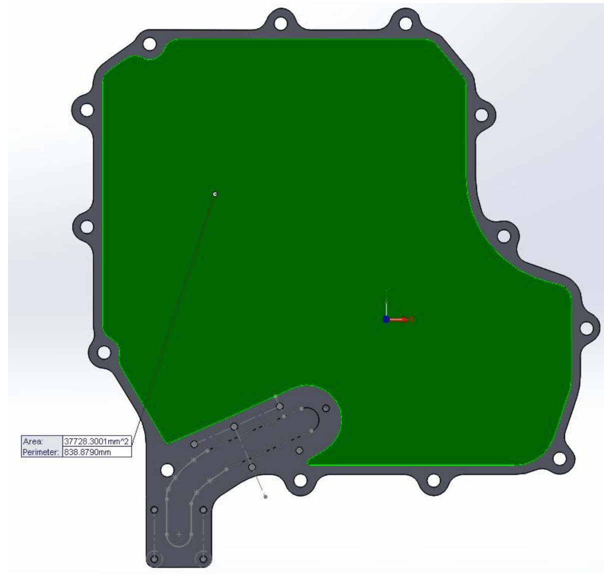
A sump pan thinner than 35 mm will not allow the installation of the Honda stock PRV. A proper way of regulating the engine block internal pressure must be designed.

G.3 Volume calculation for the wet sump pan

The Honda CBR 600 RR manual has specified the oil volume required for the lubrication system to be approximately 2.5 litres. This indicates the wet sump iteration should have the capability to contain at least 2.5 litres of oil.



(a) Highest point that oil level can arrive



(b) Sump pan

To satisfy the volume requirement, it is decided that with a shallow sump pan, some of the oil is going to be contained at the bottom of the engine block, hence, it is important to decide the highest possible point the oil level can reach without affecting the engine. Measurements had been done in the engine block and it is decided that the highest level of oil is below the internal oil pump shaft. This is shown in figure 162a. This point is 41 mm higher than the engine block bottom. From the engine bottom profile graph, the section area of the engine bottom is estimated to be 40000 mm^2. Also, from the wet sump pan CAD, with thickness below 35 mm and an external PRV channel, the oil containing area of the sump is estimated to be 37228 mm^2 (this is shown in figure 162b). Hence the oil containing height can be calculated:

The required oil volume:
$2.5L = 2.5 \times 10^6 mm^3$
The oil volume contained by the enigne block
$41mm \times 40000mm^2 = 1.64 \times 10^6 mm^3$
The required volume for the wet sump pan
$(2.5 - 1.64) \times 10^6 mm^3 = 0.86 \times 10^6 mm^3$
The required height of the sump oil pocket
$(0.86 \times 10^6)/37228 = 23.10mm \approx 23mm$

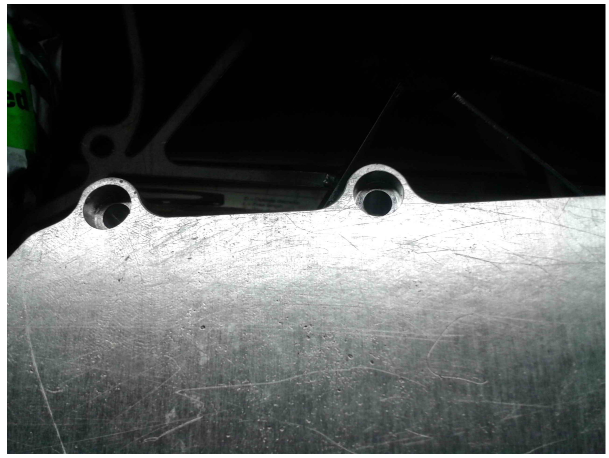
Thus, from this result, with a 23 mm oil pocket pool, plus 2 mm material at the bottom, the overall height of the oil sump is determined to be 25 mm.

G.4 Details for the wet sump pan iteration

The wet sump iteration consists of a 4 mm top plate and a 21 mm bottom plate. The test top plate was laser cut from 4 mm steel with a triangular net which enables baffles to be welded on with a flexible configuration. The final iteration of the top plate will be 4 mm aluminium with the best baffle configuration justified by the testing data. The top plate is sealed to the engine block bottom surface with some laser cut paper gasket.

The bottom plate was CNC machined from a 6065 aluminium plate of 30 mm thickness. It is designed to allow for the M6 bolts, that hold the pan to the engine block, to be counter bored so that the bolt heads remain flush with the bottom surface of the pan. The top plate and the bottom plate use M4 taper bolts and they are sealed by paper gaskets. For detail of the sump pan iteration refer to figure 163a and figure 163b.

(a) Wet sump pan iteration



(b) Counter bore at the bottom plate

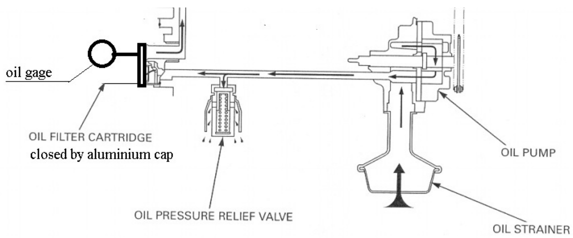
G.5 Details for the internal pressure relieve valve and PRV testing rig

The concept of an internal pressure relief valve was proposed in 2013 and a bronze internal housing was machined. The idea is the PRV is inserted into the PRV port of the engine block, with a spring of specific stiffness plugged into the housing on one end and sitting on the sump pan on the other. The relief pressure is set at 5 bar which is decided by the spring stiffness and length, as well as the the housing top area, since the housing top area has already been decided by the diameter of the pressure relief port of the engine block. The spring stiffness and length has become two tunable parameters.

(a) Custom design internal PRV

(b) Internal PRV plug into the engine block

The 2013 designed internal PRV used a spring with 27.70 mm length and this was tested in a 2014 custom designed testing rig. A waste engine block was used with part of the main oil gallery used for pressure simulating. A aluminium cap with thread was machined to closed the oil gallery at the oil filter. Also, thread was tapped on the cap to screw a oil pressure gauge in. The internal oil pump is driven by a power drill, which can reach a maximum of 3000 rpm. The schematic and the prototype of the testing rig is shown in figure 165a and figure 165b.



(a) The schematic for the PRV testing rig

(b) The prototype for PRV testing rig

The testing results for the internal PRV indicate that under small flow rates the internal pressure relief valve can regulate pressure in the oil gallery, but under high flow rates, the internal PRV can not regulate internal pressure. One of the possible improvements in the design is shown in figure 166. This design adds an external cage outside the PRV housing to guide the motion of the housing. This will allow a spring with smaller stiffness to be used. Hence the flow rate of the internal PRV is increased. This design could potentially solve the flow rate issues.

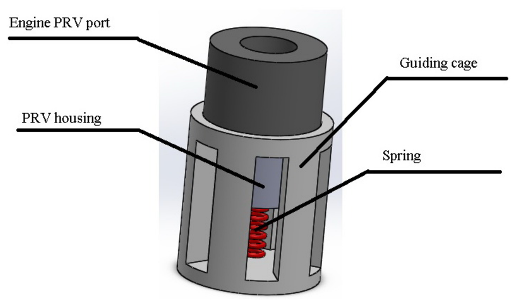


Figure 166: Concept design for improvement internal PRV

G.6 Discussion about wet sump supplementary system concept

Since a 25 mm shallow flat wet sump is designed, with a relatively small oil containing volume, a supplementary system should be designed to resist the oil sloshing and prevent the internal oil pump from picking up air. At the early stage of the design, several concepts had been evaluated.

Concept 1. Swing arm pick up
The concept for the rotating swing arm is to attach a bearing to the pick up port of the engine block. Its rotation direction depends on the centrifugal force of the car, hence the pick up port will always fill with oil when the car is turning, accelerating or braking.

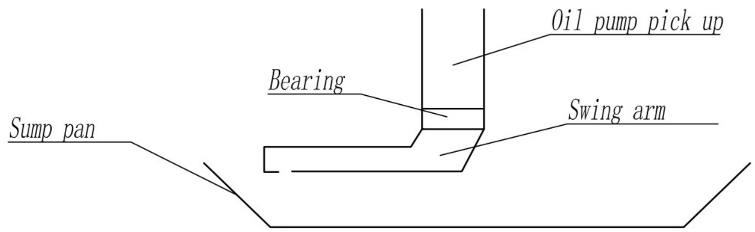


Figure 167: Concept evaluation for the wet sump supplementary system

The difficulties of this concept is ensuring good sealing between the rotating component and the engine pick up port. The mechanical component also largely decreases the oil containing volume in the wet sump. With a moving component in a 25 mm space, it is difficult to ensure reliability of the system.

Concept 2. Optimise baffles
The concept for the optimised baffles would be to have baffles parallel with the bottom surface of the flat wet sump. The baffle surface would be higher than the oil pick up port, limiting where the oil can go, allowing the pick up to have a constant supply of oil.

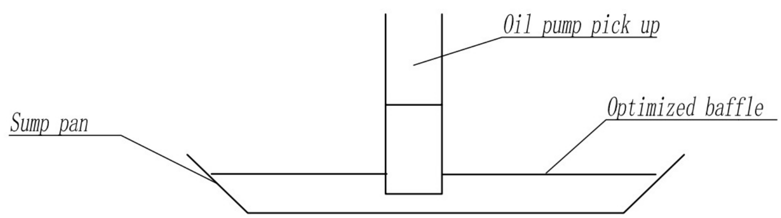


Figure 168: Concept evaluation for the wet sump supplementary system

This solution will still be susceptible to large lateral G-force situations, when all the oil sloshes to one side.

Concept 3. Active pick up mechanism
The concept is to create a mechanism with rectangular sectioned tube with an active pick up inside the tube. The centrifugal force will drive the movement of the pick up inside the tube and open the port. The tube extends to different corners of the sump, hence, the port will always fill with oil under centrifugal force in different directions.

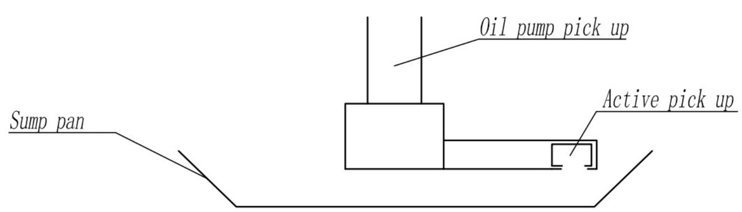


Figure 169: Concept evaluation for the wet sump supplementary system

However the internal oil pump will create large negative pressure when picking up oil, which makes centrifugal force impossible to drive the internal pick up in opening the ports. This has made the concept not feasible.

G.7 Details for hydraulic accumulator working principle

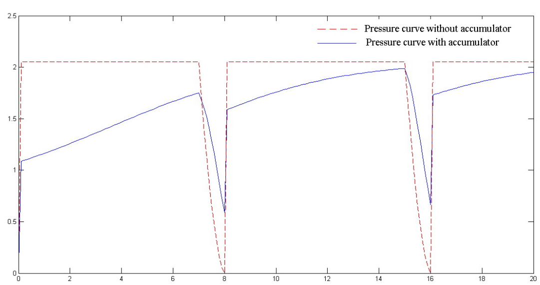
The hydraulic accumulator consists of a specially designed tube and a piston that separates the oil from the air within the tube. As oil enters one side of the accumulator, under pressure the piston is pushed back, compressing the air in the other side of the tube. The piston will always move toward equilibrium where the pressure in the air side equals the pressure of the oil side. The accumulator holds the oil in the oil side under the engine's operating pressure and can supply the engine with that oil if the normal supply is interrupted. It releases the stored oil when a pressure loss in the engine puts the piston out of equilibrium and causes the oil to exit the accumulator in an attempt to regain equilibrium.

The operating point for pressure and volume satisfies the Boyle's law which derives as:

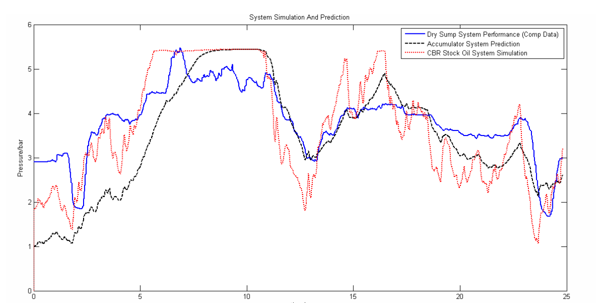
$$P_n V_n = P_g V_g = P_o V_o = C$$

Where:
P_n :The operating pressure for the oil chamber
V_n :The operating volume for the oil chamber
P_g :The operating pressure for the gas chamber
V_g :The operating volume for the gas chamber
P_o :The pre-charge pressure for the accumulator gas chamber
V_o :The pre-charge volume for the accumulator gas chamber, defined as full volume of the accumulator

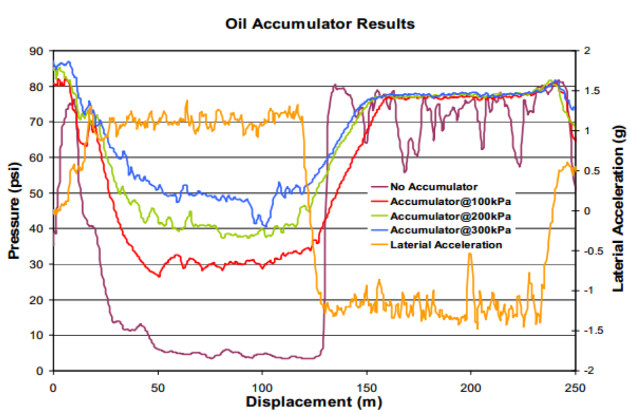
Based on the characteristic equation of the hydraulic accumulator, the internal pump parameter and the pressure data from the FSAE 2013 competition, a prediction is made for both normal and extreme pressure dropping situations.



