Formula SAE Turbocharger System Development

Senior Project Final Report

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Executive Summary

This project, Formula SAE Turbocharger System Development, was sponsored by the Cal Poly, San Luis Obispo Formula SAE team. The team proposed this project in order to have a powerful yet lightweight engine so they can be extremely competitive at their competition. The baseline output of the single cylinder 450cc engine (2006 Yamaha WR450F) was 46 horsepower and 27 ft-lb of torque. The goal of this project was to increase the output of that engine to 60 horsepower and 35 ft-lb through the use of a turbocharger.

The addition of a turbocharger to this engine required the design of multiple subsystems such as the intake, exhaust, oiling, fuel, and boost control. Ricardo WAVE engine simulation software was used to optimize designs without having to spend time, money, and resources on actual dynamometer testing of all possible configurations. The simulation's output was determined to be an accurate representation of the actual engine's output based on comparison to results obtained from the naturally aspirated engine. The final design produced a simulated power curve with peaks of 63 horsepower and 45 ft-lb of torque at 8 psi of boost.

Two turbochargers were available to select from Garrett: the GT12-41 and the GT15V. The GT12-41 is the smallest turbocharger Garrett currently has available and the GT15V features variable vanes. The GT15V was selected based on matching the compressor map from each to the engine's predicted operating range as well as quicker boost response offered by the variable vane design. To take full advantage of the turbocharger's potential for power, the static compression ratio of the engine was lowered from 12.5:1 to 11:1 and E-85 was selected as the fuel.

A new piston, connecting rod, valve springs, head studs, head gasket, clutch springs, pressure plate, and modified crankshaft were installed to withstand the increased power output. These components were selected for the engine to safely produce 85 horsepower, allowing room for future Cal Poly FSAE teams to further increase the engine's output.

During testing it was discovered that the variable vanes on the GT15V are not able to limit boost levels below 15 psi. Oil leakage past the compressor seal also proved to be problematic, with a significant volume of oil being sucked into the intake and burned by the engine. Testing also revealed that the engine's factory oil pump cannot provide enough pressure to feed the turbocharger when the engine and turbocharger are plumbed in parallel. While engine output was not measured across an RPM range, one data point was obtained which proves that this engine is capable of producing and surviving the desired power: 55 hp and 40 ft-lb at 7200 RPM and 15 psi of boost. Re-running the WAVE simulation with 15 psi of boost instead of 8 psi predicts 58 hp and 42 ft-lb at 7200 RPM. These figures are realistically obtainable with more time spent tuning and lend further credibility to the simulation's accuracy, further suggesting that this system is capable of reproducing the predicted power. We believe that switching to the GT12-41 is the most effective way to regulate boost pressure without adding weight. This should be done in conjunction with continued refinement of critical subsystems in order for the turbocharged engine to become a reliable source of power for FSAE.

1.0 Introduction

Jackie Stewart, a three time Formula 1 World Drivers' Champion, once said "It is not always possible to be the best, but it is always possible to improve your own performance." SLO Racing's challenge was to implement a turbocharger system onto Cal Poly's Formula SAE Team car to greatly improve upon the vehicle's performance. Our senior project team, SLO Racing, consists of Matt Roberts, Eric Griess, and Kevin McCutcheon who are all Mechanical Engineering students at Cal Poly, San Luis Obispo. Our goal was to successfully design and install a complete turbocharger system that would meet all SAE competition regulations for the 2013 Formula SAE car. Stakeholders of this project include the current and future Cal Poly Formula SAE Teams, supervisor John Fabijanic, aspiring engineers who are motivated by the project, and Cal Poly's College of Engineering. This report outlines the details of the project, including a background to cover the basics of internal combustion and forced induction, requirements of the project, subsystem design development, manufacturing of parts, and testing the turbocharger system.

2.0 Background

The internal combustion engine is a man-made marvel that is the driving force behind Formula SAE cars. Stringent SAE regulation of these engines pushes undergraduate engineers from all over the world to design the strongest and fastest engines they can within competition specifications. Some of these specifications include a 4-stroke cycle, displacement restriction of 610cc, an intake air restrictor, and limited fuel choice. In order to successfully design within these parameters, it is necessary to understand how each ones affects the desired performance of the engine.

2.1 Combustion Basics

For combustion, the only necessary ingredients are fuel and an oxidizer (air in our case). With gasoline, a complete combustion is achieved when the air to fuel ratio (AFR) is stoichiometric, or approximately 14.7:1. Keep in mind that this is a mass ratio, meaning that for every 1 unit mass of fuel, 14.7 units mass of air are needed for complete combustion. Complete combustion is an ideal case, however, and doesn't necessarily produce the most power. Increasing the fuel slightly (~10%) so the AFR is nearly 13:1 has experimentally produced more power than the stoichiometric AFR.

2.2 Engine Basics

The 4-stroke engine specification determines how often this combustion takes place. These 4 strokes consist of the following:

1. **Intake stroke** – The piston moves from top dead center (TDC) down to bottom dead center (BDC) inside the cylinder while the exhaust valve is closed and the intake valve is open. This creates a temporary low pressure area and the air from the surroundings rushes in to equalize the pressure. While this intake air enters the cylinder, fuel vapor is sprayed into the air (port injection) to enter the combustion chamber as an air/fuel mixture.

2. **Compression stroke** – The intake valve closes and the piston moves upwards in the cylinder, compressing the air/fuel mixture. The amplitude of this compression is the compression ratio. This ratio is defined as:

$$\text{Compression Ratio} = \frac{\text{Free volume in cylinder when piston is at BDC (VBDC)}}{\text{Free volume in cylinder when piston is at TDC (VTDC)}}$$

3. **Power stroke** – Just before the piston reaches TDC, a spark ignites the compressed air-fuel mixture to combust the fuel. The volume of the products of this chemical reaction is much greater than the reactants, so the combustion creates a very high pressure area that does work on the piston, moving it downwards.
4. **Exhaust stroke** – When the piston nearly reaches BDC, the exhaust valve opens and allows the pressure to equalize through the exhaust. Then the piston travels back upwards and expels the remaining products. At the top of this stroke the exhaust valve closes and the intake valve opens, restarting the cycle.

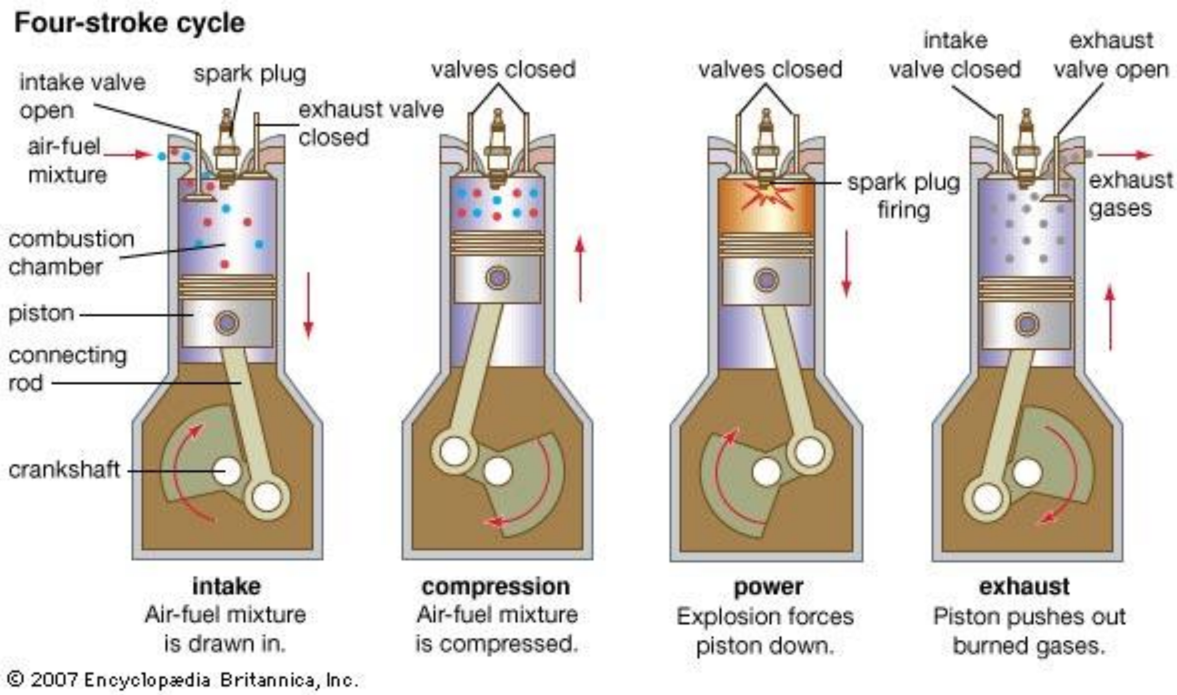


Figure 1: Four Stroke Cycle

From www.britannica.com

The important thing to note from the 4 stroke engine is that power is only produced for 1 out of the 4 strokes**.** Displacement has a large effect of how much power can be produced in that one stroke, because a larger displacement allows more air/fuel mixture into the cylinder while the intake valve is open. Combusting more air/fuel mixture creates a higher pressure, and consequently more work.

2.3 Forced Induction

There are ways to work around displacement limitations. If more air/fuel mixture is forced in during the intake cycle, higher power output numbers can be achieved. The process of creating a high pressure area outside of the intake valve so more mixture can be forced in during the intake stroke is called forced induction. The two main methods of forced induction are turbocharging and supercharging. Both methods use compressors, creating more pressure inside the intake manifold, but the main difference is how they are driven. A supercharger is mechanically driven by a belt connected to the crankshaft; while a turbocharger uses exhaust gases to power a turbine, which shares a shaft with the compressor.

Figure 2: Cutaway of Turbocharger

(From www.hipermath.com)

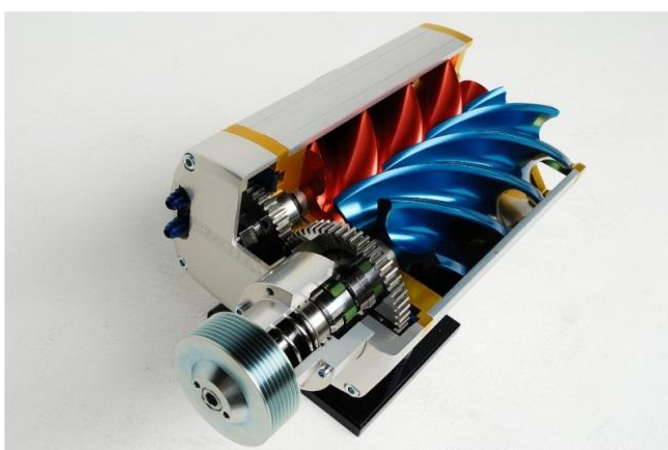


Figure 3: Cutaway of Supercharger

(From www.hipermath.com)

2.4 Air Restrictor

The power produced by an engine is the product of the torque (work) created by the engine and the angular speed at which the crankshaft is rotating. However, the faster an engine is rotating, the more air it needs for combustion. This is where the air restrictor specification plays a large role. To prevent teams from creating immense amounts of power by spinning the engine extremely fast, there is a small restrictor that the air must pass through (20mm for gasoline and 19mm for E-85). At higher speeds, the restrictor can't allow enough air flow for the fuel to fully combust and therefore "chokes" the engine. The most common way that teams have approached this restriction in the past was by designing a converging-diverging nozzle that optimizes flow and minimizes losses. However, the restrictor still plays one of the largest roles in power limitation.

2.5 Fuel Choice

Fuel choice plays a large role in the magnitude of the compression ratio that is possible, size of the restrictor, how much forced induction is possible, and choice of hardware. Fuel has such a large effect based mainly on its octane rating. Octane is a hydrocarbon (C_8H_{18} series) that is obtained in the refinement of petroleum [2] and is part of the gasoline fuel mixture. A higher octane rating indicates a higher concentration of octane in the gasoline, and increases the temperature of combustion for the fuel. Since temperature rise of the mixture is proportional to the amount it's compressed due to the ideal gas law, higher octane fuels are used with engines with higher compression ratios to prevent pre-ignition. Pre-ignition is a phenomenon in which the air/fuel mixture is compressed but reaches its temperature of combustion before the spark initiates combustion, resulting in the combustion of fuel much earlier than desired. This creates a force against the engine's natural movement which can cause catastrophic damage to the engine and should be avoided at all costs.

The Formula SAE organization also expands the choice of fuel to include E-85. This is a mixture comprised of 85% ethanol and 15% gasoline. Ethanol (C_2H_5OH) is a volatile alcohol that is mainly made by fermenting and distilling starch crops such as corn [3]. This type of fuel provides many benefits to engine operation including:

- Ethanol is an alcohol, and it therefore draws much more energy from surrounding air as it is injected into the air stream, meaning that the air/fuel mixture going into the engine is much cooler and denser than it would be using gasoline. Denser air allows for more mass of air into the cylinder, which also means more fuel can be burned creating more power. Running at cooler temperatures is also easier on engine components and increases the lifetime of the engine.
- Ethanol has an estimated equivalent octane rating of 105, and has higher resistance to pre-detonation or knocking than gasoline. This compares favorably to the 91, 93, and sometimes 98 octane gasoline provided at the competition. Therefore the use of ethanol allows for higher compression ratios which allow forced induction to be implemented with less risk of knocking problems.

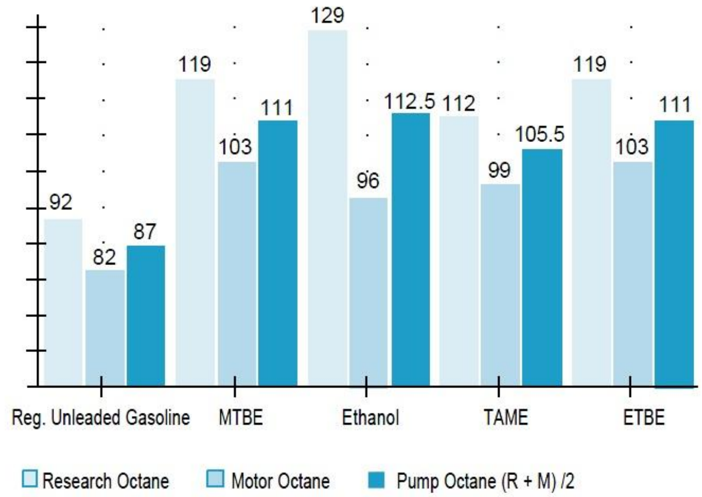


Figure 4: Equivalent Octanes of Various Fuels [9]

However, some disadvantages of E-85 compared to gasoline include:

- E-85 has a lower energy density than gasoline, meaning gasoline has a higher energy output per mass. When E-85 is used, more fuel is needed to create the same amount of energy.
- The ethanol in E-85 is very volatile and reacts poorly with many materials, including cork, some rubbers and plastics, and raw aluminum. Components in the fuel system need to be changed to accommodate the volatility of E-85 and the larger volumetric flow rate necessary to compensate for lower energy density.
- Ethanol draws water much more easily than gasoline, so it must be carefully stored in either stainless steel or certain plastic containers.

2.6 FSAE Engine History at Cal Poly

Figure 5: FSAE Car, Yamaha R6, WR450

From www.yamaha-motor.com

Until 5 years ago, Cal Poly Formula SAE implemented naturally aspirated inline 4 cylinder Yamaha R6 (600cc) engines to power their cars. Some benefits included higher power output and greater reliability. However, these engines were especially heavy, so the team switched to a naturally aspirated single cylinder Yamaha WR450 (450cc) engine. Although this engine had a much lower power output and smaller displacement, it saved almost 70 pounds on the total weight of the car, therefore improving the car's power to weight ratio.

2.7 Ways to Increase Power of the Single Cylinder Engine

The largest considerations for the engine are reliability and the expected increase in power. Reliability is important because the engine has historically been the least reliable part of the car and needs to run in order to finish the competition. Increasing the power is the entire point of this process, so naturally it is weighted very heavily. Low weight is important but the power to weight ratio is more important. Lastly, feasibility must be taken into account when considering possible options; if it cannot be successfully implemented then there is no point in pursuing it. Table 1, below, shows a decision matrix of our selection choice.

Table 1: Decision Matrix for Increasing Power

		Power Increase Methods							
GOALS	(Weight)	Forced Induction	Intake	Exhaust	Cam Profile	High Compression	Variable Valve Timing	Direct Injection	Head Porting
Good Reliability	5	**4**	4	4	3	3	2	2	5
Power Increase	5	**5**	1	1	1	1	2	2	1
Low Weight	3	**3**	5	5	5	5	4	4	5
Low Cost	1	**2**	3	3	4	4	1	1	2
Feasibility	5	**4**	1	1	2	3	1	1	3
	Total:	**76**	48	48	49	54	38	38	62

A former senior project team, Speed Systems, previously implemented intake design, exhaust design, cam profile, and high compression ratios to increase power. At the time, those options were the best, but since they have already been optimized there is very little room for further improvement.