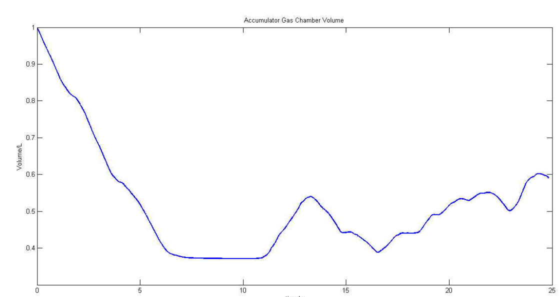
(a) Performance comparison under oil starvation situation



(b) Comparison for different system based on competition data



(c) Experimental data of hyraulic accumulator on different pre-charge pressure



(d) Volume prediction of gas chamber of hydraulic accumulator based on competition data

The prediction of the extreme (oil starvation) case is shown in figure 170a. It indicates that the hydraulic accumulator will slow down the pressure drop rate, and enable the engine to recover once the pick up port is filled with oil again.The performance comparison for different lubrication systems based on the competition data is displayed in figure 170b. It shows that with the hydraulic accumulator, the oil pressure shows less fluctuation. The volume prediction in figure 170d shows that the volume of the hydraulic accumulator will fluctuate around a certain level. The experimental data for the hydraulic accumulator on different pre-charge pressure from Monash Motorsports race is shown in figure 170c.

G.8 Details for flat dry sump system design

G.8.1 Details for flat sump pan design

Appropriate pickup locations allow you to reduce the length of oil lines and simplifies packaging. The main design consideration when deciding on pickup locations is to identify the positions of the scavenge pump and oil tank. Since the auxiliary power output shaft that powers the scavenge pump is on the left side of the engine, the oil tank is placed on the left side. Then the most appropriate location for the supply pickup is the left side of the pan. The supply pickup was designed to be short enough so as to not clash with the bottom of the alternator cover and to allow for Speedflow fittings to fit without any clashes with the engine. As for pressure relieving, a similar pressure channel as in figure 33a is designed on the flat dry sump pan. A Aviaid external pressure relief valve will be installed to regulate the internal pressure of the engine block. The top and bottom plate of the dry sump pan is connected with M4 taped bolts.

G.8.2 Oil tank design

The oil tank stores the oil for the dry sump system. The major change for the oil tank will be the reduction of the volume, from 2.5L to 2.0L. The main reason for this is the increase in sump volume.

Three inlets are designed on the oil tank. The top inlet is designed for connecting the scavenge pump outlet, which requires oil entering the oil tank with relatively small perturbations and sloshing (to prevent oil becoming aerated). The second is the pressure relief outlet and third is the internal oil pump channel inlet. The oil tank iteration is shown in figure 171

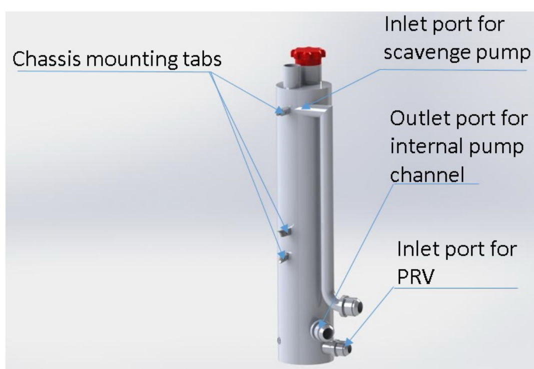


Figure 171: Oiltank iteration

The upper baffle plate is designed to slow the vortex motion of the oil and allow the oil to separate from air as it flows through the various holes on the plate. A solid lower baffle reduces the perturbations of the oil falling from the top of the tank and baffles the oils movement during longitudinal and lateral acceleration.

G.8.3 Pressure relief and scavenge pump

A similar pressure relief configuration to the wet sump system is designed for the dry sump system. An Aviaid pressure relief valve, which is shown in figure 33b is also selected as an external PRV for the dry sump system.

A Dailey two-stage scavenge pump is used for the dry sump system. Two adaptor plates were designed to mate the scavenge body with the engine block as well as the mechanical water pump. A shaft is machined to drive both the water pump and the scavenge pump. The scavenge pump assembly is shown in figure 43b.

H Recommendations

H.1 Intake

- It highly recommended that a more in depth analysis be conducted with 1D/3D coupling. Hopefully, next year, dyno access is given early so students can test and iterate through designs. Efforts need to be focused on boundary layer formation and flow separation.
- Flow Bench Validation
- If Nylon is not utilised this year, it should be investigated further as a viable means to manufacture the manifold. It possess mechanical properties and should reduce the overall weight of the system.
- Investigate internal ribs to direct flow.
- Look at manufacturing a carbon fibre intake manifold. This may require some de-featuring of some small parts to make this viable. It maybe a good idea to practice laying up early.
- Analyse CFD results on the restrictor and Bell-mouth. It maybe worthwhile designing more efficient shapes and machining them at Holmsglen.
- Investigate different injector port angles and its effect on atomisation.

H.2 Fuel

- Changing the angle of the injector can have a huge effect on fuel atomisation. It would be useful to test the differences on the dyno to see the effect on torque and power output.
- Look at possibly getting new fuel injectors. Increasing the flow and pulse of the injectors could have some benefit. Further investigating different spray patterns would be useful and if possible obtain a swirl type injector.
- If not done this year, introduce pulse width modulation (PWM) to achieve a specific fuel pressure. This will reduce the electrical load of the system. This eliminates heaat generated from the fuel pressure regulator.
- CFD sloshing of the fuel tank. Some attempts were made this year however more time was spent on the manifold. The main issue is trying to create an iso-surface to move throughout the fuel tank. The transient simulation has already been set up so this should reduce the amount of work.
- MAP-referenced fuel rail pressure (rather than referenced to atmospheric pressure) would provide better resolution in the low load portion of the fuel map. This however requires a vacuum line to the plenum.

H.3 Exhaust

- Routing the exhaust into the wake of the car should be considered in conjunction with the vehicle aerodynamics.
- If the dimensions of the chassis remain similar in future, perhaps look into a 4-1 or a twin exhaust as an FSAE rule change dictates that no header wrap can be used from next year onwards. This means more clearance is required between components, particularly the firewall.
- Attempt to gain dynamomometer access early so as to test the influence of various parameters on muffler performance.
- Investigate manufacture of an optimal tail pipe for performance gains before finalising other exhaust dimensions.

H.4 Cooling

- Look to use CFD earlier in the design process to minimise the size of the radiator as much as possible.
- Look into the use of two smaller radiators rather than the one large one. The bigger the radiator, the larger the inlet duct needs to be, hence, the more drag it will induce on the car.
- Recommend more input into inlet duct design with chassis team. Cooling requires more air flow but at the expense of drag which chassis and the aerodynamics team want to reduce.
- Look into other radiator positions rather than the conventional placement in the side-pod. May be able to place it in a position where the free stream air is sufficient.
- CFD should not be viewed as a substitute to on-track testing. Try gaining wind tunnel access or have certain aspects to the design finished early so that some testing can be gained on the 2014 vehicle.

H.5 Lubrication

- It is recommended that a precise 1-D hydraulic circuit model be construed for the CBR 600 RR engine lubrication system, this include the model for the oil galleries, internal orifice, pump as well as sliding bearing for various shaft inside the engine block. One dimensional calculation software such as GT-Power, simulink could be utilised. This model will allow the internal oil pressure being estimated based on the rpm data.
- It is suggested that a 3-D model for oil sloshing inside the oil sump be constructed based on CFD techniques. This model will allow the affect of lateral and longitudinal G-force to the oil pressure being estimated, besides baffle design could also based on the model estimation.
- Further investigation about the feasibility of the internal pressure relieve valve, improve the design to fit the flow rate requirement, this will allow a better packaging comparing with the external PRV configuration.
- Investigate the possibility to reduce the height of the wet sump which can lower the COG of the car to improve the dynamic performance of the car.
- For dry sump system, investigate the possibility to package the entire lubrication system inside the engine block, this include an internal pressure relieve valve and an internal scavenge pump.

I Scope of Works

THE UNIVERSITY OF MELBOURNE

Capstone Project

Department of Mechanical Engineering

Scope of Works

2014

Project Title: Formula SAE – Engine

Date: 24 March 2014

Project Team Information

Identifier: **CP-YYA-142**

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Academic Supervisor: Dr Yi Yang

Academic Examiner: TBD

Version: 3, 23/03/2014

1. Project

As a yearly project, MUR 2014 will seek to improve upon the car produced in 2013, to perform better in the Australasia FSAE competition in December. The engine team will focus on optimizing and refining the performance, efficiency and reliability of the engine. The main areas of interest of the engine sub-team will be intake, exhaust, tuning/testing, lubrication, cooling, and validation of assumptions. These will be tested with simulation and design tools as well as validation with dynamometer testing. Design and manufacturing specifications will adhere to the 2014 FSAE rules and regulations.

2. Objectives

The overall objective for the engine team for 2014 is to improve the performance and fuel efficiency of the MUR 2013 engine system, while ensuring reliability is not sacrificed. Our design philosophy is as follows:

- Reliability
- Dynamic performance
- Drivability
- Ease of manufacture
- Validation of all assumptions throughout the design process
- Flexible and adjustable
- Better facilitate transfer of knowledge from year to year
- Present ideas well for design event at FSAE competition

3. Definition of starting point

Based on experience from past years and delays in initial design period, we will continue to use an engine block from a Honda CBR600RR motorcycle. We will design and manufacture

- An intake system to deliver air/fuel mixture to the engine cylinders
- An exhaust system to scavenge combustion products from the engine cylinders
- A lubrication system to reduce friction wear between moving components
- A cooling system to remove waste heat from the engine block
- A fuel system to deliver fuel to the intake system and hence the engine

We will also tune the engine for optimal performance and efficiency, using a MOTEC M400 Engine Control Unit (ECU), which will be conducted on the dynometer.

We have a large collection of past MUR reports to use for research and design, as well as a vast library of international FSAE knowledge stored on the internet in papers, forums, blogs, team websites, and more. Past team members are also an excellent resource, and are very generous with their time and knowledge. The appendix features a preliminary literature review on material that was considered in the early stages of design.

4. Task descriptions

4.1 Engine Block

Although the engine block is not designed, all our systems are dictated by it. As a team we will attempt to increase the compression ratio of an engine and repair the 2013 primary engine. We will work in conjunction with other sub teams to increase engine change time for competition.

4.2 Intake

For 2014, we will be using 2013's design as a starting point. As a team we believe lasts years system was excellent, however we need to quantify some performance variables for this system first. This year we will adjust the performance to accommodate an aerodynamics package, reduce weight and increase flexibility. Given a limited budget we will be using the same throttle body from AT Power. We will use a rapid prototyped conical spline plenum as a mould for a carbon fibre intake; which with the smoother surface will ideally increase evenness of flow distribution to the four in-line cylinders. Carbon fibre runners will also allow us to create a smoother contoured path for airflow in comparison to the acetone vapour treated plenum of 2013 and prevent backfire.

There is strong emphasis to use CFX & Fluent to model the intake system and couple this with GT Power to gain maximum performance. Since the car was not lacking performance, lengths will remain the same, with shape being altered to adjust the flow separation and cylinder to cylinder air stealing and hence increase volumetric efficiency.

4.3 Exhaust

Our team believes that the 2013 exhaust system was of a very high standard and would more than adequately satisfy our team goals for 2014. As a result, we are going to maintain the Tri-Y collector system such that we continue to optimize for mid ranged RPM performance. With the overall car changing vastly, such that we are expected to operate within this low-mid range even more so; it is our aim to optimize last year’s system.
Obtaining theoretical values for geometries is extremely scarce when designing exhaust systems, so similar to last year, basic flow equations will be implemented as a starting point. These will then be iterated through via the use of GT-Power to provide an optimum solution for the desired RPM range.

Additionally, we will attempt to implement a straight through muffler which will improve engine scavenging and overall performance. The main restriction of the exhaust will be the routing of the exhaust with additional chassis constraints, such as outriggers and overall shape. We will attempt to route it in a similar way to 2013, however, we will also be conscious to potentially routing the exhaust into the wake of the car such to enhance scavenging in the cylinders.

As our budgets are extremely tight and with left over piping from 2013, the material will be maintained and as an added bonus will allow us to maintain a low weight.

4.4 Simulation/Tuning

The engine system will be simulated in GT-Power prior to manufacture, for increased ability to optimize without prototyping and testing. This year we will use coupled simulation with GT-Power and ANSYS Fluent and CFX for more accurate flow simulation through the complex 3D intake system. The 2013 team produced an excellent GT-Power model which we will adjust using 2014 numbers. However this model does not accurately predict at RPM below 4000 and hence a conscious effort must be placed to solve this issue.

We will tune our engine to have a narrow torque band this year, to take advantage of our more experienced drivers, as well as new ECU features such as launch control. This should improve acceleration, provided we can train our drivers to shift gears at the right RPM.

4.5 Lubrication

This year we are aiming to design a lubrication system which is reliable and easier to package. A wet sump lubrication system with a hydraulic accumulator has been researched and investigated for this purpose. The new sump design will ensure the car will retain a low centre of gravity (COG) based on system reliability. Components will be designed and tested in the 2013 car. If this system design proves to be successful, this should allow us to simplify oil lines in the 2013 lubrication system, which can reduce the complexity of maintenance and the risk of leaking to a great extent. The new system may also reduce the weight of the car. An Accusump accumulator will have to be purchased as well as some cheap aluminium so we are able to machine a variety of sumps with different baffles.

We will also optimise the 2013 system so that we have a contingency plan. We will need to make sure if the 2013 system is utilised it is capable of withstanding the increased lateral forces and acceleration.

We will test the use of an internal pressure relief valve, which would reduce system weight and complexity of the sump design. If time permits, we will investigate the oil volume in the system to ensure the reliability of the system, which may provide data for the following year to reduce the thickness of the wet sump.

4.6 Cooling

The main challenges facing the cooling system this year will involve packaging constraints and the introduction of a full aerodynamic package. The chassis this year will feature outriggers to increase torsional rigidity, meaning the radiator placement within the sidepod would need to be considered and perhaps altered to fit within the constraints of the car; as well as deal with any potential disturbances to air flow coming in. These disturbances may be elevated by the addition of an aerodynamic kit, particularly the presence of an undertray and a front wing. The cooling system will need to achieve its objectives despite these disturbances. Hence, experimental testing will be undertaken to see the effect on radiator performance with restricted air flow. Thermodynamic calculation indicates the 2013 radiator was overdesigned and may be optimised to help reduce mass and improve packaging. However, due to the different conditions the 2014 car is expected to encounter, appropriate performance analysis of the 2013 radiator is required to justify the purchase of a new one, especially considering the limited budget the team is expected to work within. CFD testing using a software package such as ANSYS Fluent will be used to aid analysis. The radiator will again be aided by the fan and shroud as was the case last year, this time with more emphasis on the radiator being fully sealed in the sidepod so as to improve cooling efficiency. The stock mechanical water pump will continue to be used as well as the small electrical water pump that switches on to ensure cooling is maintained when the engine is switched off. The addition of water flow rate sensors will provide a more accurate reflection of pump performance as well as indicate occasions where pump maintenance may be required.

4.7 Fuel

In FSAE, teams have the choice of using E85 (85% Ethanol) or RON98 (98 Octane petrol) as fuel. We will continue to use E85 as fuel this year, as its high knock resistance makes it safe and easy to tune, it generally yields cooler exhaust gases which may enable a lighter exhaust system and has various thermodynamic properties that we might be able to exploit. We will also investigate smaller, lighter fuel pumps, and a fuel level indicator, which would give real time feedback on the amount of fuel in the tank and has been recommended from 2013. We would also like to test the specific fuel consumption at each lambda value and adjust the fuel MAP for each event.

We will also look at pulse width modulation and the viability of using closed loop lambda to maintain fuel efficiency for the endurance events. The fuel pump and fuel rail will remain the same.

5. Duration of tasks

The following is some key dates and a rough summary of the duration for each task. More information can be found on the attached file of the Gantt chart.

Key dates:
1st CAD iteration due to Integration – 28/03/14
2nd CAD iteration due to Integration – 11/04/14
3rd CAD iteration due to Integration – 09/05/14
Final CAD submissions – 18/05/14
Final Design Freeze – 19/05/14
Progress Report #1 due – 26/05/14
Tabs welded on chassis – Early July
Run Engine – 08/08/14
Progress Report #2 due – 11/08/14
Shakedown – 05/09/14
Final Report due – 06/10/14

5.1 Engine Block

Simulate Engine Chamber - 4 weeks (If times permits)
Fix engine (Amber) - 2 Months
Get Higher Compression Engine up and Running - 3 Months

5.2 Intake

Model new intake in GT-Power/Fluent – 2 months
Test and verify 2013's Design - 2 Months
Manufacture intake system – 1 month
Run intake on dyno – Ongoing throughout the year

5.3 Exhaust

Preliminary Calculations - 2 weeks
Modelling and Optimization using GT-Power - 4 weeks
Design of Jig - 2 Weeks
Manufacturing - 4 weeks
Dyno Testing – Ongoing throughout the year

5.4 Simulation/Tuning

Model engine system in GT-Power – 2 weeks
Testing on dyno – Ongoing throughout the year
Tuning on dyno – Ongoing throughout the year

5.5 Lubrication

Wet sump system research – 1 week
Dry sump system research – 1 week

2013 Lubrication system review – 1 week
2013 oil pressure analysis – 0.5 week
System design and simulation – 4 weeks
Flat sump CAD design and analysis – 2 weeks
Component design – 8 weeks

5.6 Cooling

2013 Cooling System analysis – 2 weeks
Radiator Calculations – 1 week
Radiator Performance analysis – 8 weeks
Radiator Design – 2 weeks
Pump testing and maintenance – 4 weeks
Electrical water pump investigation - 5 weeks
Component and mounting design and manufacture – 8 weeks
Routing and assembly – 2 weeks

5.7 Fuel

Design new fuel tank – 5 week
Design and simulate Swirl Ports – 2 weeks
Manufacture fuel tank – 3 weeks

6. End point

By the competition we would like to have produced a reliable system that achieves both high performance and fuel efficiency, while increasing reliability and reducing weight. This will be indicated by our performance at the FSAE competition. Major changes that should be fully operational and tested include

- Carbon fibre intake
- Optimised Fuel System
- 4-2-1 Exhaust with straight-through muffler
- Refined dry sump and wet sump with accumulator
- Fan shroud
- Improved GT-Power/Fluent model

We will also compile a transfer of knowledge database so that we are able to best provide a foundation of the 2015 team. This will hopefully help push the University of Melbourne to be extremely competitive every year.

Appendix:

Literature Review:

Intake:

Intake performance is determined by several factors: cylinder-to-cylinder volumetric efficiency, time of choked flow in the restrictor, total pressure loss along the restrictor, sound spectrum frequency content, and physical packaging characteristics. This means we need good air distribution to each cylinder, low flow resistance, tuning effects, fuel atomisation, and standard requirements such as low weight, reliability and drivability.

In order to grasp the design aspect of Formula SAE it was decided that an excellent way to begin is to learn the basic of a spark ignition engine and the fundamental theory associated with designing a fuel and intake system. [2] provided elementary knowledge with do’s and don’ts without providing means to do so.

Heywood [1], provided an in depth look at what is happening inside the IC engine. It gives a good development of theory and practice whilst laying a foundation of thermodynamic principals and other relevant theories. Heywood suggests looking at implementing fuel upstream through the fuel injectors to ensure better mixing however this could have other adverse effects and hence testing must be done. Through understanding how working fluids in the engine behave, we are able to create simulations that accurately replicate real life conditions. Heywood also suggests using certain software created my GM for simulations. [1] also suggests parameters that affect performance such and MEP, spark timing, chamber design and mixture composition and what methods can be used for testing.

[4] and [5] will provide the necessary background in testing and tuning of engines, so that we are able to meet the Engine 2014 team goals. That is to implement varying tunning options for different dynamic events for competition

[3] was the foundation where concept selection was done. Claywell and others do comprehensive research into various intake design and reflect on various characteristics. They conclude that coupling 1D and 3D CFD will benefit students the most and more accurately predict the performance of varying intake designs. A conical design also showed the best improvements and easiest to tune. However there are packaging restraints.

If an optimised intake is produced by the end of first semester, an active intake will be looked at in order to improve and flatten the torque curve. This will provide peak torque for various RPM. [6] will provide an introduction into the problem.
[7] and [8] will provide guidance in how to design and the steps to take.
There are various short SAE articles that relate to each component of the intake system. These articles will be read to aid in design, testing, and manufacturing.

Lubrication:

The lubrication system is one of the fundamental sub-parts of an engine with two main functions. Firstly the lubrication system delivers lubrication oil to moving parts in the engine crank case including the crank shaft and pistons, which can reduce the wear and friction caused by the relative motion. Meanwhile, the flowing lubrication oil takes away some of the heat from parts and components in the crank case. This is known as the cooling effect of the lubrication system.

In order to get some basic idea and understanding of the mechanism of the engine, [1] and [2] was put in the reading list as entry reading material of engine working principle.

The overall target for lubrication system design is to ensure the oil pressure is maintained in the lubrication oil circuit. In other words, ensure there is no engine oil starvation in the crank case for any situation when the car is running.

From [10] it is shown that two kinds of lubrication systems are implemented in most of the vehicles including the Formula SAE racing car i.e. the wet sump system and the dry sump system.

The wet sump system is comprised simply of a sump pan underneath the crank case, which is also used as the engine oil container. The sump itself, the stock oil pump in the crank case and a pressure relieve valve constitutes the inner engine lubrication system. The main advantage of wet sump system is its low weight with less components, simplicity and good packing with the crankcase. However, the drawback of wet sump is also significant, including the main issue of oil sloshing in the pan in extreme situation like acceleration and turning which may cause the oil starvation. The higher COG has also caused lower dynamic performance of the racing car.

The dry sump system is designed by replacing the oil pan container by a dry oil sump which does not contain any oil. It is kept dry by sucking oil by an external scavenge pump. Circuit is designed by transferring the engine oil into an external oil tank. The advantage of dry sump system is obvious as it eliminates the oil sloshing issue and lowers down the COG. However, the negative points are the extra component which increases the overall weight compared with wet sump system.

By reading and comparing [11] [12] [13] and considering the fact catastrophic engine failure occurred in the 2012 competition, the 2012 designed dry oil sump was re-designed in 2013 with a more reasonable pick-up place and layout.

Besides, the comparison between [10] [11] [12] [13] [14] actually provides a more clear understanding of the design of the dry oil sump. [15] gives more solutions to build an engine lubrication system not only including the dry sump system but also a concept design of a modified wet sump system to deal with the oil sloshing issue.

Cooling:

The function of an engine’s cooling system is to maintain engine temperatures that are optimal for engine performance. Approximately a third of the energy the engine generates needs to be dissipated to the surroundings by some form of heat transfer. There are multiple components involved in the cooling system which allow for the removal of this heat. These include the radiator, the pumps which supply a coolant (water) throughout the system, the cooling fan and the ducting from the side pod of the car to the radiator.

In order to gain an understanding of the cooling system one must first have a sound knowledge in how internal combustion engines function. [2] provides the fundamental understanding required including naming the various components of the engine and what each component does. It also introduces the various mechanisms that cool the engine at the various stages of its operation. Heywood [1], delves further into engine theory as well as bringing a more quantitative analysis of the cooling system which includes a thorough explanation of the thermodynamic principals involved.

[16] is a technical paper that investigates a manner in which the quality of airflow for cooling can be improved without radical changes to the car's body kit. Through the use of computational fluid dynamics, the paper highlights the problem areas of the body kit, and shows how minor changes can improve the cooling performance of the car without sacrificing aerodynamic qualities. [17] also takes a look at how cooling air flow can aid the car's performance, targeting the issue of the radiator's location behind the front wheel and how that influences the handling of the car.
[18] is another technical paper about a study conducted on a small engine cooling system and how a single electrical water pump can not only produce flow rates required for the system but also how it can increase the output of the engine. This could be particular useful as the 2013 car uses both a mechanical pump for when the engine is running and a small electrical booster pump that turns on once the engine is off.
The cooling system is an often neglected aspect when considering engine performance. [19] is devoted entirely to an engine's cooling system, emphasising the importance a good cooling system can have on power production, fuel consumption and the overall reliability and durability of the engine.

Exhaust:
The exhaust is an integral part of the overall engine system and needs to be designed and manufactured well in order to provide a quality engine. There are several factors which affect the efficiency and effectiveness of an exhaust system, certain parameters such as valve sizes, compression ratios, engine speeds are some of these factors of how efficiently combustion products are dispensed of. Additionally, the exhaust is linked closely with the intake system and needs to be tuned accordingly for maximum optimisation.

Before investigating on any specifics it is important to understand the foundations of IC engines and the basics of each component of the exhaust as well as the engine itself before any design strategies could be implemented. Willard [2] has delivered the basic knowledge of how the exhaust works and how it is linked with the intake system as well as given initial understanding of certain things that can be targeted to increase performance within the engine.

[4] Also tells of exhaust backflow/reverse flow due to intake and exhaust valve overlapping and how higher compression ratios may be able to combat this. Whilst increasing the compression ratio will allow for greater performance, 'knocking' (spontaneous ignition) may be induced so appropriate testing will need to be conducted to ensure proper engine function.

As previously mentioned, in order for our systems to do well they must be tuned accordingly and resources [4] and [5] will most likely be used for this.