Variable valve timing and direct injection are two new technologies found on many modern engines that give more power and reduced fuel consumption. However, the gains from these features would be small and would not justify the resources spent on them. Head porting is the re-shaping of the intake and exhaust ports to allow for more flow, which in turn allows for more power. The issue is that this would only provide a small power increase because the intake restrictor limits the amount of air getting to the head, so air flow through the head is not the limiting factor. This leaves forced induction as the best option because it allows for the greatest increase in power while still balancing the other criteria.

2.8 Which Engine is Right for the Car?

Increasing the power of the single cylinder is great, but if it is not the best fit for the entire car then it should not be used. Table 2 below, uses past information from the 4 cylinder and single cylinder performance characteristics and compares them to the projected performance of a turbocharged single cylinder. Table 2 shows that the turbocharged single cylinder engine is the best choice for the Formula SAE car.

Table 2: Engine Choice Decision Matrix

		Engine Choice		
	(Weight)	4 cylinder	Single Cylinder	Turbocharged Single Cylinder
Good Reliability	5	5	4	**4**
Power	4	5	1	**4**
Low Weight	5	1	5	**4**
Good Fuel Mileage	1	1	5	**3**
Uniqueness	3	1	3	**5**
Low Cost	2	5	4	**3**
Feasibility	5	2	5	**5**
Drivability (torque curve)	4	4	3	**5**
	Total:	**90**	**108**	**125**

2.9 Engine Conclusion

The tight interweaving of variables such as fuel choice, compression ratio, induction method, engine type, weight, and flow restriction create design challenges that FSAE teams all over the world strive to perfect, and how teams overcome these obstacles bring out the engineering talent that each competing school has to offer. For the 2013 Formula SAE competition, Cal Poly FSAE has determined that a turbocharged single cylinder engine is the best option to power the car.

3.0 Requirements and Specifications

3.1 Goals of the project

The main objective of this project was to design and implement a functional turbocharging system to a single cylinder engine to produce more power than the previous engine design. This included installing a turbocharger, designing and fabricating an intake and exhaust to accommodate the turbocharger, installing internal engine parts that could handle the increased power output, designing a fuel system that could supply the increased demand of E-85, designing an oiling system to supply the turbocharger, and tuning an Engine Control Unit (ECU) based on optimized fuel delivery and ignition timing maps. Below is a list of requirements that SLO Racing created to define the objectives of the project:

- **60 Horsepower:** A well designed single cylinder turbocharged engine can achieve around 70 peak horsepower. However, since this was the first year implementing such a system, our goal was a more conservative 60 horsepower. We did not target peak horsepower because engine components become more prone to failure, but future teams can optimize power as they learn more about the system and ways to overcome obstacles that arise from forced induction.
- **35 lb-ft of torque:** The torque from the engine should peak in the lower rpm range and stay there throughout the power band. We aimed for a target of 35 lb-ft of torque. This allows for fewer shifts which will reduce lap times and also allows for more time on the throttle and concentration on steering and braking.
- **Intake design:** The intake has an air restrictor that must be placed after the throttle body and before the turbocharger, in accordance with SAE rules. The restrictor is a converging-diverging nozzle that is 19mm at the throat, which is the requirement for E-85 fuel. One of the most significant design issues caused by the single cylinder engine is the fact that it only draws in air for ¼ of the time, resulting in a pulsing effect. The restrictor causes the engine to starve for air with each intake stroke; therefore we added an intake plenum to store a positive charge of air in between intake strokes.
- **Exhaust design:** The turbocharger relies completely on exhaust gas to turn the turbine inside of it. The fact that there is only one exhaust pulse per two revolutions of the engine requires a very efficient exhaust design to keep the turbine rotating above its threshold rotational speed to compress the air. If the turbocharger is rotating below its threshold speed, the intake air is not compressed and there is no forced induction.
- **Boost control system:** The design of the turbocharger is such that it produces boost in proportion to the rotational speed of the compressor wheel. In order to prevent the

turbocharger from supplying too much pressure to the engine, we implemented a boost control method that limits boost to a safe level. On the Garrett GT15V (discussed later), this is accomplished with the use of variable geometry vanes in the turbine which limit the effectiveness of the turbine in converting exhaust energy into rotational speed. We designed a mechanical control system, rather than electrical, to regulate the intake boost pressure.

- **Lubrication:** Due to the high rpm at which the turbine operates, the turbocharger must have sufficient oil to lubricate the internal components and keep it cool. We used the oil already in the engine and designed a system that was directly integrated into to existing oil circuit to supply the turbocharger.
- **Intercooling effects:** Compressing air also increases the temperature of the air. When the air going into the engine is too hot, pre-ignition or detonation can occur which lowers the efficiency of the engine and can cause serious damage. To prevent this, an intercooler can be installed after the compressor of the turbocharger. However, we decided that this was not needed since we chose to use E-85 fuel with its cooling properties.
- **Weigh less than 20 lbs:** The entire system cannot increase the weight of the naturally aspirated engine system by more than 20 lbs. Increasing the power of the engine is not beneficial if the method used to do so will increase the weight of the car to the point that the power to weight ratio is not increased. The power to weight ratio is the best way to measure the success of the project.
- **Engine temp. under 200°F:** The engine must kept under 200°F in order to ensure that it does not overheat. If it were to become too hot, power output would decrease and the risk of engine damage would significantly increase.
- **Engine durability:** Heavy duty components were installed in order to withstand the increased power output. Research has shown that the most common parts to fail on the WR450 engine are the clutch, connecting rod, piston, and head gasket. To decrease the risk of severely damaging our engine, we replaced all the components mentioned above.
- **Cost under $1000:** Formula SAE is always running on a tight budget and therefore costs must be kept to a minimum where possible.

An outline of these objectives is listed below in Table 3.

Table 3. Technical Specifications to Outline the Project

Specification	Requirement Target	Tolerance	Risk*	Compliance**
Power	60 hp	-5, +20	M	A,T
Torque	35 lb-ft	-5, +10	M	A,T
Exhaust	45 cm behind rear axle, 60cm above ground, 110dB	Max	L	T,I,S
Boost Control	See below		H	T,S
Cooling	200°F	Max	L	T,I,S

Weight	20 lbs.	Max	H	A,I,S
Life	50 hours	Min	M	T,I,S
Cost	$1000	Max	H	A,I

*Symbols: H (high), M (medium), L (low)
** A (analysis), T (testing), I (inspection), S (similar design)

3.2 Risks and Verification Methods

3.2.1 POWER
To measure the success of the turbocharger system, we used Formula SAE's dynamometer and Dynomax software to obtain accurate power and torque output values. Intake and exhaust design, as well as fuel and ignition tuning, played a large role in power and torque output.

3.2.2 TORQUE
We aimed for a flat torque curve in the usable range of engine speeds with a decrease close to redline in order to minimize shifting, maximize tractive effort, and have linear power delivery. If the power delivery is not linear, the car can become unstable in corners where it is important to provide the tires with gradual power. A flat torque curve also minimizes the number of shifts needed to complete a lap and therefore reduces the risk for driver error.

3.2.3 EXHAUST
For competition, the exhaust must meet noise and height requirements to be in accordance with SAE regulations otherwise the FSAE team is penalized. The placement of the exhaust will be determined by the FSAE team when they determine how the turbocharger system will be packaged into the car. The turbocharger partially muffles the exhaust noise, but a muffle is still necessary to meet noise regulations.

3.2.4 BOOST PRESSURECONTROL
Boost pressure on the GT15V is controlled by the variable nozzle turbine (VNT) design, discussed later. It is important to have a control system that keeps the vanes open as much as possible in order to minimize back pressure and maximize overall efficiency, yet will close the vanes proportional to the demand for power. Finally, the boost control must override the pedal position and open the vanes the required amount in order to not exceed the desired maximum boost. Secondary design goals include preventing the vanes from sticking and maintaining, as closely as possible, the "stock" feel of the accelerator pedal.

3.2.5 COOLING
At steady state, the coolant of an engine is usually about 2-3 degrees cooler than the engine block itself. Coolant temperatures above 200°F drastically increase the probability of engine failure, pre-detonation, or component damage. To verify this specification, we used a temperature sensor at the top of the radiator tank, since that represents the temperature of the coolant when it leaves the engine.

3.2.6 WEIGHT
For the 2013 Formula SAE car, one of the main goals for the team is decreasing the weight of the car. Since FSAE is our sponsor, we decided to align our goals with those of the team, so we aimed to have the turbocharger system weight under 20 pounds. To determine the overall weight of the turbocharger system, we weighed each component individually.

3.2.7 LIFE
At FSAE competitions, about half of the teams do not finish all events due to issues with their cars. This proves how important it is for the car to be reliable, which is why reliability was set as one of our goals. We aimed to have 50 hours of run time without an engine rebuild. To verify this we documented how much time the engine was run during testing.

3.2.8 COST
Formula SAE is funded by the ME department at Cal Poly, sponsorships, and donations. However, lack of funding usually results in a very tight budget for the car. As our sponsors, FSAE had to allocate some of their already menial budget to fund our project. Therefore, our goal was to keep the cost of our project under $1000 to Cal Poly FSAE. Otherwise, we risk spending money that was budgeted for other parts of the car, which might decrease system reliability.

3.2.9 QUALITY FUNCTION DEPLOYMENT
Additional requirements are located in the Quality Function Deployment (QFD) graph in Appendix A. QFD is a design technique where measurable objectives are weighted against the customer's desires in order to decide which objectives are the most important to focus on. The most important requirements in the QFD are to meet the FSAE rules; otherwise the car will not be able to compete.

4.0 Design Development

4.1 Simulation – Ricardo WAVE

In order to determine the optimum design for the intake and exhaust components we used a program called Ricardo WAVE. From Ricardo's website: "WAVE is the market-leading ISO approved 1D engine & gas dynamics simulation software package from Ricardo Software. It is used worldwide in industry sectors including passenger car, motorcycle, truck, locomotive, motor sport, marine and power generation. WAVE enables performance simulations to be carried out based on virtually any intake, combustion and exhaust system configuration, and includes a drivetrain model to allow complete vehicle simulation." [8]

We did not use the drivetrain model in our simulation. The advantage to using a simulation program like WAVE is that many design iterations of a component can be performed without physically building and testing them. WAVE saved us a significant amount of time, money, and materials by optimizing the system design before dynamometer testing.

WAVE is an extremely powerful tool capable of producing results so accurate that major manufacturers all over the world use it to design engines before they ever build one. This accuracy is solely dependent on the accuracy of the computer model in relation to the actual engine. Each geometry must be properly measured and input into WAVE or the result will be irrelevant. We measured many parameters of the WR450 engine and used data given from Garrett to construct a reasonably accurate model. Greater accuracy could be obtained through the use of actual flow bench data from the cylinder heads instead of using default values within WAVE and through more time fine tuning the tube wall heat transfer and friction coefficients. The model used is shown in Figure 6.

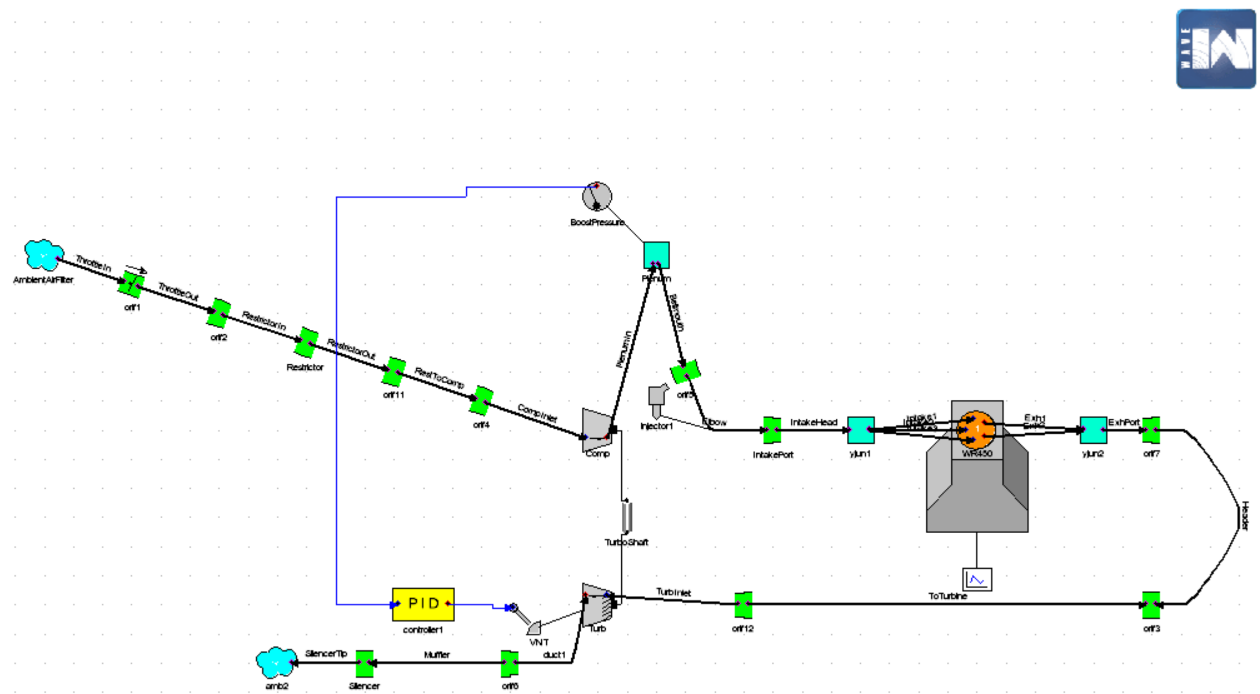


Figure 6: Ricardo WAVE model of WR450 Engine

Camshaft profiles, turbocharger properties, and other values used in the simulation are in Appendix J. Camshaft profiles were measured using a dial indicator and a degree wheel. One point to note is that both simulations used the camshaft profiles as shown, but the turbocharged engine has the exhaust camshaft timing advanced by 22.5 degrees in order to reduce the amount of overlap. This equates to rotating the camshaft gear 1 tooth counterclockwise in relation to the camshaft chain on the real engine. Camshaft overlap is when both the intake and exhaust valves are open at the same time at the end of the exhaust stroke. While this is beneficial to a naturally aspirated engine due to the scavenging effects and longer power stroke, it is undesirable on a turbocharged engine because the exhaust pressure is greater than the intake pressure, which prevents fresh charge from entering the cylinder and can even cause reversion. It is a common modification on WR450 engines to retard the exhaust camshaft timing by one tooth in order to realize these benefits, and this is what was done on the naturally aspirated engine for FSAE. Since the engine is now turbocharged it is more beneficial to return the exhaust camshaft to its original timing so as to minimize the amount of overlap.

Since an internal combustion engine is just that, combustion events cannot be directly observed. We do not know the exact patterns for how the fuel burns during each cycle, how much of the work produced is lost to friction, or how much heat transfer occurs within the cylinder. Fortunately, WAVE has built-in models for approximating these events. The Weibe combustion model, the Chen-Flynn friction model, and the Woschini heat transfer model provide valuable approximations for events that are extremely difficult to measure. The values used in the Weibe, Chen-Flynn, and Woschini models are also shown in Appendix J.

When a model is constructed to simulate results, it is extremely important to verify that the model is accurate before it can be used for design decisions. Since there the turbocharged engine is not built yet, the only option is to validate the model using data from the naturally aspirated (NA) engine. Figure 7 shows the results from the simulation with the results measured on the engine dyno for the 2012 naturally aspirated engine.

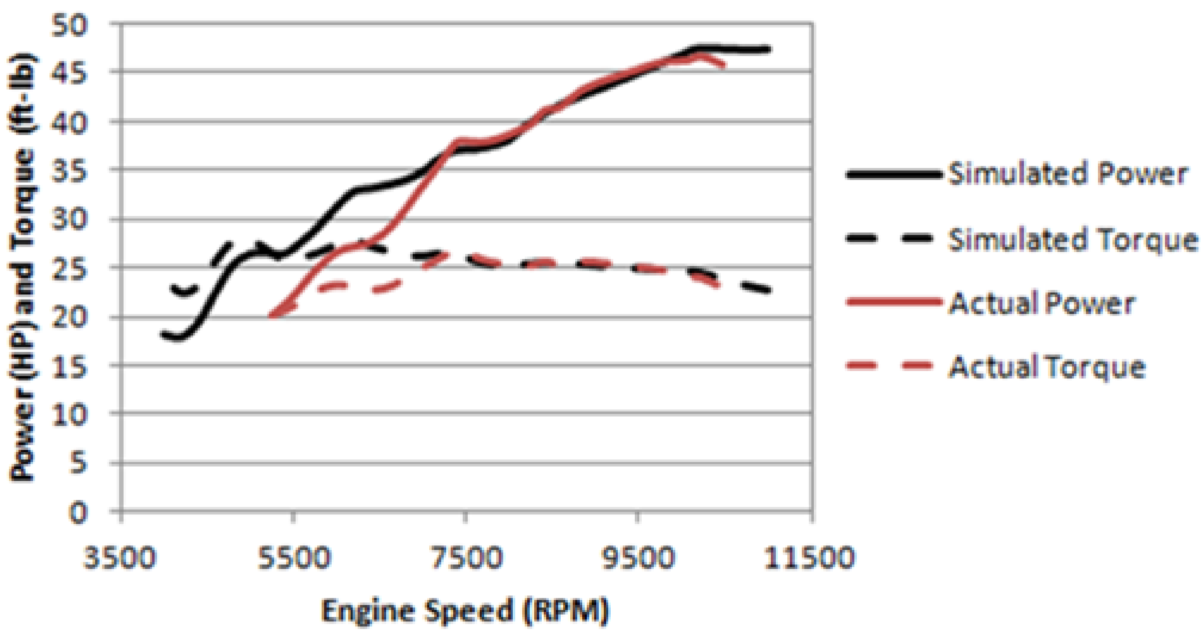


Figure 7: Ricardo WAVE simulation validation

The results from the validation show that the model accurately predicts the performance of the engine. Some ranges are slightly different, especially the lower RPM range, but the results are plenty close enough for the model to be used to base design decisions.