[20] Will be used to consider the manufacturing of a muffler, despite the current muffler having exceeded expectations in limiting the noise output.

It is apparent that the pressure waves within the exhaust play a vital role in peak engine performance and being able to identify the resonance of significant waves will allow for greater performance. [21] and [22] Will provide the understanding on how to design the specific geometries to ensure effective scavenging and good cylinder charging

Through scouting various teams at competition it was found that the Canterbury team whose car was quite impressive for a first time entrant utilized the works of Pashley [20] and I believe it would be quite handy to browse through his text. Additionally, looking at the software ‘Pipemax’ may also be of value to estimate geometries of the exhaust.

Furthermore, previous year’s final reports may be used (in particular 2013) along with FSAE articles that are relevant as somewhat of a stencil in order to maintain our goal to improve the existing system.

[1] J. B. Heywood, Internal Combustion Engine Fundamentals, New York: McGraw-Hill, 1988

[2] Willard W. Pulkrabek Engineering Fundamentals of the Internal Combustion Engine, New Jersey Pearson Prentice Hall

[3] M. Claywell, D. Horkheimer, G. Stockburgher, Investigation of Intake Concepts for Formula SAE, University of Minnesota- Twin Cities

[4] A. Graham Bell, Performance Tuning in Theory and Practice Four Strokes, Haynes Publishing Group

[5] J. Martyr, M. A Plint , Engine Testing , Elsevier Ltd

[6] M.A. Ceviz, M. Akin Design of a new SI engine intake manifold with variable length plenum, Elsevier, 2010

[7] P. Pogorevc, B. Kegl. Intake system design procedure for engines with special requirements, University of Maribor, Maribor, Slovenia

[8] Intake Manifold Design using Computational Fluid Dynamics, Matthew A. Porter, University of New South Wales at the Australian Defence Force Academy

[9] Various short SAE articles

[10] Design of a dry sump lubrication system for a Honda CBR600 F4i engine for Formula SAE applications, Massachusetts Institute of Technology, 2006

[11] Final Year Project final reports, 2011 Engine team, Mur Motorsports, The University of Melbourne, 2011

[12] Final Year Project final reports, 2012 Engine team, Mur Motorsports, The University of Melbourne, 2012

[13] Final Year Project final reports, 2013 Engine team, Mur Motorsports, The University of Melbourne, 2013

[14] FSAE Engine Dry-Sump Oiling System Design, Team MEM-06, Drexel University, 2003

[15] Lubrication system and shifting system improvement final design report, Team 2011, University of Manitoba 2011

[16] Christoffersen, L. M., Oderblom, D. and Ofdahl, L. 2008. Improving the Cooling Airflow of an Open Wheeled Race Car. (SAE 2008-01-2995)

[17] Dyverfors, N., Borre, K., Arnell, C. and Rice, J. Interaction of Downforce Generating Devices and Cooling Air Flow-A Numerical and Experimental Study on Open Wheeled Race Cars. *Training*, 2007 pp. 07—11

[18] Jawad, B., Zellner, K. and Riedel, C. 2004. Small Engine Cooling and the Electric Water Pump. *SAE International*.

[19] Bohacz, R. T. n.d. *Engine cooling systems*

[20] D. Davis, Theoretical and Experimental Investigation of Mufflers with Comments on Engine Exhaust Muffler Design, 1954

[21] P. Bush, A Design Strategy for Four Cylinder SI Automotive Exhaust Systems, SAE, 2000.

[22] P. H. Smith, Scientific Design of Exhaust and Intake Systems, 1972

J Team Administration

J.1 Budget

Budget Distribution	**$**	**Adjustments Lubrication**	**Adjustments Intake**	**Adjustments Exhaust**	**Adjustments Engine**
Intake	$300.00		$100.00		
Fuel	$350.00	-$120.00			$600.00
Engine	$915.00				$215.00
Cooling	$470.00	-$50.00			
Lubrication	$730.00				
Exhaust	$835.00	-$50.00		$185.00	
Spare	$0.00	-$150.00			
Total	$3,600.00				
Current Budget					
Intake	$131.00				
Fuel	$292.09				
Engine	$283.04				
Cooling	$160.44				
Lubrication	$51.45				
Exhaust	$227.21				
Spare	$0.00				
Total Remaining	$1,145.23				

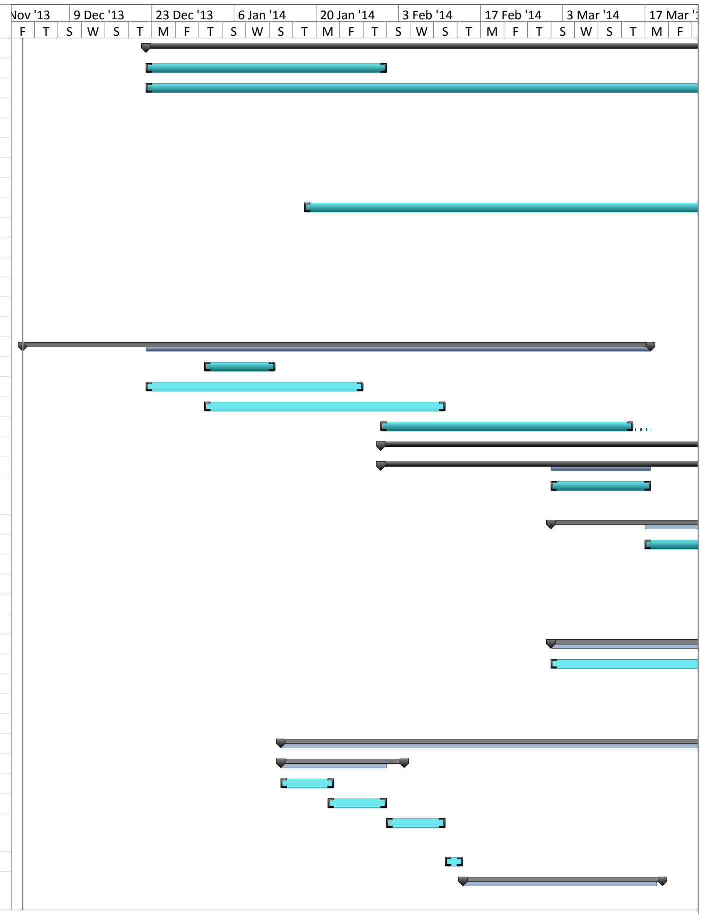
Table 32: Remaining Budget

Purchases	Intake	Fuel	Exhaust	Lubrication	Cooling	Engine	Spares
	115	$6.26	$595.79	$270.00	$29.98	$31.96	0
	54	$22.84	$12.00	$222.03	$250.00	600	0
	0	$18.86	0	$151.02	$29.58	0	0
	0	$9.95	0	$35.50		0	0

Table 33: Purchases Made by Each subteam

J.2 Gantt Chart

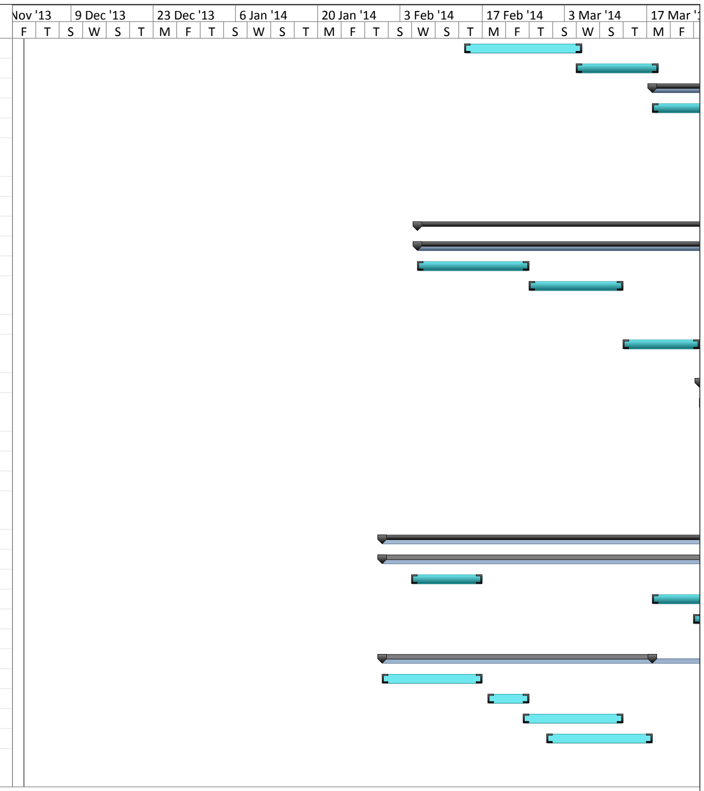
ID		Task Mode	Task Name	Duration	Start	Finish	Predecessors	Resource Names
1			**Team Timeline**	**252 days**	**Sun 22/12/13**	**Fri 5/12/14**		
2			Majority of Research Completed	31 days	Sun 22/12/13	Fri 31/01/14		
3			Design Phase	108 days	Sun 22/12/13	Mon 19/05/14		
4			1st CAD submission	0 days	Fri 28/03/14	Fri 28/03/14		
5			2nd CAD submission	0 days	Fri 11/04/14	Fri 11/04/14		
6			3rd CAD submission	0 days	Fri 9/05/14	Fri 9/05/14		
7			Component Freeze	0 days	Mon 19/05/14	Mon 19/05/14		
8			Manufacturing Phase	85 days	Mon 14/04/14	Fri 8/08/14		
9			Testing Phase	231 days	Sat 18/01/14	Fri 5/12/14		
10			Progress Report #1 due	0 days	Mon 26/05/14	Mon 26/05/14		
11			Tabs welded on chassis	0 days	Mon 7/07/14	Mon 7/07/14		
12			Run Engine on New Car	0 days	Fri 8/08/14	Fri 8/08/14		
13			Progress Report #2 due	0 days	Mon 11/08/14	Mon 11/08/14		
14			Shake Down	0 days	Fri 5/09/14	Fri 5/09/14		
15			Final Report due	0 days	Mon 6/10/14	Mon 6/10/14		
16			**Initial Appreciation (Internal)**	**79 days**	**Sun 1/12/13**	**Mon 17/03/14**		
17			Lit Review (Internal)	9 days	Wed 1/01/14	Sun 12/01/14		
18			Concept Design	27 days	Sun 22/12/13	Mon 27/01/14		
19			Final Design Concept	29 days	Wed 1/01/14	Mon 10/02/14		
20			Intake System	32 days	Fri 31/01/14	Mon 17/03/14		
21			**Intake Manifold**	**152 days**	**Fri 31/01/14**	**Sun 31/08/14**		
22			**CFD 2013 Design**	**43 days**	**Fri 31/01/14**	**Mon 31/03/14**		
23			Run Dyno for 2013 intake with varying volumes	12 days	Sat 1/03/14	Mon 17/03/14		
24			**Plenum**	**132 days**	**Sat 1/03/14**	**Sun 31/08/14**		
25			Plenum Volume Design	11 days	Mon 17/03/14	Mon 31/03/14		
26			Runner Design	11 days	Mon 31/03/14	Mon 14/04/14		
27			Restrictor Design	11 days	Mon 14/04/14	Mon 28/04/14		
28			Swirl Port Designs	11 days	Mon 28/04/14	Mon 12/05/14		
29			Print 1st Plenum	88 days	Thu 1/05/14	Sun 31/08/14		
30			**Fuel System**	**57 days**	**Sat 1/03/14**	**Mon 19/05/14**		
31			Fuel Tank Design	31 days	Sat 1/03/14	Fri 11/04/14		
32			Injector Design	37 days	Fri 28/03/14	Mon 19/05/14		
33			Fuel Pick Up Manufacture	26 days	Thu 31/07/14	Thu 4/09/14		
34			Fuel Tank Manufacture	45 days	Mon 4/08/14	Fri 3/10/14		
35			**Lubrication**	**61 days**	**Tue 14/01/14**	**Mon 7/04/14**		
36			**Lubrication System Research**	**15 days**	**Tue 14/01/14**	**Mon 3/02/14**		
37			Wet Sump System Research	7 days	Tue 14/01/14	Wed 22/01/14		
38			Dry Sump System Research	8 days	Wed 22/01/14	Fri 31/01/14		
39			2013 Lubrication System Review	7 days	Sat 1/02/14	Mon 10/02/14		
40			2013 Oil pressure analysis	3 days	Tue 11/02/14	Thu 13/02/14		
41			**Lubrication System Design and Simulation**	**25 days**	**Fri 14/02/14**	**Wed 19/03/14**		



Project: Engine_2014
Date: Mon 6/10/14

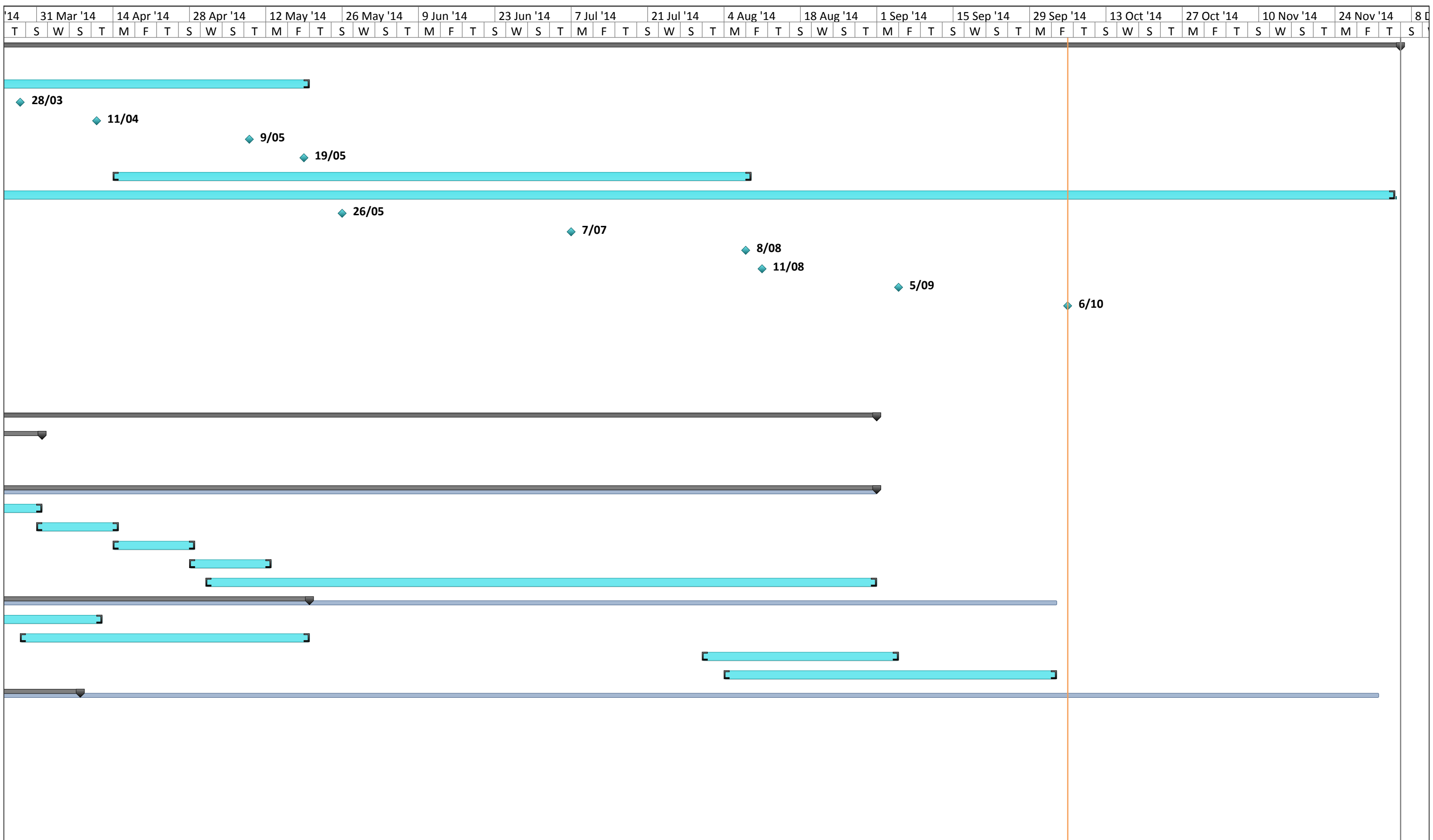
Task	Project Summary	Inactive Milestone	Manual Summary Rollup	Deadline
Split	External Tasks	Inactive Summary	Manual Summary	Progress
Milestone	External Milestone	Manual Task	Start-only	
Summary	Inactive Task	Duration-only	Finish-only	

ID		Task Mode	Task Name	Duration	Start	Finish	Predecessors	Resource Names
42			System Design	15 days	Fri 14/02/14	Wed 5/03/14		
43			System Simulation	10 days	Wed 5/03/14	Tue 18/03/14		
44			**CAD Modelling & Analysis**	**15 days**	**Tue 18/03/14**	**Mon 7/04/14**		
45			Flat sump Design and analysis	10 days	Tue 18/03/14	Mon 31/03/14		
46			Component design	40 days	Mon 31/03/14	Fri 23/05/14		
47			**Wet sump manufacture and testing**	**136 days**	**Mon 26/05/14**	**Mon 1/12/14**		
48			Manufacture all components	41 days	Mon 26/05/14	Mon 21/07/14		
49			Testing Phase	96 days	Mon 21/07/14	Mon 1/12/14		
50			**Cooling**	**174 days**	**Thu 6/02/14**	**Mon 6/10/14**		
51			**Radiator design**	**44 days**	**Thu 6/02/14**	**Mon 7/04/14**		
52			Radiator design - Calculations	13 days	Thu 6/02/14	Mon 24/02/14		
53			Radiator design - Modelling, analysis, testing	13 days	Tue 25/02/14	Wed 12/03/14		
54			CFD Radiator, Shroud, Fan	40 days	Tue 12/08/14	Sat 4/10/14		
55			Airflow into radiator: Ducting. Effect of Aero	9 days	Thu 13/03/14	Tue 25/03/14		
56			**Water Pump analysis**	**20 days**	**Wed 26/03/1**	**Tue 22/04/14**		
57			Mechanical Water Pump: take a part, conduct own test	8 days	Wed 26/03/14	Fri 4/04/14		
58			**Plan Manufacturing + Layout**	**14 days**	**Mon 14/04/1**	**Thu 1/05/14**		
59			Header tank CAD	3 days	Mon 14/04/14	Wed 16/04/14		
60			Other assembly parts	11 days	Thu 17/04/14	Thu 1/05/14		
61			Manufacture of all components	113 days	Thu 1/05/14	Mon 6/10/14		
62			**Exhaust**	**157 days**	**Fri 31/01/14**	**Fri 5/09/14**		
63			**Muffler**	**51 days**	**Fri 31/01/14**	**Thu 10/04/14**		
64			System Considerations	9 days	Wed 5/02/14	Sun 16/02/14		
65			Calculations	12 days	Tue 18/03/14	Wed 2/04/14		
66			System Modelling & Analysis	12 days	Tue 25/03/14	Wed 9/04/14		
67								
68			**Manifold**	**33 days**	**Fri 31/01/14**	**Mon 17/03/1**		
69			Design Considerations	12 days	Fri 31/01/14	Sun 16/02/14		
70			Prelim Runner Calcs	5 days	Tue 18/02/14	Mon 24/02/14		
71			Optimization of Geometry	14 days	Mon 24/02/14	Wed 12/03/14		
72			CAD Modelling & Analysis	13 days	Fri 28/02/14	Mon 17/03/14		
73			Manufacture and Weld Exhaust	60 days	Mon 16/06/14	Fri 5/09/14		



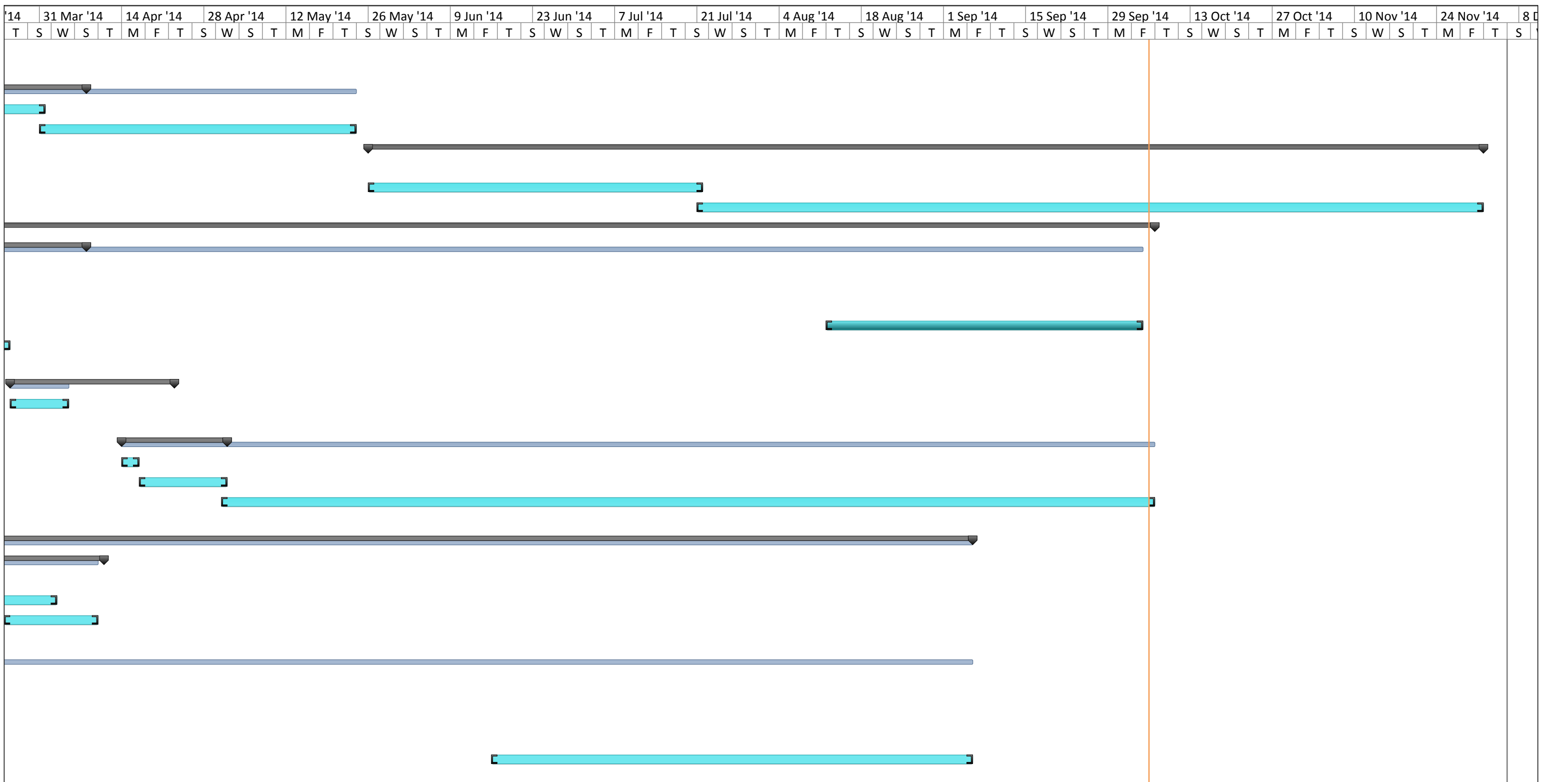
Project: Engine_2014
Date: Mon 6/10/14

Task	Project Summary	Inactive Milestone	Manual Summary Rollup	Deadline
Split	External Tasks	Inactive Summary	Manual Summary	Progress
Milestone	External Milestone	Manual Task	Start-only	
Summary	Inactive Task	Duration-only	Finish-only	



Project: Engine_2014
Date: Mon 6/10/14

Task		Project Summary		Inactive Milestone		Manual Summary Rollup		Deadline	
Split		External Tasks		Inactive Summary		Manual Summary		Progress	
Milestone		External Milestone		Manual Task		Start-only			
Summary		Inactive Task		Duration-only		Finish-only			



Project: Engine_2014 Date: Mon 6/10/14	Task		Project Summary		Inactive Milestone		Manual Summary Rollup		Deadline
	Split		External Tasks		Inactive Summary		Manual Summary		Progress
	Milestone		External Milestone		Manual Task		Start-only		
	Summary		Inactive Task		Duration-only		Finish-only		

J.3 Project Diary

03.01.14 Meeting Minutes
Team:
Rabieh: Present
Jim: Present
Aaron: Present
Dwayne: Absent

1. Lit Review
- Text books, articles that will be used or have been used and what is expected to be gathered from them.
- Materials can be found in user folders from admin log in as well

2. IA/Presentation
- Done as a group -> PowerPoint
- Each member produces their sub part

3. Concept submission
- Due 26th.
- What will be designed, how?
- Previous year's info available

4. Sensor list
- Radiator temperatures (down the line)
- Duct (pressure drop?)
- Intake related sensors
- Lubrication: at this stage may already have data required

5. MoTEC Manual Read
- Read

6. Also like to discuss how everyone is going with their individual parts
- Everyone on track except exhaust which at this stage progress unknown

09.01.14 Meeting Minutes
Team:
Rabieh: Present
Jim: Present
Aaron: Present
Dwayne: Absent
Clarke (integration): Present

1. Lit Review and Collation
- Lit Reviews done by end of Saturday for Sunday due date

2. IA to be completed with presentation
- To be done as a PowerPoint for presentation on Sunday
- Collective IA on general engine + doing our own subsections

3. Concept Design Update
- Most of the team still in research phase
- Due 26th Jan
- Designs based on your top level goals

4. Timeline
- Need to work with suspension and chassis for engine mountings and other location issues
- Throughout March these issues need to be close to complete
- By end of January majority of research reading to be completed to aid in designing and CAD work
- End of February GT Power and Solidworks need to be learned by the team

5. Track Day possibility and what needs to be done/data acquisition
- Track day tomorrow
- Learn/do checks
- Information to be gathered:

intake - MAP pressure for warm up and track day, lambda during warm up and track
lubrication - most data already attained
cooling - most data attained however take note of the hotter day relative to comp
exhaust - most data already attained