4.2 Turbocharger Selection

Since the decision had already been made that turbocharging the car would be the most valuable, our goal was to determine the best turbocharger for our system. With numerous manufacturers and multiple types of turbochargers, this seemed like it would be a major part of this project. However, our task was made much simpler after the discovery of Honeywell's FSAE Sponsorship Program. We found that Honeywell would provide a turbocharger free of charge to any FSAE team that wanted to implement forced induction onto their vehicle. All we had to do was to provide supporting calculations to show that the turbocharger would in fact increase our performance. After contacting Honeywell, they provided us with drawings and specifications on the two different turbochargers that they were willing to provide, one being the GT12-41 and the other one being the GT15V. The main difference between the two, other than a slight difference in size, is that the GT15V has a Variable Nozzle Turbine or VNT.

4.2.1 TURBO BASICS

To understand the benefits of the VNT, one must first understand the aspect ratio, which is the cross-sectional area over the radius of the turbocharger. The aspect ratio is also known as the A/R ratio. To understand this concept, it is best to examine a fixed geometry turbocharger or FGT. With a FGT, the cross-sectional area and the radius used to determine the aspect ratio are products of the turbocharger geometry. This results in a fixed aspect ratio that is constant for a given turbocharger. The aspect ratio remains constant for a given FGT because as the radius increases so does the cross-sectional area resulting in a constant ratio of the two values. Figure 8 shows a FGT with a constant aspect ratio.

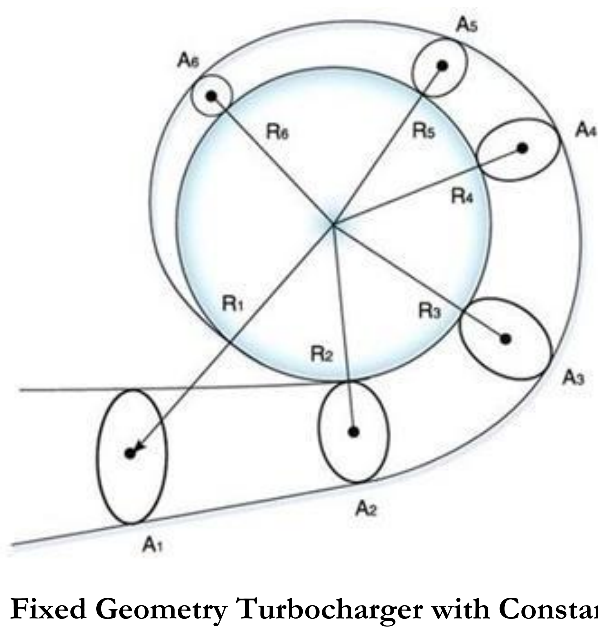


Figure 8: Fixed Geometry Turbocharger with Constant A/R Ratio

From junkyardturbos.com

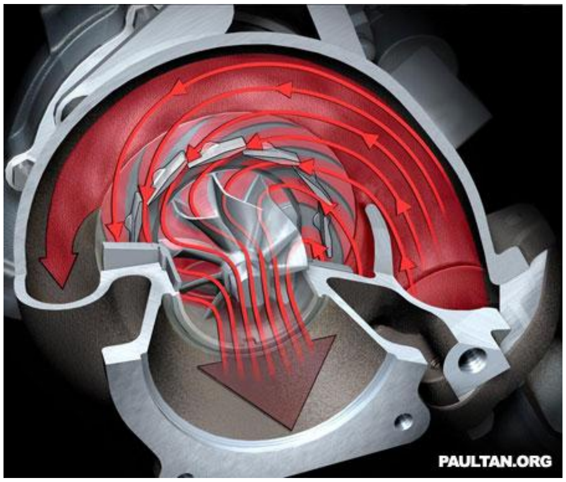
The importance of the aspect ratio is that it characterizes a turbocharger with a single value. A large turbocharger will have a large aspect ratio and a small turbocharger will have a small aspect ratio. A larger turbocharger will take more exhaust gas energy to produce boost and therefore will produce almost no boost during low engine speeds. This means that a larger turbocharger will have a higher boost threshold, which is the minimum engine speed required to produce boost, than a smaller turbocharger. However, above the boost threshold, a larger turbocharger will produce more power than a smaller one since the engine can pull more air through the turbocharger. In other words, a larger turbocharger will not restrict flow as much as a smaller one which results in higher power production from the engine. A small turbocharger will not require as much exhaust gas energy to produce boost pressure, therefore it will produce more boost during low engine speeds and will have a lower boost threshold resulting a quicker response time than a larger turbocharger. However, at high engine speeds the small turbocharger will restrict the airflow into the engine so that the engine will not be able to produce as much power as when it had a large turbocharger.

To summarize, an advantage for a small turbocharger is it will have a lower boost threshold so it will produce boost quicker and during low engine speeds but its disadvantage is it will choke the airflow during high engine speeds. While a disadvantage of a large turbocharger is it will have a higher boost threshold so it will respond slower and produce very little if any boost at low engine speeds but its advantage is it will not restrict flow into to the engine at high engine speeds.

4.2.1 VARIABLE NOZZLE TURBINE

Now that the advantages and disadvantages of both a small FGT turbocharger and a larger FGT turbocharger are understood, it is easy to see the overall advantage of the VNT. Basically, it provides the advantages of both the small and large turbocharger without the disadvantages of either. That is not to say that it is not flawed, as it does have its disadvantages which will be explained later.

The VNT effectively act like a small turbocharger during low engine speeds and a large turbocharger during high engine speeds by using a ring of adjustable vanes in the turbine housing that can be pivoted to change the effective cross-sectional area of the turbocharger. Figures 9 and 10 show the effect of adjusting the vanes from closed to open as would occur with an increase in the engine speed.



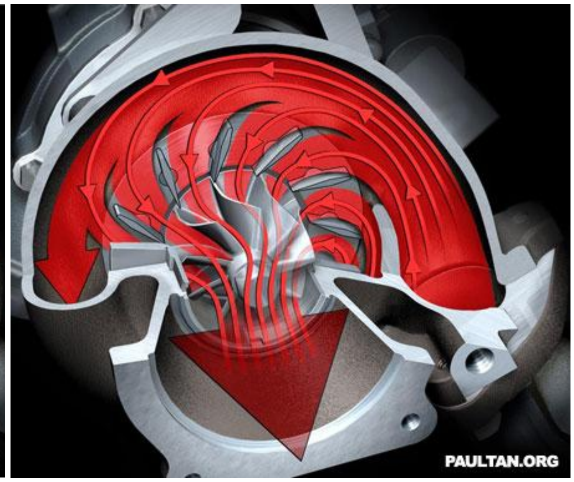


Figure 9. VNT with vanes closed. **Figure 10. VNT with vanes open.**

With the vanes closed, the effective cross-sectional area of the turbocharger is decreased. Therefore the aspect is also decreased and the VNT acts like a small turbocharger. When the vanes are closed the airflow is sped up and it is directed tangentially across the turbine blades. The combination of these two effects is what produces quick response and low boost threshold. When the vanes are open the airflow is not sped up and it is directed more radially to the turbine blades. This allows more airflow through the turbocharger and prevents the turbocharger from producing too much boost. Because the vane position can be varied so that the turbocharger is always operating at the optimal aspect ratio for a given engine speed, the turbocharger becomes much more efficient.

The Variable Nozzle Turbine does have its drawbacks. First, all the exhaust gas passes through the turbine housing (where with a FGT some exhaust gas passes through a wastegate) which results in higher temperatures inside the housing. To account for this increase in temperature inside the turbine housing, VNT turbochargers typically are made of more exotic metal alloys that are more resistant to heat. This results in higher prices for VNT's compared to a similar size FGT turbocharger. Second, the VNT turbocharger is much more complex than a FGT turbocharger and to be more beneficial the vane position must be monitored and controlled. The boost control system for a VNT is more complex than one than that of a FGT which will result in more time and money spent to install and tune this system. Figure 11 shows a VNT and how it operates to adjust the gas flow.

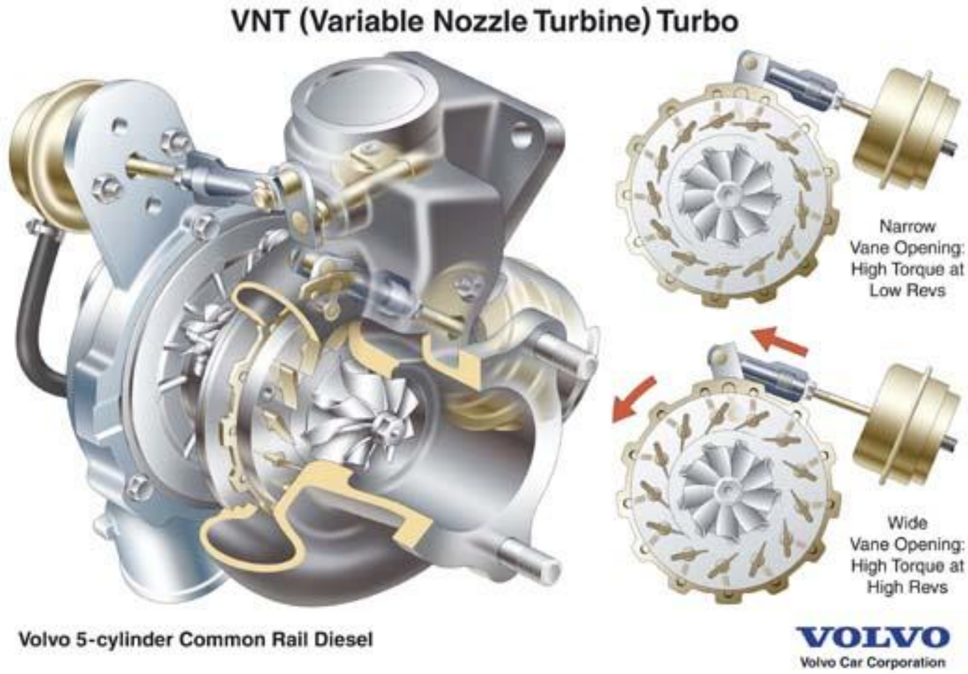


Figure 11: VNT Turbocharger

From www.motorsportscenter.com

To decide whether it would be best to go with the GT12-41 or GT15V turbocharger, we first referred to the "Garrett by Honeywell, 2009" catalog [4]. The first thing to do was to decide on a horsepower target, which for us is 60 hp. Then knowing the Air/Fuel Ratio (8:1) and estimating Brake Specific Fuel Consumption (0.75 lb./hp*hr.) we calculated actual airflow, Wa, using the equation:

$$Wa = HP * \frac{A}{F} * \frac{BSFC}{60}$$

*Note: HP = Horsepower

$\frac{A}{F}$ = Air Fuel Ratio

BSFC = Brake Specific Fuel Consumption

For our and engine, fuel choice, and horsepower target we would need and actual air flow of 6.0 lb./min.

Next, we needed to calculate the required manifold pressure, MAPreq, which is the pressure of the air after it has left the compressor of the turbocharger and just before it enters the engine, to meet the horsepower target. For this we used the gas constant (R=639.6 in/°R), intake manifold temperature (Tm=100°F), Volumetric Efficiency (VE=0.95 for a 5-valve head), engine speed in rpm (N=9000), and engine displacement in cubic inches (Vd=449 cc = 27.39 in^3) and calculated the manifold pressure using the equation:

$$MAP_{req} = \frac{Wa * R * (460 + T_m)}{VE * \frac{N}{2} * Vd}$$

*Note: MAP_{req}= Manifold Pressure Required
Wa = Actual Airflow
T_m = Intake Manifold Temperature
VE = Volumetric Efficiency
N = Engine Speed
Vd = Engine Displacement

For our required target, our manifold pressure needs to be 18.4 psi. However, because of pressure drops due to the plumbing from the turbocharger to the intake of the engine, the discharge pressure of the compressor has to be greater than the manifold pressure. We calculated the temperature drop due to the fuel vaporizing upon injection and found that with E-85, an intercooler is not necessary to sufficiently reduce the intake charge temperature. Typical pressure drops from the discharge of the compressor to the manifold are about 2 psi with an intercooler, but since an intercooler is not needed for our application we estimated a 1 psi drop. Therefore, the exit pressure of the compressor needs to be 19.4 psi.

We now need to know the pressure at the compressor inlet in order to calculate the pressure ratio for the compressor maps. This is simply the atmospheric pressure, 14.7 psi, minus pressure drop due to the air filter, throttle body, and restrictor leading into the compressor inlet, typically 1 psi without the restrictor. We also took into account the fact that we will not be operating at sea level. The pressure of 14.7 psi only refers to sea level elevation, but in Lincoln, Nebraska, where the Formula SAE competition is held, the elevation is 1189 ft. and the atmospheric pressure is 14.1 psi. With the pressure drop our compressor inlet pressure is 13.1 psi, but the 19mm restrictor drops the pressure by 15%, so it is actually 11.6 psi.

Now we get the ratio of the exit to inlet pressure (pressure ratio) of the compressor, which for our application is at 60 hp is 1.67.

Finally, we need to know the actual air flow for max torque using the same variables as mentioned earlier, except our engine speed (N) is now 6000 rpm for max torque, with the equation:

$$(Wa)_t = \frac{MAP_{req} * VE * \frac{N}{2} * Vd}{R * (460 + T_m)}$$

For our application, this lower air flow is 4.01 lb./min. With the actual air flow for horsepower and torque we can plot how far to the right of compressor map we will go and the pressure ratio will tell us high to go up. For the best performance, we need to be as inside of the inner most island of the compressor map as possible. Too far to the left means the compressor is too big to where the boost will fluctuate wildly (surge) which can be damaging to the turbocharger, and too far to the right means the compressor is too small and we won't get the boost we need.

To further confirm which turbocharger to choose we re-iterated our calculations for other possible various other horsepower of 65, 70, 75, and 85 hp. This is because our engine will only be the foundation from which future teams can produce even more power. We want to make sure that the turbocharger we select will be sufficient to provide more power in future years as well. The results are given in Table 4 below.

Table 4: Compressor Map Calculations for Desired HP

Horsepower Target (hp)	60	65	70	75	80
Compressor Pressure Ratio	1.67	1.8	1.93	2.06	2.19
Wa (lb./min)	6.0	6.5	7.0	7.5	8.0
$(Wa)_t$ (lb./min)	4.22	4.55	4.88	5.22	5.55

From the compressor maps in Appendix B, the GT15V puts us closest to the inside of the efficiency islands, which is the reason we chose it. Upon submitting our calculations and turbocharger selection, we were advised by Honeywell that we would be the first Formula SAE team to try to implement the VNT system onto a single cylinder engine.

After completing the calculations and selecting the turbocharger, we needed to make sure that choke flow, which is the maximum mass flow rate downstream of the restrictor, exceeds the required flow rates for our horsepower targets. If the maximum mass flow rate is less than our calculated required value, it means that it is physically impossible for the engine to receive enough air to make the desired power. Given the conditions in Lincoln, Nebraska (the location of the FSAE competition), choke flow is 8.45 lb./min, exceeding the actual flow rates for all desired horsepower levels. All calculations for turbocharger selection are shown in Appendices D and F.

After approval of our calculations from Honeywell, they sent us the GT15V turbocharger shown in Figure 12.

Figure 12: Turbocharger from Honeywell

4.3 Boost Control

To successfully integrate a turbocharger into an engine system you must accurately control the air flow into the engine. If the turbocharger is providing more airflow than the engine can accept, then extreme pressure can build in the intake manifold and cylinder that will most certainly damage the engine and possibly the turbocharger. There are several ways to control this pressure, commonly known as boost pressure, so that it does not reach damaging levels. However, because we have chosen the Variable Nozzle Turbine (VNT) turbocharger, the responsibility of controlling the boost pressure lays solely on the correct controlling of the vane position. The four types of control systems that are used to accomplish this are: hydraulic, pneumatic, electrical, and mechanical. Most often a combination of two of these types is used to create the control system for the vanes.

Hydraulic control is certainly the least used method of vane control and is usually only used in aerospace applications. This method requires a separate pump to provide the needed pressure of the working fluid, usually engine oil, to control the vane position. Figure 13 shows the hydraulic routing needed to operate a hydraulic control system.