16.01.14 Meeting Minutes
Team:
Rabieh: Present
Jim: Present
Aaron: Present
Dwayne: Present
Clarke (Integration): Present
Richie (2013 Integration): Present

1. Initial Appreciation
- More specific with values
2. Timeline/Gantt
- Look at Integrations timeline, they plan to google calendar it. Need a specific breakdown for our group to help achieve timelines
3. Concept Designs
- Optimise dry sump before trying a wet sump (maybe second half of year, have a look at wet sump for next year's team)
- Split of roles regarding ECU and tuning from IA should be revised. Better to have same people do both
- Look into RPM to maximise
- Look into testing yoshima muffler with new cams
- Obtain lambda at various operations from track days
- Design geometries for plenum via calculations
- Investigate backfire problems in intake
- Speak to Race Radiator sponsors as early as possible (have rough numbers for them)
- Work closely with chassis (body kit) for side pod designs and ensuring any design fits within the parameters
- Speak to multiple 2013 team members (Integration and engine guys) for different perspectives and important information
- Speak to Honda, they should see the car
- Ask Honda of possibility of removing 1st and maybe even 2nd gear in engine

23.01.14 Meeting Minutes
Team:
Rabieh: Present
Jim: Present
Aaron: Present
Dwayne: Absent
Peter: (Integration): Present

1. Concept Design
- Dot points with ideas; e.g. wet sump, titled radiator, concepts (keep realistic). Can use drawings.
2. Progress
- Wet sump: CAD it and join with engine model, try to work out solutions to current design idea. Gt
- Verify reasoning to drop curved radiator unless a reason can be found to pursue it. Contact Race Radiators, tag Lav to the email so sponsors know who to deal with.
- Duct design to be undertaken with Jared.
- Fuel has been investigates, swirl implementation into intake is being looked at (trying to avoid turbulence)
- ANSYS is becoming increasingly important and to be looked into in the very near future
3. Gantt Chart
- Make one, but take into account benefit gained to how much time it will take to do

4. Franken Repair
- Fix it in next couple of days, need to unjam it.
5. Repair of speed flow
- Need to drain oil sump, speak to Johnno
- Take speed flow apart, fix against leaks
6. Action items:
- Rabieh: Gantt chart, contact Race radiators
- Jim: Drain sump
- Aaron: Gantt chart, contact Honda and Valvoline
- All: Oil up and put Franken back together, setup meeting with Yi, contingency plan if Dwayne cannot do Capstone

31.01.14 Meeting Minutes
Team:
Rabieh: Present
Jim: Present
Aaron: Present
Dwayne: Present
Peter (Integration): Present

1. Status update
- Lubrication: looking at (Accusump), may be difficult/pricey to implement. Would still need a wet sump system but perhaps one that is not as complex or full proof with the accumulator there as back up when needed. Meanwhile important to optimise dry sump design and make the necessary improvements (manufacture better, get rid of leaks, and look at properly utilising the catch can).
- Intake + fuel: Running CFD on last year's intake. Getting pressure data and running simulations with it. Play around with spacers for volume alterations of plenum. CAD up some swirl ports, improves mixing and flow inside chambers.
- Exhaust: Get on the track top to try and get RPM data to see what engine RPM to optimise for. Want to also find out the effect of shifting gears as last year they were essentially stuck in second. Looking at pressure waves for performance improvement (and perhaps sound, but we have room to move in that area)
- Cooling: Radiator calculations to be done over the coming 2-3 weeks so that a shape can be devised in conjunction with the body kit design for the side pod. Look at electrical water pump failures of the past and run various tests to on the mechanical water pump to get an idea of flow performance. Want to be able to justify why the electrical water pump on its own does not deliver the flow rates provided by manufacturers.
2. Franken Needs to be fixed
- Organise meeting with Johnno for Sunday to have a look at Franken. Want to avoid taking off side of the engine to fix it.
3. Speed Flow needs to be fixed
- Speed flows are tightened but evidently not screwing on correctly as they leak on occasion more than we would like. Another thing to consult Johnno on.
4. Action items:
Jim: Contact Johnno for Sunday to look at speed flows.
Rabieh: Contact Honda, Race Radiators
Dwayne: Contact Valvoline, Speedflow
Aaron: Contact Yi
All: Ensure engine is in running order for a potential track day next week (do checks).

06.02.14 Meeting Minutes
Team:
Rabieh: Present
Jim: Present
Aaron: Present
Dwayne: Present

Peter (Integration): Present

1. Status Update
- Intake + Fuel: Running simulations in fluent, looking at flows with choked conditions for restrictor. Trying to get flow in each cylinder, different resonating frequencies in plenum, trying to solve a characteristic equation for 2 cylinder so that it can then be applied for 4 cylinders
- Exhaust: Looking into getting some simulation going in the coming week. Try to get pipe max program as well as others. Look into trying a different muffler on the car.
- Lubrication: Further investigation into wet sump + accumulator. Canterbury uni use it and have said it works. Look into simulation program for the accumulator and oil pressures. Look into emailing Accusump for more info (mounting, weight etc.)
- Cooling: Radiator design for the next 2-3 weeks. Once have all values, basic cad and modelling. Look into side pod/shroud design with chassis.
2. Gantt Chart (Urgent)
- Needs to be done ASAP by all team members with an aim of setting dates till about May.
3. Components list
- Need to do; list all components, how they will be made or got, rough timeline if you know them. Include components you would use for mounting etc.
4. Other
- Log anything you do on, an engine.
- Engine Identification need to do:
    A: new engine (broke last year, seized valve)
    B: Should be in the car, scuffed on clutch cover
    C: Franken
    D: 09 Engine
    E: Know this one
- Issue with Franken is head
5. Action items
- Jim: Johnno didn't show up last Sunday, contact him again for speed flow issues. Look at spare parts for seed flows.
- Rabieh: Contact Honda, Race Radiators
- Dwayne: Contact Valvoline, Speedflow
- Aaron: Contact Yi again. Want access to dyno (inductions)
- All: Look at what material is used for speed flow wrenches as they would be a great exercise for Holmesglen machining. Contact sponsors next week now that Lav is back. Put Franken back together.

13.02.14 Meeting Minutes
Team:
Rabieh: Present
Jim: Present
Aaron: Present
Dwayne: Present

1. Status update
- Lubrication: Looked at a past final report suggesting a 3 stage scavenge pump. Limited space between chassis and engine block may make this unfeasible. Simulations are being done for the lubrication system. Need to set up parameters for the system; pump capability, diameters etc. The point is it should allow us to input RPM data with oil pressure being the output of the system. The software has not been attained by legal means so may not be allowed to use it in final report. May try to justify any solutions gained from it in MATLAB. Need to speak to chassis and suspension about where some components may be placed. The simulation will hopefully allow us to work out the volume available for a potential wet sump and see if the 16mm wet sump can work. Speed flow being cleaned by Aaron, Jim and Peter. Some speed flow issues encountered.

- Intake + fuel: Taking a break from ANSYS for the time being. Trying to do transient analysis of the cylinders. Trying to get realistic values for intake from some control theory amongst other things. Nothing new on fuel at this stage.
- Exhaust: Going with a 4-2-1 system for sure. Need to get some parameters so that GT Power simulation can be run for optimisation. Expect work rate to pick up from this point. Gio gave details of materials for the system.
- Cooling: Looking into all calculations produced from last year regarding the radiator, flow rates in and out of air and water. Some MATLAB code has been written attempting to justify some of last year's calculations.
2. Sponsors
- With the template letter for sponsors finally complete, emailing sponsors must commence ASAP. Ensure that Lav is tagged in all emails so that he can keep a log of all sponsor related discussion. Any issues with emailing (such as unsure what to say or how elegantly you have said it) consult other team members or Lav.
3. Gantt Chart
- Lubrication and Exhaust are still yet to contribute to Gantt chart. Do this ASAP. For now you only need to be projecting forward 2-3 months but ensuring that you are progressively updating it.
4. Upcoming design Reviews:
- Expectation of the upcoming design reviews is essentially to present work that you have done to that point. This really should be in the form of figures plots, simulations, CAD etc. These design reviews will be scrutinised by your peers (and a few weeks later by academics, members of industry) so as to ensure that all aspects of your designs have been addressed. Hopefully different people looking at your designs from different angles will aid your design progress.
- Our presentation will be the Tuesday due to Aaron's absence on the Monday. Aim to have done by the Saturday before the presentation.
5. Action items:
Jim: Try to get lubrication simulation done by next week first looking at a stock lubrication system, followed by simulating last year's dry sump system and see how results compare to comp data, before finally running simulation with accumulator.
Rabieh: Breaking down radiator calculations from last year (noting they were based on 2012). Then from 2013 comp data produce my own calculations within the week. Contact Yi with my calculation process and with any questions that may arise.
Dwayne: Some preliminary calculations for the exhaust system to be done this week
Aaron: Try to get dyno inductions going, continue research into SMO (sliding mode observer) design to aid intake calculations.
All: Continue fixing speed flow

20.02.14 Meeting Minutes
Team:
Rabieh: Present
Jim: Present
Aaron: Present
Dwayne: Present
Peter (Integration): Present

1. Status update + Preview of upcoming internal design review:
- Lubrication: Accusump have been emailed with our enquiries. Basic modelling has been done. Now want to step it up and model the engine properly in this program. Worked out the benefit of the circuit connection to the engine rather than a direct connection. May be able to make the sump a little deeper due to the addition of the undertray. Look into testing the pressure relief valve on the dyno. Will present some wet sump ideas as well as a system diagram with the accumulator incorporated. Look into seeing how the simulations can be done in MATLAB.
- Exhaust: Worked out average RPM range the car experienced during competition. 6500 RPM appears to be the target/designed RPM that will be aimed for (not set in stone yet). Look to verify/improve Gio's calculations, perhaps looking into different approaches or exploring them in more detail. Look into Pipemax, maybe send out an email. Reasonably priced but still look into if GT Power can do these things. Need to get geometries at some stage and use GT Power. Will present findings so far.

- Cooling: Got values for various UA values that dissipate the required amount of heat with air and water flow rates at "average" level. Now a case of selecting what is deemed the best of these based on lowering the range of what is considered average. Currently calculating U, correcting errors that have been found in previous year's code. From that an area can be computed and more radiator specific/manufacturing parameters will be looked into via contacting Race Radiators.
-Intake + Fuel: ANSYS been done. Looking into supersonic conditions and results for various flow rates. Transient response models are proving troublesome to implement in ANSYS. Modelled restrictor, trying to minimise pressure losses across the restrictor. Look into getting sensors for the fuel system (fuel pressure, temperature) so that some analysis can be done. Looking into changing the amount of fuel being injected by changing width. Will present ANSYS simulations and some research.
<u>2. Speed flow:</u>
- Fixed female speedflow part. Peter will call Dailey Engineering for the male part. He also wants to be the one who puts them on by connecting directly to the scavenge pump.
- Look into the videos for Speed flow if it helps us get free parts.
- Measure some of the speed flow that our spanners don't fit and make spanners for them at Holmesglen. Should be easy to do on Mastercam.
<u>3. Sponsor update:</u>
- CAD models from Honda, ask Lav to organise Honda site visit, remember electrical boys have a sensor placement question.
- Race Radiators questions
- Look into who fuel supplier is.
<u>4. Gantt Chart:</u>
- Lubrication and Exhaust Gantt charts still undone. Today PLEASE!
<u>5. Getting on the Dyno:</u>
- Yi has been emailed, await a response. Need to get inducted ASAP.
<u>6. Action items:</u>
Jim: Gantt chart, model the engine for the lubrication system
Rabieh: Contact Race radiators, contact Honda (email), get some figures/graphs for design review
Dwayne: Gantt chart, preliminary calculations
Aaron: Continue working on ANSYS and on design review
Peter: Speedflow issues (contacting, putting back on car etc.)
All: Put Franken back together, pretty up car for O-week presentation, CAD up speed flow spanners.

27.02.14 Meeting Minutes
Team:
Rabieh: Present
Jim: Present
Aaron: Present
Dwayne: Present
Peter (Integration): Present
Matt (Integration): Present for design review feedback

<u>1. Internal design review feedback:</u>
- Engine general: Test Cindy for increase in compression ratio impact. Priorities; lot of ideas, need to ensure that we are not wasting time researching without manufacturing/testing.
- Intake + Fuel: Supersonic intake flow? Parameter needs to be changed in simulation for subsonic.
- Exhaust: Maximise RPM range reasoning. Be careful with sound from muffler, low noise more due to new cams. Get some torque curves to test optimise RPM range.
- Cooling: Pump data may need to be more accurate (check if in system), taken apart. Try to get MAP data for heat dissipation.
- Lubrication: Look into getting Monash data for accumulator.
<u>2. Subsystem update:</u>

- Intake + Fuel: CADed up spacers (for plenum) and swirl ports. Trying to get it on dyno and testing. Delay of throttle response increases a bit by increasing plenum volume, need to ensure not a major issue. May try adding guides in potential new intake design. E85 will still be fuel of choice, want to validate why it's used.
- Exhaust: Sacrifice top end power for torque. Finding appropriate target RPM range. GT Power needed for geometry optimisation of exhaust. Try to do some lap sims with Optimum Lap; input torque curves for 2013, lower RPM range, higher RPM and compare to find some optimum range for RPM. Try to also create car with aero, put in appropriate parameters and get an idea of its effect.
- Cooling: Have values for diverging duct and not duct (car speed = air speed). Next step get experimental data for air flow.
- Lubrication: Pushing on with getting an accurate model for the lubrication system. Trying to get flow rates through the galleries to get an idea of whether a 1qt accumulator is sufficient. About 600ml of oil are in the galleries, model should hopefully give a result close to this. Need to look at a wet sump design and see if the 16mm pan is sufficient.