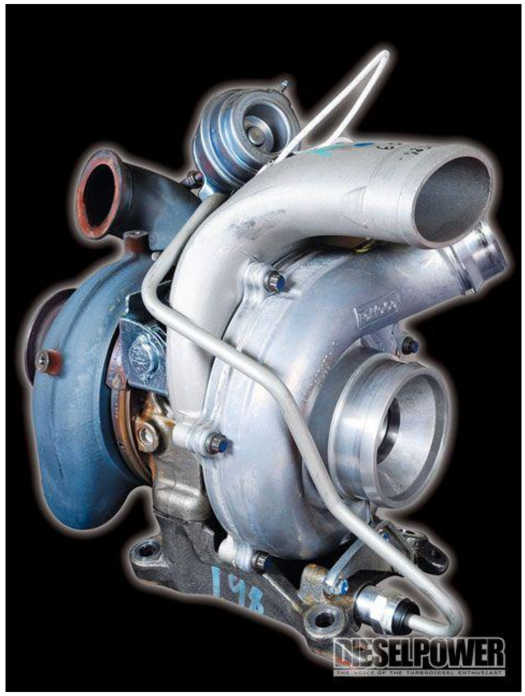


Figure 13: VNT Turbocharger with Hydraulic Vane Actuation

From Dieselpowermag.com

Pneumatic control is the most common control method in the automotive industry. A separate pump may be used, but is not always necessary, to provide positive or negative pressure depending on the type of actuator that is used. Either a vacuum actuator or a positive pressure actuator can be used and the air pressure can be taken from the intake or exhaust manifold so that a secondary pump is not needed.

Electrical control is very common in the automotive industry, mainly in high performance or race applications. Most electrical control systems are integrated into a pneumatic or hydraulic system to produce quick response and additional benefits like enhanced boost settings and tuning.

Mechanical control is the simplest way to control the vane position. This method takes advantage of the fact that the vane position is directly proportional to the throttle position. A linkage system with a spring is utilized to tie the throttle to the vane actuator. Figure 14 shows a mechanical linkage that is used to control the vane position.

Figure 14: Mechanical Boost Control Linkage

From Dieselpowermag.com

Based on our understanding of these systems, we have chosen to use a pneumatic control system. We chose to use the pneumatic system over the other possibilities mainly because of the simplicity of the system and the feasibility of successfully implementing it within the timeframe of our senior project. We will leave the optimization of this control system to a future senior project team or the FSAE team. For now, we have decided that the added benefit of an electric or hydraulic system is not worth the extra time or money.

4.4 Fuel System

One of the first decisions our team needed to make was which fuel to use to power the turbocharged engine. We needed to make the fuel type selection first because it would dramatically affect the calculations that would need to be done in order to correctly size a turbocharger for the engine. It would also affect the decision of what components we were going to need to buy to complete the turbocharging system. For example, the fuel injector that is currently on the car cannot be used with ethanol fuel. Many factors came into play in this decision, and they are all shown in the decision matrix in Table 5 on the following page.

Table 5: Fuel Choice Decision Matrix

		Fuel Type		
Criteria	**(Weight)**	**E-85**	**91 Octane Gasoline**	**97 Octane Gasoline**
Availability (Testing)	2	2	5	4
Availability (Competition)	5	5	5	3
Cost	4	3	4	1
Power Output	5	5	2	3
Weight	5	5	1	3
Fuel Efficiency	3	3	3	3
Compatibility	4	4	5	5
"Forgiveness" in tuning	4	4	1	3
Uniqueness	4	4	1	2
TOTAL		**148**	**103**	**106**

The largest factors weighing in on the fuel selection were the availability of the fuel at competition, the potential for safe power output, tunability, and the weight the fuel type would contribute to the car. Since the competition organizers provide the fuel and teams cannot bring their own fuel to run, selection is limited to what the organizers provide. Historically the higher octane race gas has been in short supply at competitions. Cal Poly has run into problems in the past with tuning an engine for race gas and then being stuck with pump gas at competition, causing reliability issues. E-85 has always been provided and is another high performance alternative to pump gas. Since reliability is one of the major design considerations we must tune for the fuel that we know the car will be running at the competition. The following paragraphs outline the fuel decision process in depth.

4.4.1 FUEL PROPERTIES

In order to begin the decision process, it was crucial that relevant properties of each fuel be fully understood.

Table 6: Properties of Various Fuels [6]

	FUELS		
	Ethanol	**Gasoline**	**E-85***
Density (kg/m^3)	770	700	759.5
Lower Heating Value (Q_{LHV}) [MJ/kg]	26.8	44	29.4**
Latent Heat of Vaporization [MJ/kg]	0.84	0.35	0.77**
Stoichiometric AFR	9	14.7	9.9
Ideal AFR	8	13.23	8.5

*Perfect mix (85% Ethanol, 15% Gasoline) assumed.

** Reliable values could not be found. Values interpolated with mixture quantity.

Density plays a role in determining how much volume of fuel is going to be required. For example, if we need equal mass amounts of gasoline and E-85 is going to take up more space on the car.

Lower Heating Value represents the amount of energy released from the complete combustion of one unit mass of fuel. Notice that E-85 has a lower value – implying that for a given amount of power output, 1.5 times the mass of fuel must be combusted (ideally).

Latent Heat of Vaporization is the amount of heat that the fuel takes from its surroundings in a constant temperature phase change from liquid to vapor.

The **Stoichiometric AFR**, as aforementioned, is the ratio of mass of air to fuel needed for a complete combustion of the fuel.

The **Ideal AFR** is the air to fuel ratio recommended for peak power output.

Another property that plays a large role in hardware choice is corrosiveness. Although E-85 itself isn't much more corrosive than gasoline, its chemical properties cause it to attract water. This in turn accelerates oxidation in fuel lines, pumps, tanks, etc. Although this isn't a large issue during engine operation because the fuel is constantly moving through the system, it can potentially damage components if left sitting for an extended period of time. [7]

4.4.2 FUEL CONSUMPTION

Using the ideal air fuel ratios, we can calculate the mass of fuel required for peak power output at varying engine speeds. At competition, the richer mixture necessary for vehicles running on E-85 is compensated using a correction factor of 1.3, as shown in the following figure.

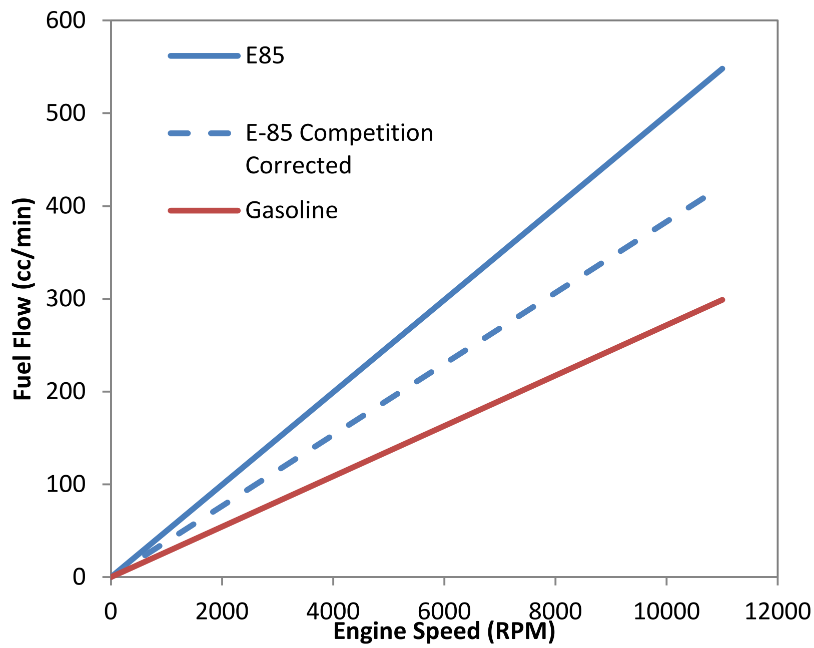


Figure 15: Theoretical WR450 Fuel Consumption with Various Fuels

The figure above is used as a visual representation of how fuel flow is affected by different AFRs, but will not necessarily reflect values seen during testing. The main reason is because the flow rates shown above assume a constant volumetric efficiency of 95% across the entire engine speed range, but that's not true with actual operation. Also, the volumetric efficiency with forced induction can easily exceed 100%.

This model was mainly used to get ballpark numbers on the difference in the amount of fuel the car would need to complete the endurance race. In order to calculate these numbers, it was assumed that the car would be operating at an average of 8000RPM for 30 minutes. With that, gasoline needed a 1.75 gallon tank resulting in a total fuel weight of 10 pounds. E-85, on the other hand, necessitated a 3 gallon tank with 18.5 lbs of fuel. That resulted in a weight increase (excluding the extra material for larger gas tank) of 8.5 lbs.

Due to the lower heating value of each fuel, 18.5 lbs of E-85 will theoretically produce 427MJ of energy, while 10 lbs of gasoline will provide 200 MJ. This results in 85% more weight in fuel for 24%

more available energy. Although the extra weight is detrimental to our objectives, the following paragraphs will outline how that weight could possibly be negated.

4.4.3 COOLING EFFECTS

Another property we must take into consideration is the latent heat of vaporization of each fuel. In order to analyze this, a simple thermodynamic model was created with hot air entering the control volume (intake runner), liquid fuel sprayed into the duct, then the combined fuel and air exiting the control volume (into the engine). Assuming that the intake runner is perfectly insulated and all the energy from the intake charge is used to vaporize the fuel, the temperature drop of the intake was calculated.

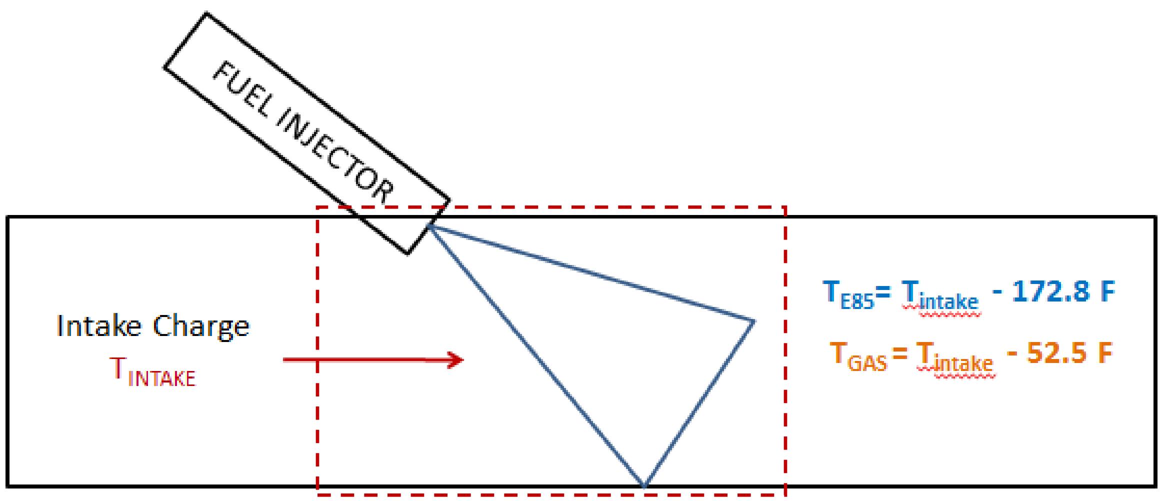


Figure 16: Theoretical Temperature Drop Across Injector

The figure above shows that the temperature of the intake charge after the fuel spray drops 172.8 °F with E-85, and 52.5 °F. Note that this temperature drop is independent of initial temperature of the intake charge. Some general temperatures we expect to see from the compressor outlet are 200-300°F, so cooling the intake charge is extremely important.

A cooler intake charge is vital because it decreases chance of pre-detonation and knocking, and allows a denser mixture into the engine (more mass of air and fuel) allowing for more power output. Although it is possible to cool the intake charge with gasoline, it would be necessary to implement an intercooler. This would increase both packaging issues and system weight by 5-10 pounds.

4.4.4 POTENTIAL POWER

An important aspect of fuel selection is what operating point it allows the engine to run at. Operating point refers to maximum allowable compression ratio and boost pressure before pre-detonation occurs. To study this relationship, the following figure is introduced on the following page.

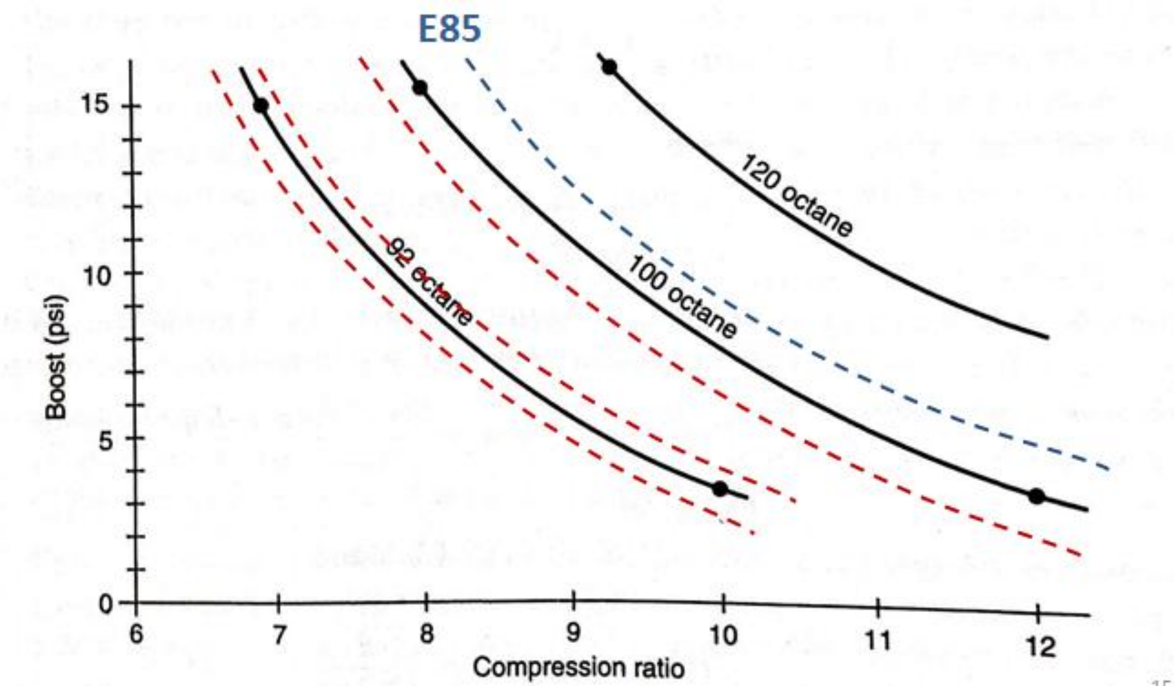


Figure 17: Recommended Limits for Compression Ratio and Boost Pressure [5]

Finding an operating point with a balance of high compression ratio and high boost pressures is desirable. The high compression ratio lets the engine produce as much of its own power as possible without the need for high boost pressures, increases efficiency and improves low-end torque. However, we also want the power increase associated with increasing boost pressure. The three red, dotted lines represent the different octanes of gasoline available at the FSAE competition. Notice that at a given compression ratio, E-85 always has a higher boost pressure limit. Furthermore, the properties of E-85 give us a larger margin of error while tuning around the upper limit.

4.4.5 TRADE OFFS / CONCLUSION

Each of the available fuels has advantages and disadvantages. With gasoline, we get lower fuel consumption, better local availability, and higher compatibility with the current FSAE engine system. However, E-85 negates the need for an intercooler, has higher potential for power output, increased tuning margin, and more design points. Those are the main reasons we chose to fuel our engine with E-85, but in doing that we have to accept certain disadvantages. The corrosiveness of the E-85 will clog the fuel injector if left sitting for more than a week and may damage internal fuel system components. We can address this by running gasoline through the fuel system if the engine will not be run for a long period of time. Its tendency to attract water necessitates a sealed, plastic or stainless steel drum that cannot be exposed to air and must be stored in a cool, dry area. Also, the nearest E-85 gas station is in San Jose, which is a 4 hour round-trip from San Luis Obispo. Since we don't want to make regular trips due to the expense of gas and time, we plan to store 20 gallons of fuel as carefully as possible to extend its useful life.

4.4.6 COMPONENT SELECTION

To address certain disadvantages of E-85, alcohol-compatible components were chosen.

For fuel supply, we chose an Aeromotive in-line fuel pump (MN #11109). After talking with an Aeromotive technical representative, the 11109 was recommended for our application. Although only 66.2 lb/hr of fuel is needed at peak operation, this pump can supply 325 lbs/hr @ 45psi – but it was their smallest model. Aeromotive has a large selection of alcohol-compatible components and the team has had good experience with their products currently used in the FSAE fuel system.

Figure 18: Aeromotive Fuel Pump and RC Engineering SH4-750 Fuel Injector

From aeromotiveinc.com and rceng.com

With the fuel flow estimates in Figure 15, the estimated fuel flow was 550 cc/min. The RC Engineering SH4-750 fuel injector was chosen because its high flow rate (750cc/min), compatibility with hardware, and technical support. Although the FSAE team has had issues with the aggressive spray pattern that these injectors have, they sent us an injector free of cost. Testing will be done to see if the spray pattern provides adequate fuel atomization for low-end response.