3. Deadlines (Gantt Chart):
- Seem on track. Key is don't want too much time committed to research, things need to be made and tested.

4. Bill of Materials:
- Due Sunday. Do an individual one for your system and send them to be on Sunday morning at the latest so I can compile and submit it.

5. Meeting with Yi (date + preparation):
- Organise time to meet and prepare questions you want to ask him. He is marking us so important to find out his expectations.

6. Sponsor update:
- Speedflow haven't got back to us. Pete may call them.
- Race Radiators next week for a visit.
- Contacted Honda. Electrical boys have been sent the answer to the question
- Contacting Performance Exhaust.

7. Action items:
Jim: Simulations continued. Get simple answer, then expand from there. Acquire Monash data.
Rabieh: Visit Race Radiators. Plan experiment for the side pod.
Dwayne: Optimum lap sims, and get some appropriate torque curves to input. Contact Performance Exhaust.
Aaron: Swirl port iterations and start dyno plenum volume iterations.
Peter: Get and learn GT Power and review engine timelines.
All: Refill engine with oil and check for leaks.

05.03.14 Meeting Minutes
Team:
Rabieh: Present
Jim: Present
Aaron: Present
Dwayne: Present
Peter (Integration): Present

1. Subsystem updates:
- Intake + Fuel: Printing out some spacers before track day. Doing some fuel calculations to get fuel tank size. Some of last year's calculations are a little confusing. Got some graphs relating pressure and flow rates. Intake positioning is being considered for sake of aero package.
- Exhaust: Different geometries from textbook, working with GT Power.
- Cooling: Can't use wind tunnel. Will try to acquire air velocities during an upcoming track day. Need to get some form of pressure sensor and place it in the duct so that data can be logged.
- Lubrication: Got a result from simulation for a general lubrication system. Next step is to improve the model to incorporate the different sump systems.

2. External design reviews:
- Similar to internal design review, with different audience. Academics, last year's team, sponsors will be in attendance.

- Try to prepare better as far as length

3. Franken update:

- Sheared bolts for the cylinder head. Contacting Honda for a solution. If Greg doesn't reply, contact Graham.

4. Meeting with Yi:

- Tomorrow at 1. Prepare any questions and get register sheet done.

5. Sponsor update:

- Race fuels: need to contact them
- Performance exhausts: will be in contact with them soon
- Race Radiators: will drop down soon

6. Track day next Friday:

- Do checks, put in oil, run engine later this week or early next week.

7. Timeline:

- 28/3: first iteration cad assembly
- 11/4: second iteration
- 9/5: third no clashes, optimised,
- 19/5: final design freeze

8. Action items:

Jim: Acquire Monash data, improve simulation
Rabieh: Find pressure sensor, see Race Radiators directly
Dwayne: External design reviews. GT Power
Aaron: Finish fuel calculations, volume for fuel tank

12.03.14 Meeting Minutes

Team:
Rabieh: Present
Jim: Present
Aaron: Present
Dwayne: Present
Peter (Integration): Present

1. External Design Review feedback:

- Too much analysis, not enough design.
- 16 mm wet sump probably won't work based on what Ray from Holden said. May need to lift sump. Speak to suspension but likely raise engine about 8mm so that we can be more confident wet sump will work. Sticking to a 16mm sump would mean the dry sump contingency would also need to be planned out.

2. Track day:

- Work out what data you need.

3. Contact Graham of Honda for more bolts. If they are going to charge us, buy 20.

4. Scope of works due on 24/3

5. Deadlines need to be met: notice needs to be given if deadline not going to be met.

6. Action items:

All: Checks, design stuff, contact Honda, track day data
Dwayne: Rout exhaust through chassis CAD, GT Power
Aaron: Finish CAD for new intake
Rabieh: Look into blocking airflow for track day to see effect on cooling performance
Jim: First iteration of wet sump design

18.03.14 Meeting Minutes

Team:
Rabieh: Present
Jim: Present
Aaron: Present
Dwayne: Present

1. Subsystem updates:
- Intake + Fuel: Getting different torque curves for different runner lengths using GT Power. Started cadding up new intake for this year's system. Run simulations based on MAP data acquired from track day. Fuel tank being looked at (internals, being done by Rabieh).
- Exhaust: Look at reroute to back of an engine, a 4-2 system would need another muffler, the same muffler and would add weight so probably a poor idea. May rout the exhaust up, increase COG however. Running through GT Power sims, getting data and seeing effect of various parameters. Looking to work on dyno. Exhaust can be accommodated by chassis.
- Lubrication: Has CAD 3 different style baffles. Thickest sump 20mm, look at 24mm. Test internal pressure relief valve.
- Cooling: Emailed Race Radiators. Asked about surface areas of tubes. Spoke to Darcy about packaging, maybe triangle radiator. Look at the effects of outriggers on air flow. Maybe delay giving radiator dimensions, to acquire better data, water flow sensor maybe put in soon. Most likely take a couple of days to make radiator.
2. CAD:
- Need to start for first iteration due date (28/3)
3. Emissions clean up:
- Thursday
4. Meeting with Yi and Kai:
- Yi: 1-2 prepare any questions
- Kai: time not finalised
5. Scope of works due on 24/3:
- Be done by Sunday.
6. Sponsor/Supplier updates:
- Honda:
- Performance exhaust:
7. Action items:
All: Clean emissions
Dwayne: Finalise geometry of exhaust based on current chassis
Aaron: Finish off intake cad design, look at effect of various lengths with 6 litre plenum
Rabieh: First radiator iteration cad, research into the fuel tank
Jim: Get material for sump, get information about PRV, Monash email

25.03.14 Meeting Minutes
Team:
Rabieh: Present
Jim: Present
Aaron: Present
Dwayne: Present

1. Sub-team update:
- Intake + Fuel: Will be doing CAD for submission.
- Exhaust: Re-modelled engine on GT Power. Done most torque and power graphs for different iterations. CAD work.
- Cooling: Prepare track day tests and CAD for first submission.
- Lubrication: Simulating sloshing in wet sump in ANSYS (issues encountered). Researched PRV and test rig for it to be done at Holmesglen.
2. 1st CAD iterations:
- Aaron assembling CAD, get CAD in to him by Thursday
3. Emissions:
- clean this afternoon
4. Track day:
- do your checks tomorrow morning
5. Kai meeting 3.15 Thursday

6. Action Items:
- All: do checks, CAD
- Dwayne: Take exhaust off
- Aaron: Fluent sim of oil sump, fuel pump choice
- Jim: testing PRV, investigate sump material, drain sump and catch can, and contact Rex and Hash
- Rabieh: track day tests, fuel tank baffle research, re-do radiator calcs with safety factor = 1

08.04.14 Meeting Minutes
Team:
Rabieh: Present
Jim: Present
Aaron: Present
Dwayne: Present
Peter: Present

1. Sub-team updates:
- Intake + Fuel: Pressure losses across components, joint solver on ANSYS to iterate through and get angles for intake for smooth flow. Dyno Thursday.
- Exhaust: Runner lengths calculations being done, diameter values are odd. CAD exhaust for next iteration to help guide calculations. Iterate through using GT Power. Check how much exhaust material available.
- Cooling: Radiator CAD iterations. Race Radiators will get to us a surface area of tubing value that can be compared to calculations already done.
- Lubrication: Some sump analysis in ANSYS. Jig done to test internal PRV. Taking it to Holmesglen. Estimated oil in wet sump system as being about 2.39 L. Chinese uni did it with 30 mm sump.
2. Continental parts:
- Aftermarket sponsorship, look at catalogue, list of possible things that would be needed, Lav needs this ASAP
3. CAD 2nd iteration:
- Due at different dates for different components.
4. Sponsor update:
- Accumulator purchase. Get Lav on Precision International and see if any sponsorship can happen.
- Got Oil from Valvoline
5. Johnno meeting 5.30pm Friday
6. Timelines:
- Re-look at Gantt
7. Action Items:
- Jim: Look into jig manufacture
- Rabieh: Revise radiator calculations
- Aaron: Oil simulation
- Dwayne: Performance exhaust contact
- All: CAD

22.04.14 Meeting Minutes
Team:
Rabieh: Present
Jim: Present
Aaron: Present
Dwayne: Present
Peter: Present

1. Sub-team updates:
- Intake + Fuel: Run simulations on another iteration of intake CAD. Trying to increase MAP Pressure whilst trying to optimise in ANSYS with varying runner lengths. E-85 will be used for sure, due to its properties in particular for use in Franken engine.

- Exhaust: Chassis change, twin exhaust may be considered. Need to CAD up new exhaust to match new chassis design.
- Cooling: Contacted Race Radiators. Awaiting response regarding tubing area and manufacture time. Looking at newest CAD iteration to change current radiator iteration.
- Lubrication: Jig being made for PRV. Need to make new adapter plate for the sump.

2. Manufacturing Time Lines:
- Sump iterations most important. Need to be making both dry and wet, and testing the wet sump system.

3. Engine checks, Running Engine, Track day:
- Done checks for non-running engine. Need electricals on the car to finish off checks.

4. Timelines/Setting due date:
- Gantt chart updates. Gantt project could be used (needs to be downloaded, easy to use).

5. Dyno:
- Not working. Gio needs to debug it. No power through the ECU. Need to speak to Gio.

6. This week's goals/Action items:
- Franken put back together, Pete to do rig for engine change, CAD of first sump iteration, Accumulator order, Exhaust CAD by Friday, Contact Honda regarding engine assembly/disassembly and Amber repair

15.04.14 Meeting Minutes

Team:
Rabieh: Present
Jim: Present
Aaron: Present
Dwayne: Present
Peter: Present

1. Sub-team updates:
- Intake + Fuel: Simulation of runner to runner stealing of air, middle two runners not getting air. Circular shape, grooves around edges.
- Exhaust: Diameters and lengths done. Now need to select. Called performance exhaust, didn't get back to us. Look at what tubing can be got. Scrap metal for muffler design tests. Exhaust CAD to be re-done (add mass, clash with engine, solid tubes).
- Cooling: Await CAD update to see how radiator placement is going. Prepare for track day next week for radiator tests. Contact Race Radiators.
- Lubrication: Sump simulations still happening. CAD baffles for sump, PRV jig made over next couple of week, is it fails, CAD up external PRV. Loom at dumping oil from relief into a strategically chosen quadrant.

2. Dropping engine for welding, putting it back on for next week track day.

3. Action Items:
- Jim: jig manufacture
- Rabieh: track day test preparation
- Aaron: Sump simulation
- Dwayne: Performance exhaust contact

29.04.14 Meeting Minutes

Team:
Rabieh: Present
Jim: Present
Aaron: Present
Dwayne: Present

1. Status update:
-Intake + Fuel: Intake simulation failed.
-Exhaust: CAD has been made with routing, compared geometries between CAD and simulations (not too bad). Similar to last year, smooth torque curve.

-Cooling: Wasn't able to get radiator tests done from test. Planning radiator positioning, I might call Race Radiators to get details. If none forthcoming, persist with similar radiator to previous CAD with clear mounting points defined.
-Lubrication: Tested internal PRV. 4.8 bar was the release pressure. Oil checks to be changed to include for track days.
2. Franken:
-Put back together this week. New Gasket is in it.
3. Timelines:
-Gantt Chart for progress report 1 to be done as well as report itself which is due in a few weeks. Progress report draft to be submitted in by 23/5.
-Endeavour picture next Tuesday.
4. Action items:
-Franken back together, Cooling CAD, further PRV testing regarding flows, finish sump CAD, another Exhaust CAD iteration and simulations, fuel calculations.