The Aeromotive 13105 fuel pressure regulator used on the current car will be used because it is also alcohol compatible, and covers the operating pressures needed by the fuel injector.

To connect the entire system together, Aeroquip Teflon stainless steel hose was chosen. The Teflon walls of this style hose (PTFE) are far more resistant to the alcohol than traditional rubber hoses and also allow for tighter bends without causing kinks. Coupling them will be JEGS 6-AN fittings.

Figure 19: JEGS Hose End and Aeromotive Fuel Pressure Regulator

From www.jegs.com and www.aeromotiveinc.com

4.5 Oiling System

For the oiling system of the turbocharger, oil has to be teed off from a suitable location of the engine where there is adequate pressure for the oil to flow through the turbocharger bearings. The oil must be below the maximum temperature specified by Honeywell.

According to "Maximum Boost," a good rule of thumb is to have a minimum oil pressure of 5 psi (0.34 bars) at 0.1 gal/min at idle and 25 psi (1.72 bars) at 0.5 gal/min at maximum load going into the turbo. The GT15V, however, requires a pressure of 0.6 bar at idle, 1.7 bar at peak torque, and 5 bars at the rated speed. Oil temperature is usually one of the biggest issues that would cause turbocharger failure as coking occurs due to insufficient flow. This usually happens when the engine is shut off immediately after a very long, hot run and oil flow is stopped. When this oil flow stops the turbine and compressor are still spinning at a very fast rate heating up the oil hot enough for it to coke and destroying the turbocharger. To possibly prevent oil coking in the turbocharger Honeywell specifies that the oil entering the turbocharger be less than 130 °C (266 °F). If it is found out that the oil temperature is above this, then an oil cooler must be added to the system.

Contaminants and metal shavings from engine wear can also cause catastrophic failure to the turbocharger. The average engine usually has an adequate enough oil filtration system. However, turbocharger specifications will specify the oil filter size. If the OEM filtration system is not to the turbochargers specification an in-line oil filter to the turbocharger can be added. In the case of the GT15V, it requires an oil filter of 15 microns.

To be sure the pressure and filtration requirements are met, the best place to T off the oil is the after the oil pump and filter. Most street vehicles use a spin-on oil filter (Figure 20, left) where a sandwich plate can be added with no modifications to the engine block that even includes ports for pressure gauges and temperature sensors. However, the WR450 uses a cartridge filter, so this option does not apply to us.

Figure 20: Spin-on Oil Filter and Cartridge Filter

From prosportgauges.com and knfilters.com

A solution to this would be to tee off the oil from the oil delivery pipe that goes from the oil filter to the top of the engine block, highlighted in red in Figure 21 on the following page. This part should have enough pressure for flow and it is also an external interchangeable part that would be a cheap fix in-case a mistake is made during the modification process. To ensure that the oiling system works without any leaks and problems the correct hoses and fitting should be used. These include –3an and -4an fittings and steel braided hoses.

Once the oil has been teed off it has to go into the turbocharger at an angle that allows the outlet to be within 20° of vertical with the vehicle on level ground. This will ensure that gravity can drain the oil from the turbocharger sufficiently to keep the oil circulating. After the oil has gone through the turbocharger it will have to be either drained back to the sump or crank case. In this case, oil will be drained back to the case.

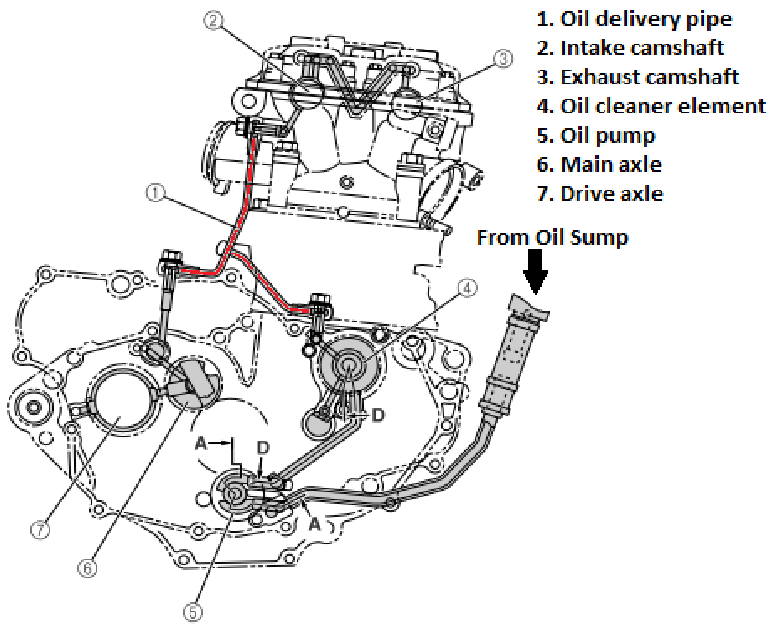


Figure 21: WR450 Lubrication Diagram

Another failure in the oiling system can happen when the oil returning enters below the oil level in the case. This can cause oil to back up into the turbocharger, heat up, and coke up the bearings. If it is not possible to return the oil above the engine oil level, a scavenging pump will be required to ensure that the oil will not back up into the turbocharger. The turbocharger should also not be mounted below the pump. It is extremely important that the scavenging pump is capable of out-flowing the oil leaving the turbo. To ensure that a scavenging pump will not be required, the turbocharger will be located well above the oil level in the engine case [4].

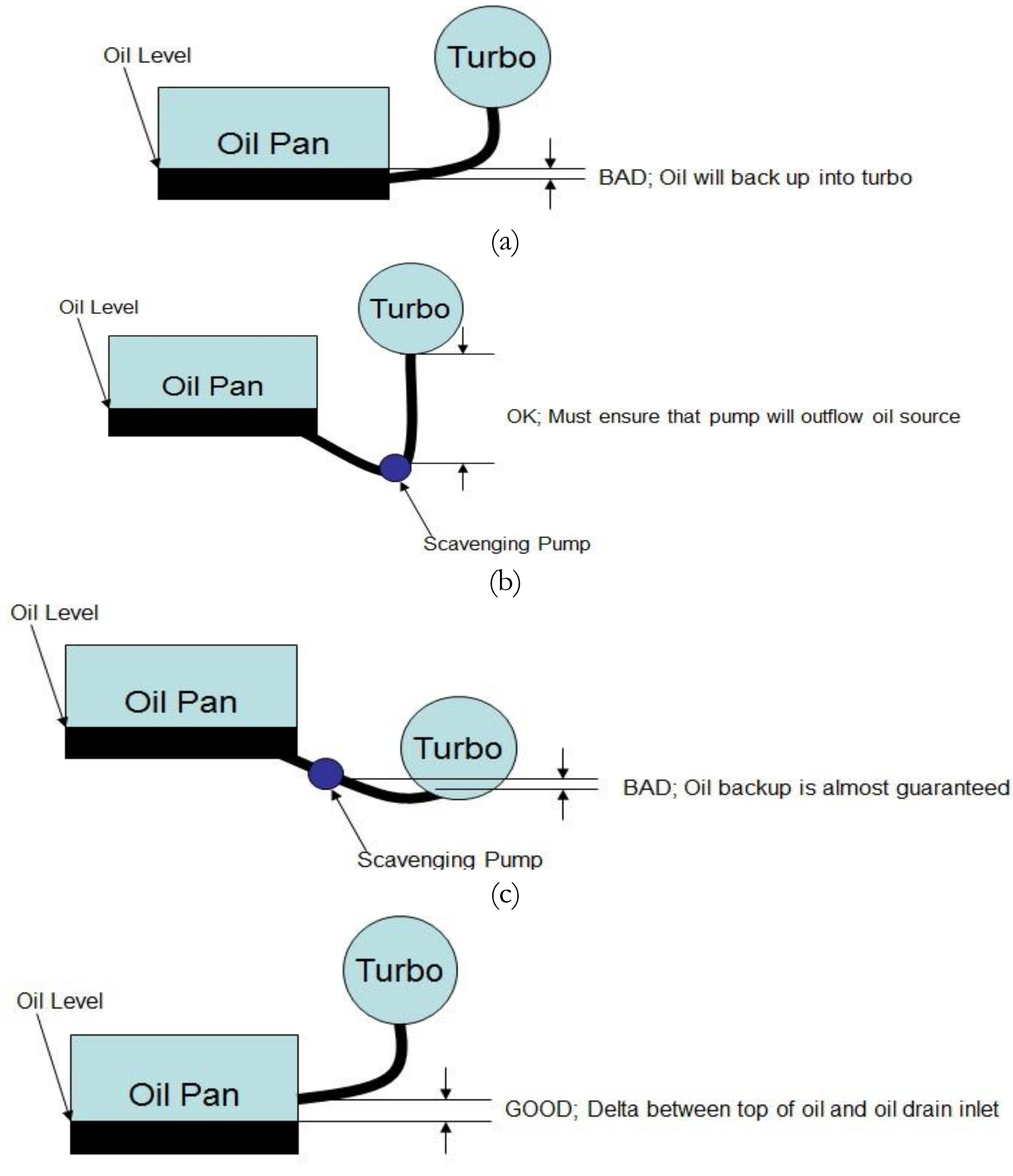


Figure 22: Turbo Oil Drainage Options

Due to the regulations on the order of intake components, there will be some issues concerning the compressor oil seals. The problem arises due to the throttle being placed ahead of the compressor. This creates a significant vacuum under certain conditions such as idle and quickly letting off the throttle. In turn, oil can be drawn from the center section of the turbocharger, past the compressor seals, and into the intake tract. This oil burning causes the buildup of deposits in the combustion chamber and can even cause oil starvation in the crank case if enough is burned. To prevent this, Honeywell has suggested a style of positive crankcase ventilation to keep the pressures in the compressor and oil return line equalized. With this setup, the entire crank case is under vacuum so that the oil is sucked into the drain line instead of the intake. In order to comply with FSAE rules, the crankcase vent line must first be routed to a catch can and the connection to the intake must be located ahead of the restrictor as shown.

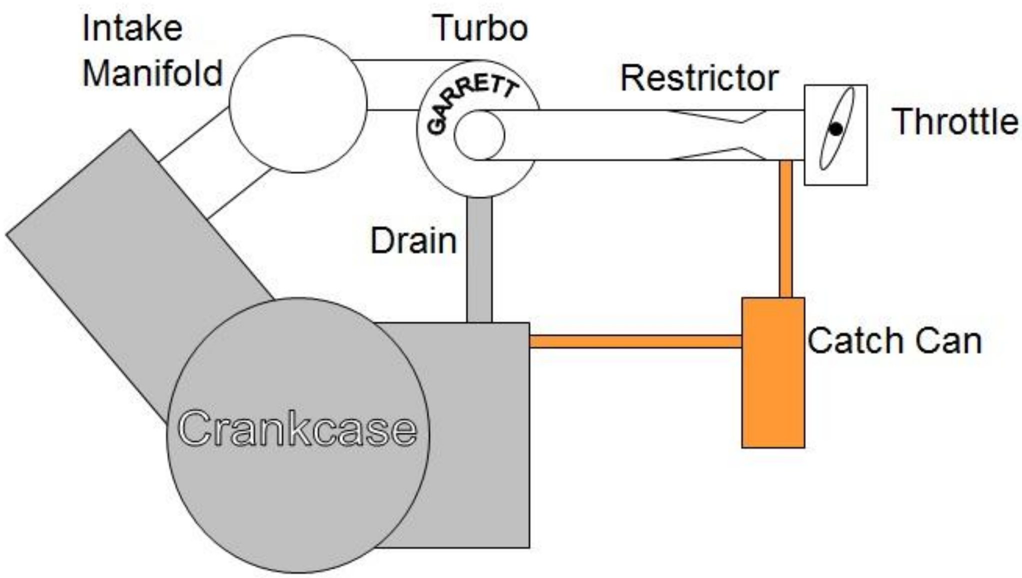


Figure 23: Suggested PCV routing from Honeywell.

4.6 Exhaust System

The primary goal of the exhaust system is to maximize the amount of energy that comes from engine to power the turbo. This allows for quicker spooling of the turbo, and therefore the ability to reach desired boost pressure at lower engine speeds. This translates to more available torque in the lower rev ranges, requiring less shifting. The two main ways to maximize the energy feeding the turbo are by minimizing energy loss from the engine to the turbine and by taking advantage of resonance tuning.

4.6.1. MINIMIZING ENERGY LOSS

Energy is lost through the exhaust mainly through head loss and heat loss. Head losses are losses associated with bends in tubing, rough walls, and any obstruction to flow. In order to minimize this, smooth bends will be used during manufacturing. In our case, we define a smooth bend as one with an R/D (radius of bend/diameter of pipe) ratio of 2 or above. Rough walls can be addressed by using a material with a relatively smooth surface finish, and by smoothing out any manufacturing defects such as weld beads on the inside. A constant diameter pipe from the engine to turbine will also be used.

To minimize heat loss, surface area of the exhaust can be reduced by shortening the length, or insulation can be used. Packaging constraints don’t allow for excessive shortening of the exhaust, so insulation will be implemented. Two options we have are exhaust wrap or ceramic coating, or a combination of the two. At this point, we have not decided on the exhaust insulation type. Another consideration is that thicker tubing walls allow for more heat transfer, so finding a balance between strength and small wall thickness will be necessary.

Figure 24: Ceramic coated exhaust and wrapped exhaust manifold.

From images.gizmag.com and precisionenginetech.com

4.6.2. RESONANCE TUNING

When the exhaust valve opens, the high pressure inside the combustion chamber drives a positive pressure wave that is sent through the exhaust pipe until it hits an obstruction – the turbine nozzle in our case. The wave bounces back towards the engine and then hits the exhaust valve. Although resonance tuning is normally used to increase power in the intake manifold by taking advantage of the pressure waves, our goal is to avoid these pressure waves. If a positive pressure wave comes back to the exhaust while it is open, then it pushes exhaust gases back into the cylinder – decreasing volumetric efficiency and power. The key to this is timing the pressure wave to hit the exhaust valve at the right moment. To accomplish this we needed the profiles of the camshafts that we were using. We set up the engine on a mill, turned the engine a few degrees at a time, measured the angle of rotation, and used a dial indicator to measure how far the camshaft was pushing down on the valve bucket. The following figure shows the results.

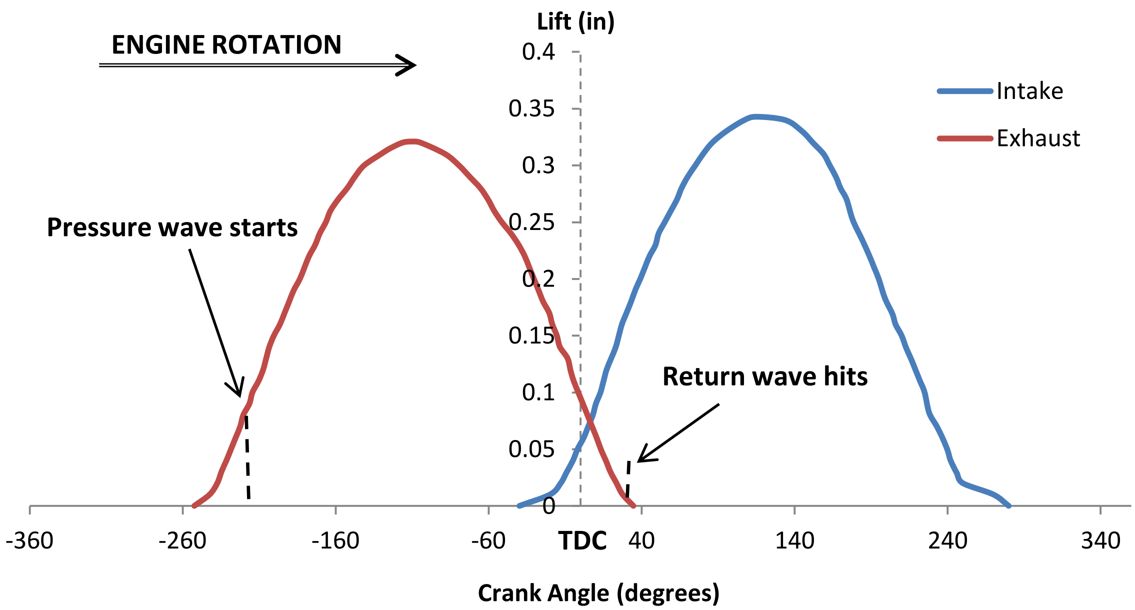


Figure 25: WR450 Camshaft Profile and Resonance Tuning

In order to do the calculations, we assumed that the pressure wave started when the camshaft was about 1/3 of the way open. From the figure above, we can see that the pressure wave starts at about 220 degrees BDTC, and should return at 34 degrees ATDC. This means that the engine has to rotate 254 degrees to be in the ideal position.

$$N\left(\frac{Rev}{min}\right)\left(\frac{360\ deg}{1\ rev}\right)\left(\frac{1\ min}{60sec}\right) = 6N\left(\frac{deg}{sec}\right)$$

N = Engine Speed (RPM)

Now we find the time it takes the engine to rotate 254 degrees.

$$time = \frac{distance}{velocity} = \frac{254\ deg}{6N\ deg/s} = 42.3N\ (seconds)$$

Since pressure waves travel at the speed of sound, that speed must be calculated in hot, exhaust conditions.

$$c = \sqrt{kRT} = 330 \text{ ft/s}$$

c = speed of sound (ft/s)

k = Ratio of specific heats = 1.4 (for air)

R = Gas constant for air = $53.34\ \frac{ftlbf}{lbmR}$

T = Temperature of air = 1000°F +459.67 = 1459.67 °R

Since velocity and time are known, length can now be calculated.

$$Velocity = \frac{Distance(Length)}{Time}$$

$$2 * Length = Velocity * \frac{Time}{n}$$

$$Length = \frac{c_{sound}t}{2 * n}$$

The two in the denominator accounts for the fact that the wave has to travel a round trip to get back to the engine. Since the calculated length is sometimes not realistic, we can let the wave take another round trip before it hits the ideal location, or have the wave come back every other revolution. The number of round trips is represented by the variable n in the denominator. From these equations, we will study the lap-simulator developed by FSAE to choose an engine speed for which to tune the exhaust.