06.05.14 Meeting Minutes
Team:
Rabieh: Present
Jim: Present
Aaron: Present
Dwayne: Present
Peter: Present

1. Status update:
Intake + Fuel: Ran intake sims. Increase runner lengths by 10 mm, and trying to see effect of lower band torque curve moved up. Want to increase torque at lower bound and main used bound. Fuel calculations being done regarding fuel consumption. Aiming for lambda of 0.9ish. Looking at MEP and how it effects performance. Doing CAD for fuel tank.
Lubrication: Mounting positions for Accusump being determined. Tested internal PRV, doesn't perform well at high RPM. Will try a different spring. Need longer but less stiff spring.
Cooling: Design finalised this week with all CAD done. Radiator positioning, dimensions all finalised.
Exhaust: CAD geometries inputted into GT Power to see results. Results seem fine. Need to ensure clearance to firewall.
2. CAD Submissions + Machining:
-Expectations for CAD: nuts, bolts, tabs etc. All components.
-Components to be machined need to be designed and finalised as soon as possible to maximise Holmesglen time.
3. Honda Engine lesson + Honda repairs:
- Leave it to a later time.
- Repair Amber and see shifter arms
4. Meeting with Yi + contact with Examiner:
- Meet Yi this week.
5. Franken
- Put bolts back on tomorrow.
6. Action items:
-Peter: Sort out waste oil disposal
-Jim: CAD + oil routing
-Rabieh: CAD + water routing
-Aaron: CAD + fuel stuff
-Dwayne: CAD + contact Barry regarding welding lessons
-All: Franken

13.05.14 Meeting Minutes
Team:
Rabieh: Present

Jim: Present
Aaron: Present
Dwayne: Present
Peter: Present

<u>1. Status Update:</u>
- Lubrication: Accumulator to be moved so that it can be mounted from its correct points. In the case of a dry sump configuration, oil tank will go on opposite side with catch can moving to the other site.
- Intake + Fuel: CAD done for intake. Fuel tank routing won't come out of firewall. Considering baffles or flaps within fuel tank. Thinking about manufacture. 2.4L plenum volume. Fuel tank is longer but narrower.
- Exhaust: Looking into making new y-collectors as dimensions are still too close to firewall. Variation in dimensions that can fit relative to best GT Power results is small. May make a heavier contingency exhaust before an optimised one. Need to look into jig design.
-Cooling: CADing routing

<u>2. Action items:</u>
-All: Do checks
-Rabieh: Get CAD ready for Race Radiators, finish routing and tabs
-Aaron: Full iteration of proper intake. Prepare for track day.
-Jim: Fix baffles in sump, re-position Accusump, have sump ready for machining
-Dwayne: CAD iterations of exhaust, ask Lav about material
-Peter: Get aluminium billet

20.05.14 Meeting Minutes
Team:
Rabieh: Present
Jim: Present
Aaron: Present
Dwayne: Present

<u>1. Status updates:</u>
-Intake + Fuel: Started running some fluent and GT Power models together. Iterating through intake design. Started 3d manufacturing of the plenum. Franken is back together. Fuel tank CAD would be re-done because of seat clash.
-Exhaust: Exhaust has some clearance now. Position of first collector is what is closest to clashing. Muffler design in GT Power being done, Performance Exhaust have offered a free muffler for us to test. Want to test on Dyno. Finish CAD in coming days. Looking into sourcing material. Thinking about putting an aluminium plate between firewall and engine but that will be subject to mass and how it may affect suspension. Look into practicing welding.
-Lubrication: Fix PRV yesterday. O-ring issue within PRV. Finalise sump iteration with a laser cut frame so that baffles can be removable. Can be put into Mastercam now. Going to print out a part for the pickup point. May need additional treatment to avoid damage within the sump. Need new gasket for sump.
-Cooling: Routing is done. Will look to optimise the way it is done having all other parts around it. Need to move header tank as it is in the way of the fuel rail. Need to fix some inaccuracies in CAD.

<u>2. Run engine:</u>
- To be done Thursday evening.

<u>3. Progress Report 1:</u>
- Draft in Friday.
- Latex file to be set up
- Make an account Share Latex

<u>4. Franken:</u>
- Find a starter motor and set up test rig.

<u>5. Honda:</u>
- Contact Honda regarding Amber so that engine can be taken down.

<u>6. Machining:</u>
- Need to Mastercam adapter for sump

7. Action items:
- All: Tell Pete about taking Amber down
- Jim: Confirm Accusump mounting, CAD baffles
- Dwayne: Get Mastercam, finalise CAD for exhaust, source material
- Aaron: Make plenum
- Rabieh: Finish Eng drawings for Radiator to be sent off

03.06.14 Meeting Minutes
Team:
Rabieh: Present
Jim: Present
Aaron: Present
Dwayne: Present
Peter: Present

1. Status updates + Latest CAD submission:
-Intake + Fuel: Intake and Fuel CAD done. CAD up to 3D print top half of plenum.
-Exhaust: Exhaust CAD close to done. Tab needs to be optimised. Need to double check material order. Jig design has been started, will likely be cut at Herron or ARB. Herron happy to-do orders in multiple lots.
-Lubrication: Pretty much finished both Wet and dry sump CAD. Mounting Accusump lower, still on right hand side.
-Cooling: CAD done. May have some routing inconsistencies due to the change in the positioning of the scavenge pump.
2. Machining:
- Get Sump Mastercam checked. Billet may be a while.
- Sight glass adapter. Second operation needs to be Mastercam-ed
- Exhaust port adapters. Should be done in very near future.
3. Action items:
- Peter: Pete work out time for Honda drop off of Amber.
- Jim: Sort out Accusump mounting and have final Mastercam of sump
- Dwayne: Make a design/manufacture time for exhaust jig.
- Aaron: 3D print top half of plenum.
- Rabieh: Work on radiator tabs.

10.06.14 Meeting Minutes
Team:
Rabieh: Present
Jim: Present
Aaron: Present
Dwayne: Present

1. Exhaust Jig status:
- Most of the tubes have been jigged. Just need to finish and finalise all cavities.
2. Dyno timeline:
- Aaron sent out a timeline that for now will be followed until further notice regarding dyno. Will hear from Yi maybe this Friday regarding Dyno availability.
3. Machining things:
- Port Adapters to be done on either Mazzak or Mori Seiki. CAD is done for it. Material is there (stainless).
- Sump Mastercam needs to be re-done. Don't have material yet. '
- Sight glass adapter first op is done just needs to be machined on Mazzak.
4. Accumulator Mounting:
- mounting at the bottom near the jacking bar area.
5. Radiator/Shifter clash:
- Shifter will be moved up.

6. Tabs:
- Accumulator: Tabs aren't known yet as position of accumulator isn't certain.
- Exhaust: Positioning is same as most recent CAD submission. Tab design/optimisation not finalised.
- Radiator: Positioning is same as most recent CAD submission. Tab design/optimisation not finalised.
7. Budget:
- Lubrication is in negative 282 as billet not accounted for in earlier year budget and with speedflow not considered the deficit will increase.
- Fuel will account for 120 of deficit, 50 out of cooling, 50 out of exhaust, 50 out of spare (all conservative figures). May be getting some money from suspension ($100-150).
- Further fund related info to come
8. Status Update:
Lubrication: Dry sump routing on CAD is done
Intake + Fuel: Run calculations regarding volumetric efficiency, might be getting higher pressure at intake. May need to design a duct to the intake. Fuel tank re-designed. Run a sim with improved compression ratio, 1% power increase (same fuel map).
Cooling: nothing new
Exhaust: nothing new
9. Action items:
- Rabieh: Contact Harry regarding air flow from latest CFD.
- Aaron:
- Jim:
- Dwayne:

24.06.14 Meeting Minutes
Team:
Rabieh: Present
Jim: Absent
Aaron: Present
Dwayne: Present
Peter: Present

1. Status Updates
-Exhaust: Jig being finalised, to be done by the end of the weekend. Port adapter's tech drawing to be ready for machining.
-Intake + Fuel: Re-printing top half of plenum. Need to see Yi regarding Dyno. Baffling or surge tank to be designed.
-Lubrication: Parts to be machined such as sight glass adapter, sump, pick up etc. On dropbox will be the files. New way of machining sump. Some laser cutting stuff to be done for baffles. Wet sump/external PRV will be first thing tested. Route/mount accumulator, CAD dry sump.
-Cooling: Tab design to be complete, confirm Race Radiators order. Work with Harry regarding CFD modelling.
2. Fixing amber:
Contact Graham and see if we can take done Amber tomorrow
3. Final year Reports:
Set up Latex file which we will update periodically.
4. Fuel Tank Design:
Look into papers and last year's surge tank design, as well as fuel tank with baffles.
5. Tests for Track:
Maybe Radiator experimental data.
Fuel map change.
6. Tab optimisation:
Tab designs to be done by Monday and optimised soon after.
7. Action Items:
Dwayne: Finish Jig
Aaron: Print Intake, order O-rings

Rabieh: Confirm Radiator Order, finish tabs for everyone, email Honda regarding Amber
Jim: CAD

17.07.14 Meeting Minutes
Team:
Rabieh: Present
Jim: Present
Aaron: Present
Dwayne: Present

<u>1. Status Updates</u>
Exhaust: Looking into muffler design or adjusting last year's muffler.
Intake + Fuel: Printing some parts, big printer not working for plenum at the moment. Fuel tank stuff sent to Rob Black. Surge tank and fuel pick up design done.
Lubrication: All parts for wet sump are obtained, start routing it and prepare it for next week track day. Get gasket done after tab jigs.
Cooling: Radiator has been ordered. Waiting for CAD to better position EWP and re-do routing.
<u>2. Cost report:</u>
To have an assembly up by Tuesday each.
<u>3. Design report:</u>
Have been submitted, await next submission.
<u>4. Manufacture:</u>
Machining:
Port adapters
Exhaust Sleeves
Fuel pickup
Fuel tank port
Dry sump
Non-machining:
Oil tank
Catch Can
Fill entry
<u>5. Honda engine stuff:</u>
Amber is stuffed.
<u>6. Action Items:</u>
Dwayne: cut tubes, muffler cad
Aaron: O-rings order, intake printing stuff
Rabieh: Call Race Radiators
Jim: 2014 Lubrication assembly, O-rings order
All: Cost report, organise Yi meeting, laser cut tabs and baffles

30.07.14 Meeting Minutes
Team:
Rabieh: Present
Jim: Present
Aaron: Present
Dwayne: Absent

<u>1. Lubrication update:</u>
-Look into another baffle design as well as pre-charge for Accusump.
-May need oil tank for dry sump.
<u>2. Cooling update:</u>
-Call Race Radiators
-Re-CAD routing and find hose

3. Progress Report 2:
Pushed back slightly for Yi to return on the 12/8. Draft PowerPoint to be done by 8/8 5pm. See last year's stuff for help.
4. Final Report:
Link to be sent of the Latex file
Structure like last year
Aim to put key information in the body; specifically numbers and graphs rather than lots of pointless text.
5. Cost report:
Fuel tank
Lubrication
6. Manufacture:
Machining:
Port adapters
Exhaust Sleeves
Fuel pickup
Fuel tank port - CAD needs to be done
Dry sump
Non-machining:
Oil tank - TBD
Catch Can - DXF nearly ready
Fuel tank - DXF nearly ready
Fill entry - TBD
7. Action Items:
Dwayne: Jig needs to be welded and exhaust welding started.
Aaron: Fuel tank Mastercam components to be in CAD
Rabieh: Find out when Radiator will be done, re-CAD routing, find hose
Jim: Put dry sump on, look at last track day data
All: PR2 to be done

02.09.14 Meeting Minutes
Team:
Rabieh: Present
Jim: Present
Aaron: Present
Dwayne: Present

1. Status and manufacturing update:
Intake + Fuel:
-Intake to be printed in the next week by sponsor AMS
-Injector assembly to be done this week
-Fuel tank to be done end of next week
-Order air-filter this week
Exhaust:
-Need to machine Y-collector material
-Position for pyro and lambda sensor needs sorting
-Order stainless steel billet
-Get clamp for exhaust
-Coating next week
-Need tail pipe
-M7 nuts on their way
-Springs to be bought
Lubrication:
-PRV on the way
Wet Sump:

-Accumulator spot sorted
-Design tab for accumulator
-Gasket Paper (better than last stuff)
-Teflon tape for adapter on sump
Dry Sump:
-Oil tank manufacturing (baffles, tops and bottom laser cutting, sort out weld ons, sort out oil temp sensor bung)
-Sump
Cooling:
-Routing (CAD done, find and source hoses)
-Header Tank manufacturing (tube profiled, drill holes still need to be done and welded, lid)
General:
-CAMS refurbishment
-Honda parts ordered
-Get minions to design engine change rig
<u>2. Budget:</u>
-Got $1100
  600 Cams
  215 Engine Parts
  100 Filter
  185 Exhaust
<u>3. Action items:</u>
Jim: Accumulator tab, sort out oil tank
Dwayne: Sort out coating, sort out position for pyro and bung, order the stainless steel billet
Aaron: Fuel injector system together, acetone intake, sort out springs purchase
Rabieh: Sort out routing, header tank manufacture

21.09.14 Meeting Minutes
Team:
Rabieh: Present
Jim: Present
Aaron: Present
Dwayne: Present

<u>1. Final Report:</u>
-Draft of whole report due 27/9.

01.10.14 Meeting Minutes
Team:
Rabieh: Present
Jim: Present
Aaron: Present
Dwayne: Present

<u>1. Final Report:</u>
-First two subheadings should be the same for consistency.
-Summary should say what you are trying to achieve.
-Body be done by tomorrow
-Redo lit review
<u>2. Endeavour Poster:</u>
-Give some thought to poster
<u>3. Conference:</u>
-Think about PowerPoint
<u>4. Run engine/Shakedown:</u>
-Header tank manufacture

-Muffler cut
-Diffuser re-printed
-Get fuel tank and catch can
-Manufacture new pick up
-Implement new baffle design
-Weld tab to muffler
-Copper gasket
-Gasket paper
-Bolts needed

5. Action items:

-Finish body of report by tomorrow.
-Pick up Honda stuff next week
-Order Copper Gasket