4.6.3. MATERIAL/STRESSES

The exhaust system can see heat cycles room temperature to 1600 °F, so it is important to consider the effects of thermal expansion. The exhaust shouldn't be fully constrained by supports, otherwise high stresses will be induced with expansion. Ideal characteristics of the chosen material are low thermal conductivity and coefficient of thermal expansion, relatively high strength, resistance to cracking, machinability and cost.

The two options that best fit the criteria are mild steel and stainless steel. Stainless steel is weldable with other stainless steels, crack resistant, and very corrosion resistant. However, it has a high coefficient of thermal expansion so large clearances for bolts mating flanges would be necessary. On the other hand, mild steel does everything relatively well and is cheaper. Although it is very susceptible to corrosion and transfers more heat than stainless, chrome or ceramic coating would address that. For the machinability and cost, we chose mild steel for our exhaust material. To address the stresses involved in fully constrained thermal expansion, a section of flex pipe will be implemented.

Figure 26: Section of stainless steel flex pipe

4.7 Intake System

The intake system consists of the air filter, throttle body, restrictor, compressor, plenum, bell mouth, and fuel injector as shown in Figure 27. The specific order of these components is determined by FSAE rules but the design of each has a large impact on the overall performance of the final system. For instance, a proper sized plenum can add about 15 hp to a naturally aspirated engine over no plenum. That equates to roughly 33% of the final power output. The bell mouth shape for the duct exiting the plenum allows the engine to make about 20% more power in the higher RPM range. A properly designed restrictor can enable the engine to generate about 10% more peak power than a flat plate. Even the length of ducting between the restrictor and the compressor can affect the power output. Every component needs to be properly designed in order to generate the maximum power possible.

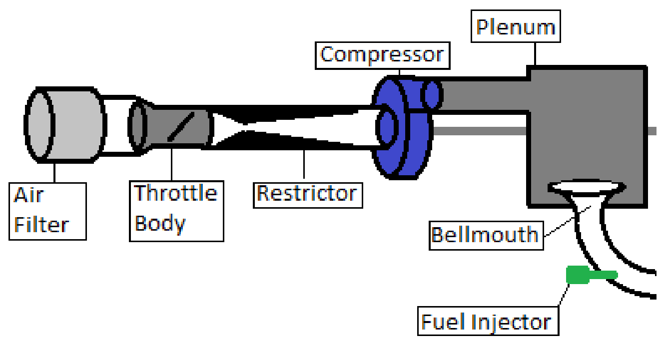


Figure 27: FSAE-governed order of intake assembly components

The main foci of the intake system design are the size and shape of the plenum, the length of ducting between the restrictor and the compressor, the distance between the entrance to the bell mouth and the intake valve, and, to a lesser effect, the design of the restrictor.

The throttle body and air filter are identical in size and design from the naturally aspirated car. Throttle body sizing is a tricky subject in FSAE because of the restricted design of the engines. If the throttle body is too large there is a lack of throttle resolution. This is because a small change in the blade angle will cause a much larger change in the airflow through the engine. As a result of this, the engine will reach its maximum flow rate (choked flow through the restrictor) well before the throttle reaches its fully opened position. The driver will feel like the engine is more difficult to control because small changes in pedal position will result in large changes in engine power. Additionally, the last portion of pedal travel will cause almost no change in engine power because the throttle body will already have been supplying the engine with maximum airflow. However, if the throttle body is too small then it can become the most restrictive element in the engine system instead of the air restrictor and will reduce the engine's output. Based on available throttle bodies and driver feedback from previous cars, a 40mm butterfly type throttle body was selected. This was the smallest one available on hand.

The rest of the intake system was optimized using Ricardo WAVE. Five variables were tested over a range of values in order to determine the optimum size of each. The variables are listed in Table 7 along with the constant value that four of them were held at while the last variable was being investigated.

Table 7: Intake variables tested in WAVE

Variable	Size	Units
Plenum Volume	1500	cc
Runner Length	225	mm
Compressor Inlet Length	50	mm
Plenum Inlet Length	110	mm
Restrictor Diffusion Angle	7	deg

4.7.1 PLENUM

The plenum is simply a volume of air after the restrictor from which the engine draws its intake air. Since the intake stroke is only one of the four strokes in the cycle, air is only being drawn into the engine for ¼ of the time, which results in large pressure pulses. If there are multiple cylinders operating this is not as much of an issue because the intake strokes are offset, creating a more uniform suction. The WR450 engine is a single cylinder so these pulses are quite large. The purpose of the plenum is to minimize the effects of the single cylinder pulses. On a naturally aspirated engine, these pulses reduce the amount of air that can flow through the restrictor. For the turbocharged engine, the compressor is placed between the engine and the restrictor, resulting in a much smoother draw through the restrictor. The plenum is to store a pressurized volume of air in preparation for each intake pulse. WAVE was used to find the effects of plenum sizing and the results for sizing the intake plenum are shown in Figure 28.

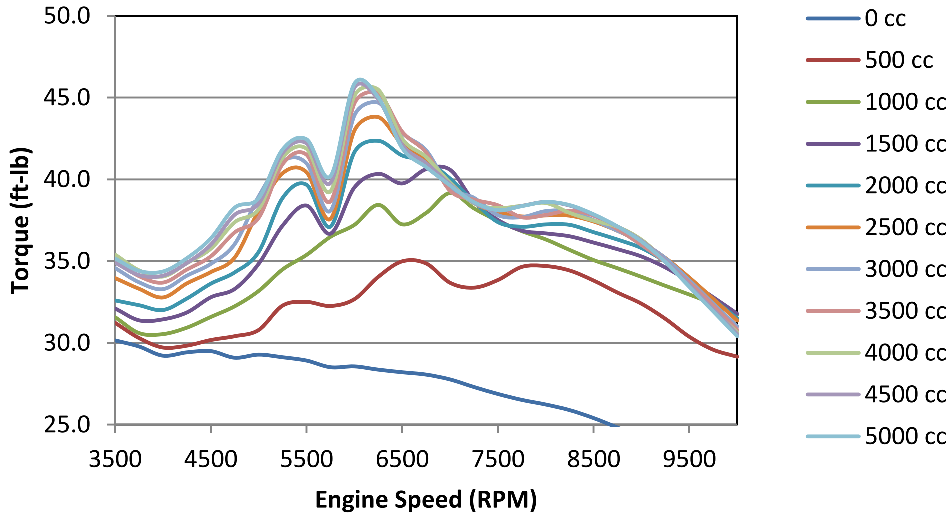


Figure 28: Simulated effects of plenum volume on engine torque output.

There is a tradeoff, however, because the plenum is placed between the throttle valve and the engine. If the plenum is made too large, throttle response suffers. Throttle response is how long it takes the engine to respond to a change in throttle position from the driver. This means that there is an ideal size for the plenum that will balance peak output and throttle response. Figure 29 shows how a plateau is reached for the peak output values as the plenum volume is increased.

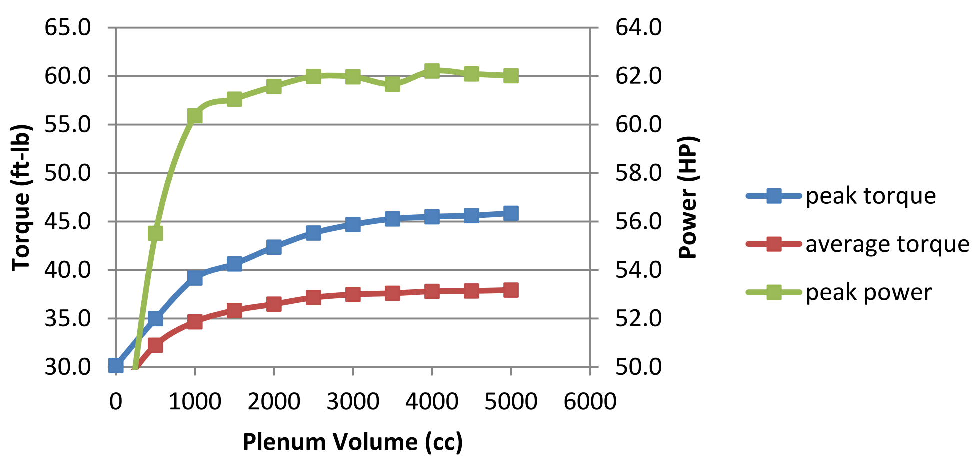


Figure 29: Simulated effects of plenum volume on torque and power.

Due to time constraints and limited availability of WAVE, only steady state testing was performed. Based on trends from previous FSAE teams for plenum sizing with minimum sacrifice to throttle response, the final plenum size was selected based on the minimum volume necessary to achieve 90%

of the plateau reached for peak power, peak torque, and average torque. With these criteria, a plenum volume of 1750cc (1.75L) was selected.

4.7.2 RUNNER LENGTH

The intake runner length is the distance between the intake valve and the plenum exit. This length is responsible for the resonance effects discussed in Section 4.6.2. Simulations in WAVE show that changing shorter lengths produce more power at higher RPMs and longer lengths shift the power into lower RPM ranges. It is typical for turbocharged engines to use longer runner lengths than naturally aspirated engines to take advantage of the resonance effects at lower engine speeds where the turbocharger is not fully spooled up yet. The simulation results shown in Figure 30 clearly show how the engine speed at peak power shifts with each runner length.

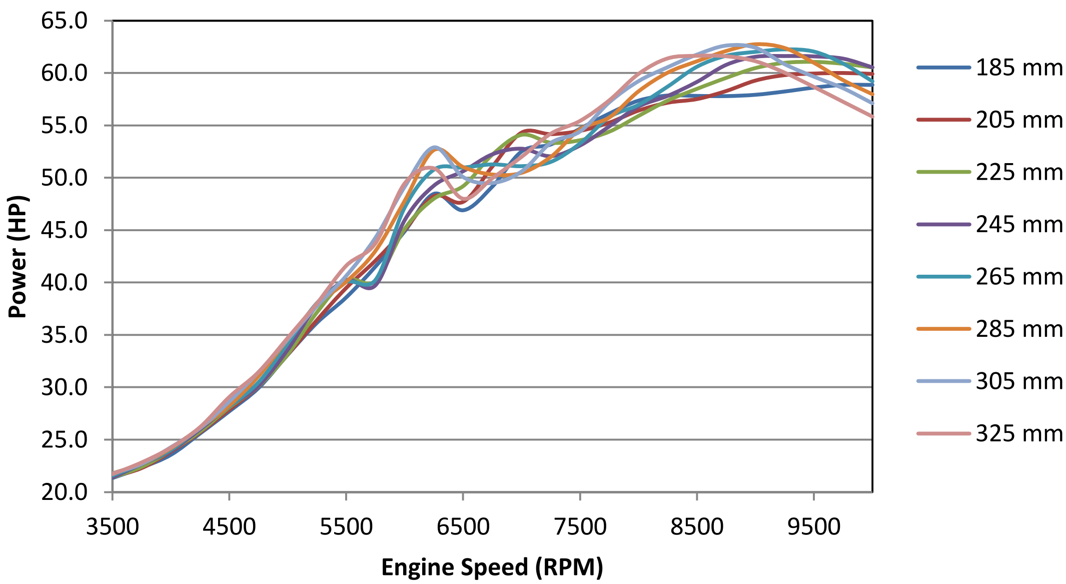


Figure 30: Simulated effects of runner length on power.

A side effect of reducing the engine speed necessary to produce peak power is that the car can be geared so that the engine does not have to rev as high as it would if its power was at a higher engine speed. This allows for a lower operating range of engine speeds while driving, which in turn reduces stress on internal engine parts and improves fuel economy. In addition to just shifting the power curve, changing the runner length also had an effect on the peak power and torque output as shown in Figure 31. Based on these results, an intake runner length of 305 mm was selected for the engine.

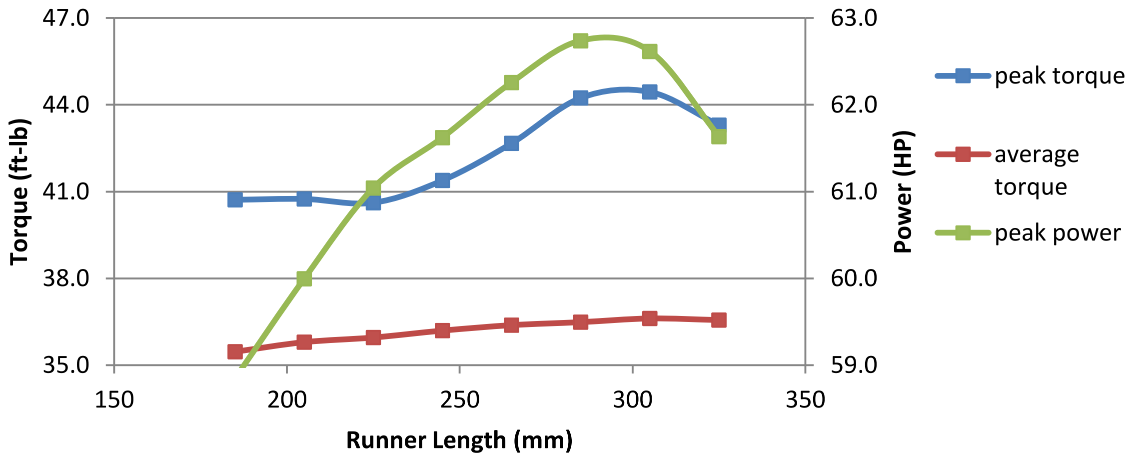


Figure 31: Simulated effects of intake runner length on torque and power.

4.7.3 RESTRICTOR

Formula SAE rules mandate that all air passing through the engine must pass through a circular orifice 19mm in diameter for cars running on E-85 (20mm for gasoline). The rules do not specify any details for the restrictor beyond this one requirement. This means the ducting leading to and from the restrictor can be designed to maximize the amount of air that can flow through it. These parameters can have a very significant impact on the overall performance of the engine. Figure 32 shows the difference in engine performance with two different restrictor designs. The red line is the projected power curve with the final restrictor design and the gray line is the projected power for a 19mm restrictor plate.

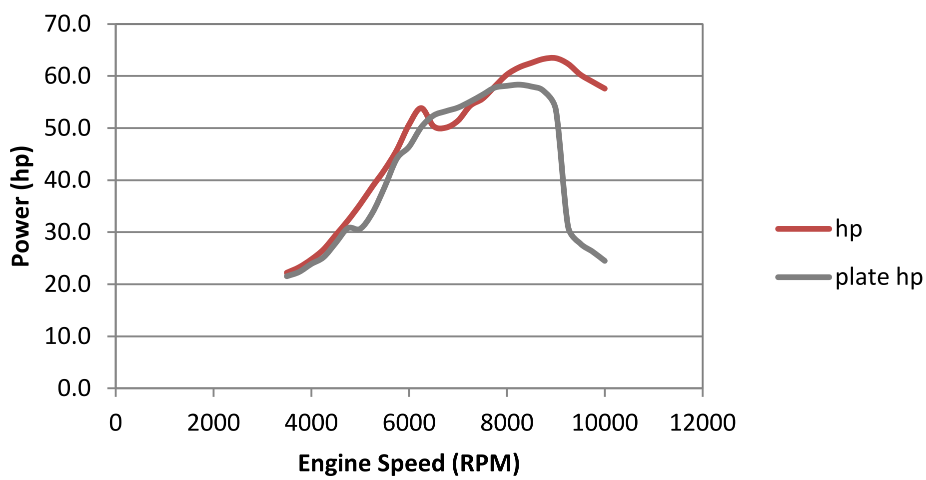


Figure 32: Simulated effect of restrictor design on power curve

A previous senior project by SPEED Systems already designed the restrictor for maximum flow rate and determined that a converging-diverging nozzle with a throat diameter equal to the restrictor size is the best design. We are assuming that the existing design is sufficient for the converging side of the nozzle but are investigating the effects of changing the angle on the diverging section. The results from the WAVE simulation are shown in Figure 33.

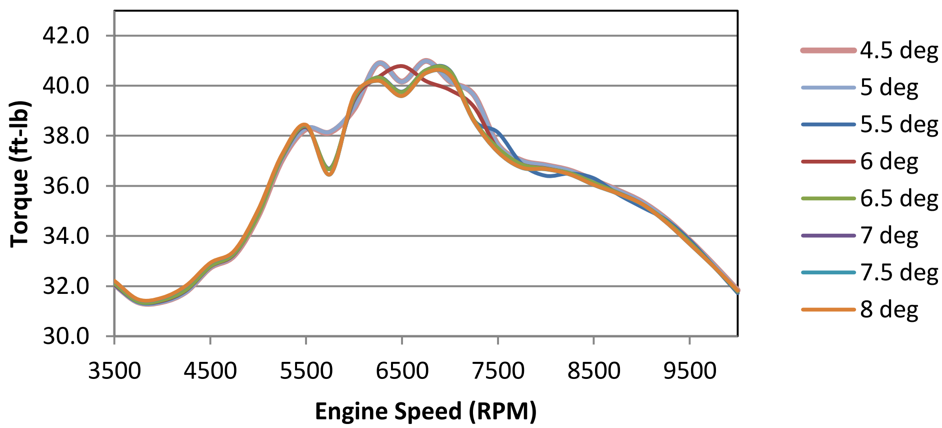


Figure 33: Simulated effects of restrictor diffusion angle on engine torque

The simulation results show that the diffusion angle has a minor effect on engine output, which makes sense because the engine is not expected to reach maximum flow rate through the restrictor in its current state. Figure 34 further demonstrates how little of an effect this angle has on the engine's output. Peak torque only varies by about 0.5 ft-lb over the range of values tested, peak power varies less than 0.5 hp, and the average torque value is almost constant. Even though the diffusion angle has very little effect on these values, it does play a role in smoothing over some dips in the torque curve. For this reason, a diffusion angle of 5 degrees was selected. A drawing of the final design of the restrictor is shown in Appendix C.

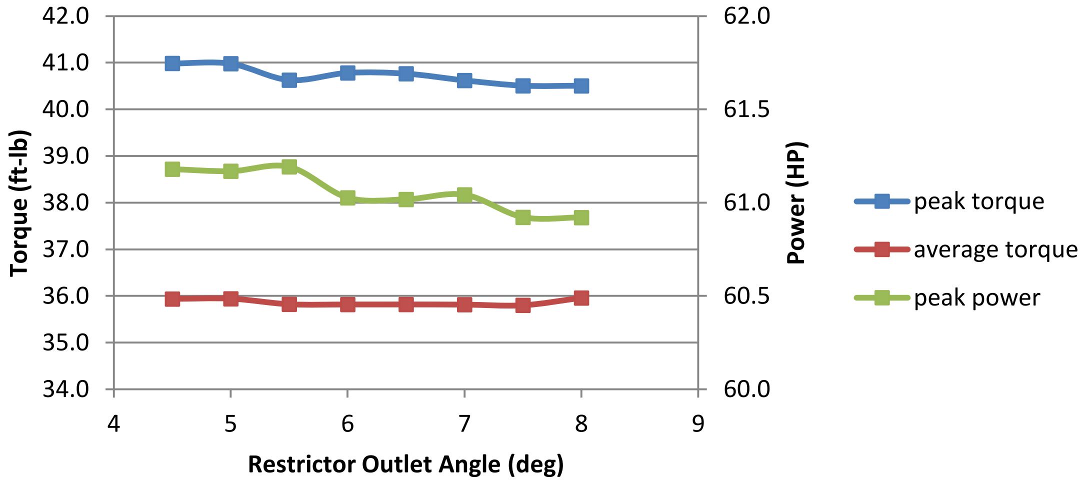


Figure 34: Simulated effects of restrictor diffusion angle on torque and power.

4.7.4 MINOR FACTORS

The last two factors investigated in WAVE, the length of ducting between the restrictor and the compressor and the length between the compressor outlet and the plenum, had very small effects on the output of the engine. Figure 35 shows that the compressor inlet length has a minimum value of 110mm, above which the torque curve smoothes out slightly.

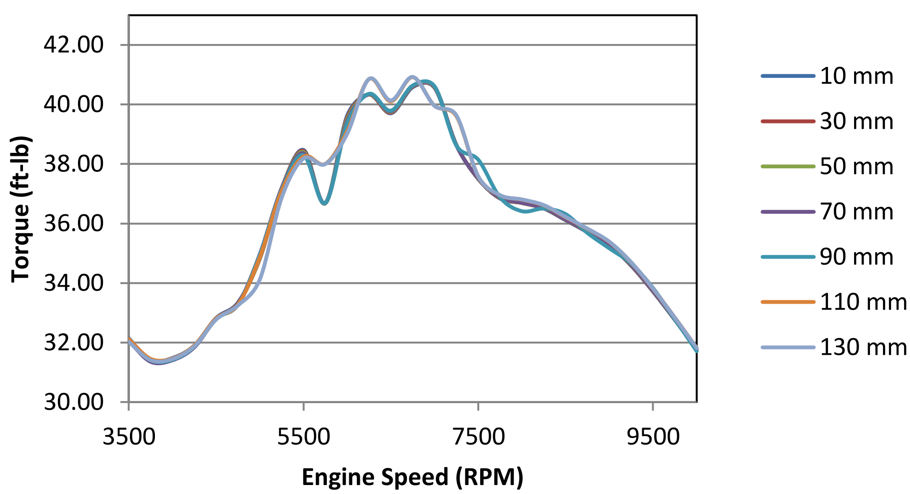


Figure 35: Simulated effects of compressor inlet length on engine torque.

The length of ducting from the compressor outlet to the plenum also had very little effect on the simulated performance of the engine. The results show that peak power slightly increases as this length is increased, but more importantly they show that this length can be adjusted for packaging considerations with little effect on performance.

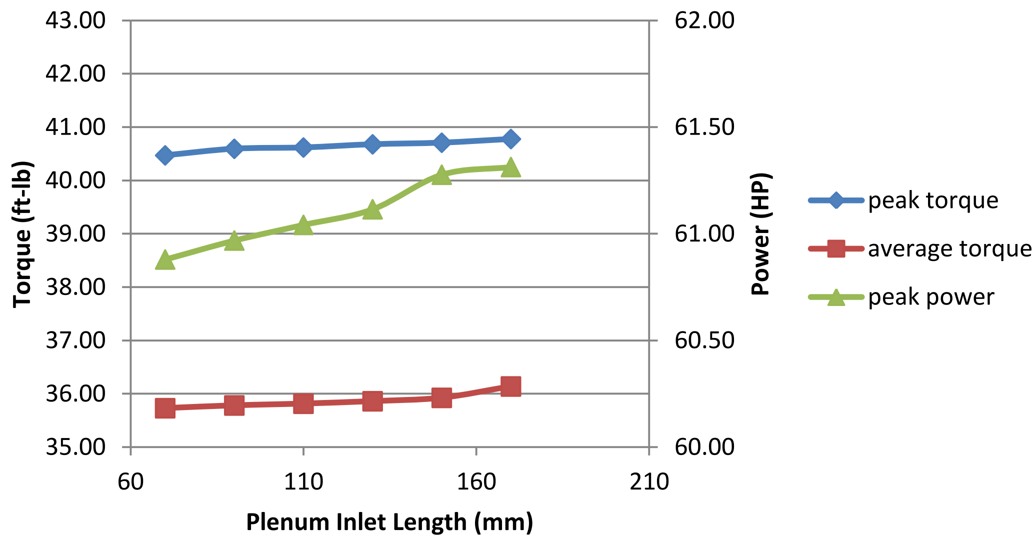


Figure 36: Simulated effects of plenum inlet length on torque and power output.

4.7.5. FUEL INJECTOR LOCATION

Less emphasis was put on the location of the fuel injector than was put on other components on the intake system. Ideally, the fuel injector would be placed as close to the cylinder as possible for the best performance. There are a number for reasons for this placement. One is that it takes time for the fuel to travel from the injector to the cylinder. The farther away it is from the cylinder, the longer it takes for the fuel to get to the cylinder when the demand for fuel changes. Additionally, when fuel is injected, not all of it is immediately atomized into the air. A portion of it will stick to the walls of the intake tract instead of going into the cylinder as desired. Under steady state conditions this is not a problem because the air moving over the fuel on the walls will evaporate it at the same rate as the injector is adding more. During transient operation, such as rapid throttle application, the air flow rate increases and these two rates are no longer balanced. More fuel must be added to compensate for the increased evaporation rate off the walls or the engine will not receive enough fuel. This is known as "acceleration enrichment" and is a tunable feature in fuel injection programming. This enrichment factor is small and simple to determine when the amount of fuel adhering to the walls is at a minimum, i.e. when the injector is close to the cylinder. Another advantage to placing the fuel injector close to the cylinder is better atomization due to the hot surfaces of the cylinder head. Our intake design will place the injector as close as possible to the cylinder head and pointed downward towards the intake valves.

4.8 Engine Preparation

4.8.1 Compression Ratio

The compression ratio is an important parameter of the engine that will affect the power delivery characteristics, fuel choice, boost level, and efficiency.

The static compression ratio (static CR) of an engine is defined as

$$SCR = \frac{V_{bdc}}{V_{tdc}}$$

Where V_{bdc} is the volume in the cylinder when the piston is at bottom dead center and V_{tdc} is the volume in the cylinder at top dead center. This is the compression ratio most often discussed and listed in engine specifications. The higher the compression ratio, the more work can be extracted from the fluid. The dynamic compression ratio (dynamic CR) is the effective compression ratio that the engine sees while running. While the static CR is defined simply by the geometry of the engine, the dynamic CR is influenced by multiple factors such as the engine geometry, cam timing, intake pressure, connecting rod length, and volumetric efficiency.

The dynamic compression ratio (DCR) of an engine is defined as

$$DCR = \frac{V_{ivc}}{V_{tdc}} * \left(\frac{P_{boost} + P_{atm}}{P_{atm}}\right) * VE$$

Where V_{ivc} is the volume in the cylinder when the intake valve closes, P_{boost} is the boost pressure above atmospheric, P_{atm} is atmospheric pressure, and VE is the naturally aspirated volumetric efficiency of the engine. Due to different volumetric efficiencies at different engine operating speeds, only the maximum value is used to calculate the dynamic compression ratio.

It is the dynamic CR that determines how much the fluid is actually compressed during engine operating and therefore the minimum octane rating necessary to avoid predetonation. This is why some engines require 100+ octane with an 11:1 compression ratio while others are perfectly fine on 91 octane with a 13:1 compression ratio. E-85 has an equivalent octane rating of 105, and with an expected operating temperature of around 180°F, the maximum dynamic CR is slightly above 10.5 according to Figure 37.

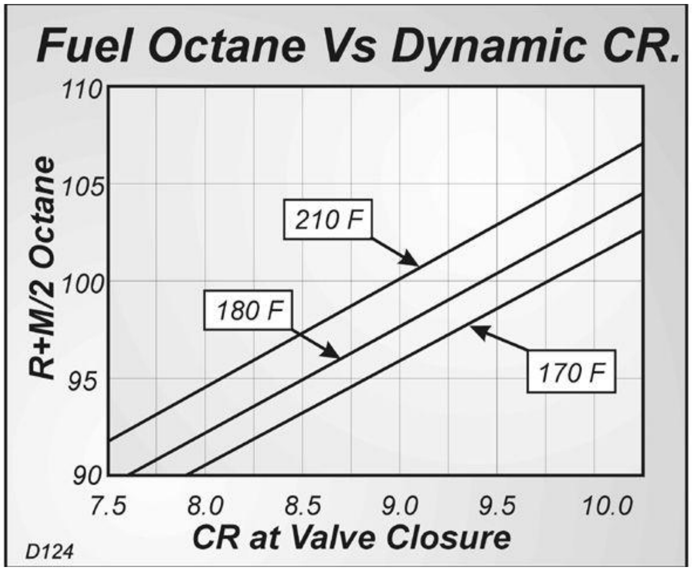


Figure 37: Fuel octane requirements for Dynamic CR values at coolant temperatures. [10]

When it comes to turbocharged engines, the effective compression ratio depends largely on boost pressure. As the Dynamic CR approaches the maximum value, the fuel and ignition tuning have almost no room for error. We want this system to have some margin for error because it is the very first iteration, but we also want to leave room for future FSAE teams to be able to expand. The results for the calculated effective compression ratio over a range of static ratios and boost pressures are shown in Table 8. Calculations for the dynamic CR are shown in Appendix I. The reason that the lowest static CR was not selected even though it is safe at all boost levels on the chart is because lower compression ratios will not give good off-boost performance. The total power output from the engine can be split into two categories: power from the base engine and power added by the turbocharger. With a lower static CR the power from the base engine is reduced and the turbocharger is relied upon more heavily to produce the desired power. Since the engine must be spinning at a certain speed for the turbocharger to produce boost, lowering the static CR reduces performance below this threshold. This system will be running 7- 8 pounds of boost, and based on the criteria a compression ratio of 11:1 was selected.

Table 8: Dynamic CR as function of static CR and boost pressure

Boost (psi)	4	5	6	7	8	9	10	11	12	13	14
Power (hp)	51	54	56	59	62	64	67	70	73	75	78
Static CR	Dynamic CR										
9	5.93	6.25	6.56	6.88	7.20	7.52	7.83	8.15	8.47	8.78	9.10
9.5	6.23	6.56	6.89	7.22	7.56	7.89	8.22	8.56	8.89	9.22	9.55
10	6.52	6.87	7.22	7.57	7.92	8.26	8.61	8.96	9.31	9.66	10.01
10.5	6.82	7.18	7.54	7.91	8.27	8.64	9.00	9.37	9.73	10.10	10.46
11	7.11	7.49	7.87	8.25	8.63	9.01	9.39	9.77	10.15	10.53	10.91
11.5	7.41	7.80	8.20	8.59	8.99	9.39	9.78	10.18	10.57	10.97	11.37
12	7.70	8.11	8.52	8.94	9.35	9.76	10.17	10.58	11.00	11.41	11.82
12.5	8.00	8.42	8.85	9.28	9.71	10.13	10.56	10.99	11.42	11.84	12.27
				DCR<9.5	9.5<DCR<10.5		DCR>10.5				

4.9 Simulation Results

After the designs of the subsystems were finalized, the features were input into WAVE to predict the engine power curve. The results are shown below in Figure 38. There is a considerable improvement with the turbocharger system over the naturally aspirated engine's performance.

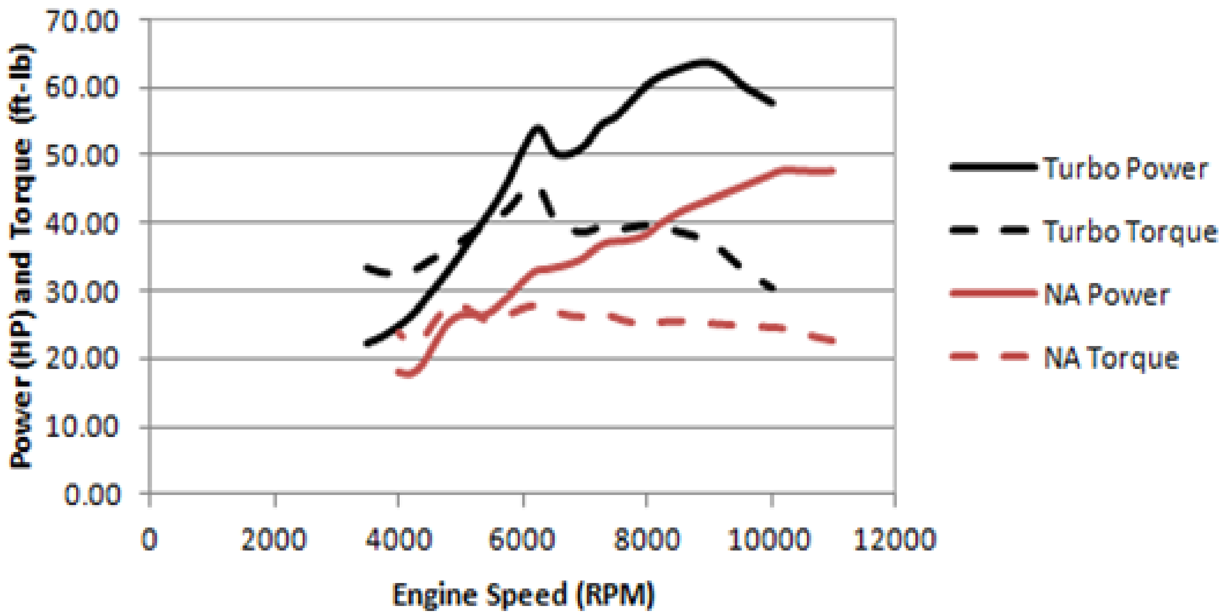


Figure 38: Simulated projected engine power curve

5.0 Model Layout

A model layout of the components is shown in Figure 39, below.

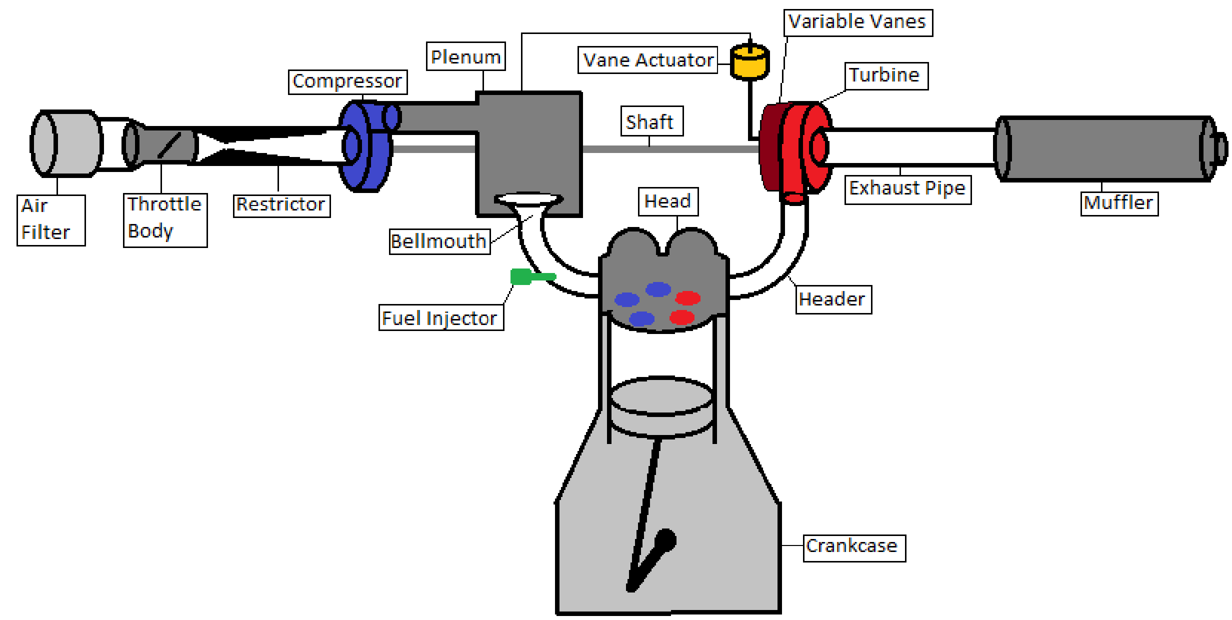


Figure 39: Model Layout of Turbocharger System

6.0 Management Plan

Designing a turbocharger system requires an extensive amount of time within one sub-system. Therefore, we decided that it would not be feasible or required for everyone to be working on the same sub-system at the same time. The turbocharger system has three main subsystems: intake, exhaust, and boost control, which we split between the group.

Intake:

Matt Roberts

Exhaust:

Eric Griess

Boost Control:

Kevin McCutcheon

The intake lead was responsible for the design of everything before the engine. This included the air filter, throttle body, restrictor, intake plenum, fuel injector, and all appropriate sensors. The exhaust lead was responsible for everything after the engine. This includes the header pipe going into the turbine, the exhaust pipe leaving the turbine, the muffler, and all appropriate sensors. The boost control lead was responsible for controlling the variable vanes within the turbine. All three leads worked together on installing the heavy duty components, the cooling system, the oiling system, and tuning the engine.

Major milestones of the project:

- Completed Turbocharger Calculations: February 17, 2012
- Calculations Approved by Honeywell: February 21, 2012
- Received Turbocharger: February 23, 2012
- Obtain Ricardo Wave (Simulation Software): March 27, 2012
- Order heavy duty parts: April 1, 2012 (Deadline to spend MESFAC money)
- Finalized Ricardo Wave Simulation to optimize component sizing: May 11, 2012
- Finished SolidWorks designs of intake and exhaust: May 18, 2012
- Parts built: June 8, 2012
- Heavy duty parts installed: June 8, 2012
- Finalized Complete Assembly: October 1, 2012
- Software integration and instrumentation installed: October 14, 2012
- Began testing: November 17, 2012
- Finished testing: November 29, 2012

One issue that Honeywell advised us about was the pulsations from the exhaust of the single cylinder engine. These pressure waves affect how the turbocharger is powered, so we had three different approaches to exhaust manifold design: straight, multiple pipe, and plenum. Although straight is the most common, we found that exploring the other options gave us valuable insight. The two main variables that we had hoped to increase with exhaust manifold design were faster turbocharger spool speed and higher peak power.

These pulses also have an effect on the intake, which is why such extensive design and testing was performed on the intake side of the engine. A former senior project team, Speed Systems, designed the intake for the naturally aspirated engine that is currently on the car. This design was the first one that was tested and the other possibilities were determined by using the software tools described below.

Due to time and money constraints we could not build several different designs and test them all. To overcome this, heavy usage of computer software was used for design and simulation. Software such as SolidWorks (CAD), Abaqus (FEA), and Ricardo Wave (Simulation) were used for modeling and simulation. We used these tools to optimize the dimensions of all necessary components before building to eliminate design or manufacturing iterations, which would have been time consuming.

6.1 Project Schedule

To help manage our time and to keeps the project on track, we created a spreadsheet in Google Drive as a project schedule. Being on Google Drive made the project easily accessible and updateable. This schedule included all the major milestones to create a base for the timeline of the project. It also included more detailed instruction, such as the tasks to be completed, dates to be completed by, and the person of the group that was to complete it. The schedule was periodically updated to show our progression and to reflect a realistic timeline of the project. This schedule was undoubtedly a vital tool in the progression of the project. By assigning specific responsibilities and deadlines, we were able to stay on task and always knew what had to be completed next. The schedule is not included in this report because of the size. Figure 40, below, shows a small section of the schedule. Red cells were deadlines, yellow cells showed when the task should have been being finished, and green cells showed when the task should have been being worked on.

				R	F	S	S	M	T	W	R
subsystem	task	description	date:	24-May	25-May	26-May	27-May	28-May	29-May	30-May	31-May
intake	design	wave									
		packaging									
	order										
	build	TPS to MAP/MAF Tune (No Turbo)									
		fabricate									
exhaust	design	packaging									
	order										

Figure 40. SLO Racing's project schedule.

7.0 Final Design and Manufacturing

7.1 Overall Manufacturing Timeline

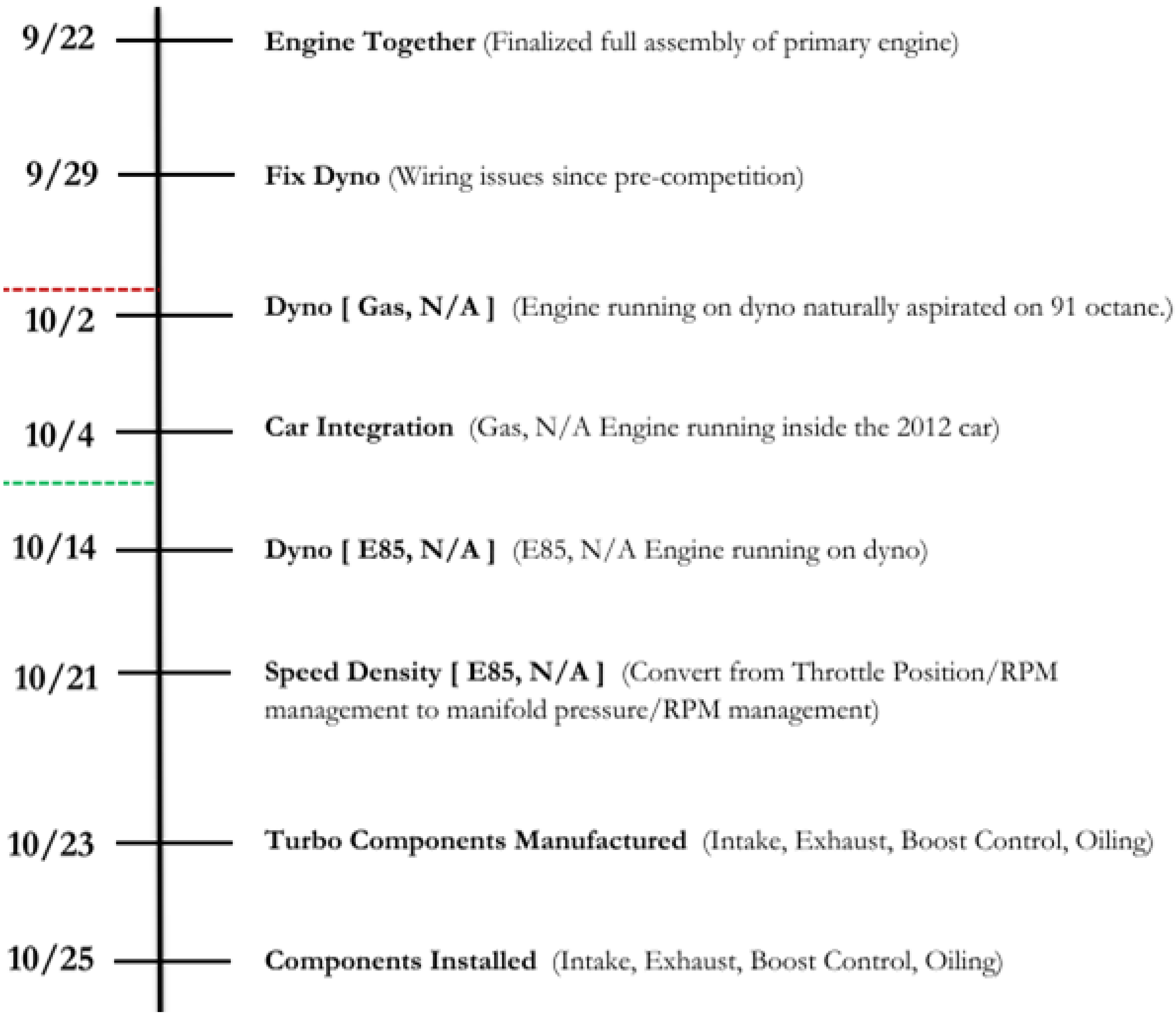


Figure 41: Manufacturing Timeline

7.2 Engine

Whenever an engine's output is raised significantly above its stock power level, the chance of component failure greatly increases. Some stock parts are strong enough to withstand the increased stress but there are always weak links in the system. Due to limited time, money, and resources, we relied on the experience of professional motorcycle engine builders to determine what components need to be upgraded instead of waiting for our engines to destroy themselves. These professionals have spent years testing hundreds of engines at all power levels from completely stock to absolutely crazy so they know exactly what is going to break and at what point it is going to break. Based on their recommendations for the WR450 engine we selected the following components to be upgraded:

- Piston
- Connecting Rod
- Crankshaft
- Piston Oil Jet
- Valve Springs
- Cylinder Head Gasket
- Cylinder Head Bolts
- Clutch Springs
- Clutch Plate

7.2.1 PISTON

The piston needed to be changed in order to lower the compression ratio from the stock 12.5:1 to the new 11:1. Cal Poly FSAE is sponsored by CP-Carrillo, so the major design features were left to them. The piston shown in Figure 42 features an 11:1 static compression ratio, the use of two compression rings, thicker rings and ring lands, and a fully boxed design to withstand the increased pressures. All of these design features make the piston better suited for a turbocharged application.

Figure 42: New 11:1 piston from CP Carrillo

7.2.2 BOTTOM END

The connecting rod is responsible for transmitting the force generated by the combustion process to the crankshaft and for controlling the motion of the piston. These two forces, the first in compression and the second in alternating compression and tension, are caused by two distinct engine parameters. When the torque level is increased, the force acting through the connecting rod is increased as well. The second force depends entirely on the mass of the piston, the mass of the connecting rod, and the speed of the engine. It is the combination of everything from increased power to more mass that drives the need for a stronger connecting rod, even though the maximum engine speed will be reduced from stock. Again, the design work was left to the engineers at CP-Carrillo who based their design on 90hp and a 10,500rpm redline: specifications we gave them. The power number is much higher than what we will see on our engine, but we want to make sure this engine can be the base for future iterations at Cal Poly that may make power closer to that level.

Figure 43: Connecting rod from Carrillo and balanced crankshaft installed

The next component to be upgraded was the crankshaft, specifically balancing of the crankshaft. The crankshaft itself is strong enough to withstand up to about 90hp but it needed to be balanced with the new beefier components in order to ensure it will last. With the piston and connecting rod receiving significant redesigns compared to their stock counterparts, their weights increased. There are two different types of imbalance to take into consideration with respect to piston engines: rotational and reciprocal. Rotational imbalance is driven purely by the rotating mass; think of how car tires need to be balanced. The reciprocal imbalance is caused by the mass of the piston and little end of the connecting rod rapidly moving up and down in the cylinder. It is typical for crankshafts to be intentionally imbalanced rotationally in order to balance the reciprocating mass. The rotational imbalance is then counteracted with a counterbalance shaft that spins in the opposite direction of the crankshaft with its own rotational imbalance 180° out of phase from the crankshaft's imbalance. It is impossible to completely cancel all of the forces in a single cylinder engine but the vibrations and stress on the engine are greatly reduced when imbalances are minimized. The balancing was performed by Q&E Engine Machine Shop in Anaheim, CA and involved adding weights to the bottom of the crankshaft in addition to drilling material out of the top of it.

A common upgrade on these first generation 2003-2006 WR450 engines is the addition of a piston oil jet. The jet directs a small stream of oil at the bottom side of the piston in order to ensure it is adequately cooled and the wristpin is lubricated. Yamaha implemented a piston oil jet when they redesigned the WR450 engine for 2007 and this part can be retrofitted onto the older model engines. The oil jet can be seen protruding into the cylinder from the right side of the engine (Figure 44) and directed up towards the bottom of the piston. This is one of the simpler upgrades to the engine but it was recommended in order to help keep piston temperatures down and prevent problems down the road.

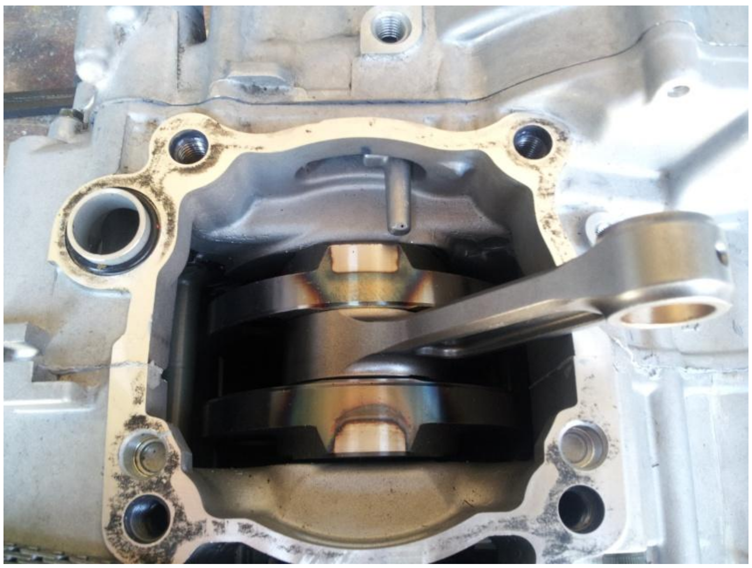


Figure 44: Piston oil jet installed in engine

7.2.3 CYLINDER HEAD

Now that the bottom end had been appropriately strengthened, the next step was to look at the cylinder head. When an engine is turbocharged, both the intake and exhaust pressures are raised to well above atmospheric conditions. This means that the pressure, and therefore the force, acting on the back side of the valves has increased as well. The purpose of valve springs is to close the valves after they have been opened and to hold them closed until the next time the camshaft pushes them open again. Since the forces of the intake and exhaust pressures are acting in the direction that would push the valves open, the effective spring rate, and therefore the force, closing or holding the valves closed has been reduced. At 8 psi of intake boost (about 12 psi for the exhaust) this equates to a reduction of 7 pounds (23% of seat pressure) for each intake valve and 11 pounds (37% of seat pressure) for each exhaust valve. This increases the risk of "valve float," where the valve does not stay in contact with the camshaft, and corresponding valve to piston contact. The solution to this problem was to install stiffer springs from Kibblewhite Precision Machining. These dual rate springs shown in Figure 45 hold the valves closed with 33% more force than the stock Yamaha springs (40lbs vs. 30lbs) and have a 42% higher spring constant (121 lb/in. vs. 85 lb/in.) [11][12]. In addition to withstanding the boost pressure, these springs will allow future Cal Poly FSAE teams the option to switch from the stock titanium valves to aftermarket stainless steel valves. Yamaha installed titanium valves on the WR450 in order to reduce the reciprocating mass and allow the engine to rev higher, but stainless steel is better at conducting heat away from the cylinder and will typically last longer before wearing out. It was decided that stainless steel valves would not offer any performance benefits at this time so they were not installed.

Figure 45: Dual coil (left) and stock (right) valve springs

Higher power levels, especially when achieved by forced induction, produce much higher cylinder pressures. These higher cylinder pressures place more force on all of the sealing surfaces around the cylinder. When everything is sealed properly all of the force goes into driving the piston down and producing power. However if there is one weak point, all of the pressure will try to escape from there. This is why it is so important to install a high quality head gasket between the cylinder and the cylinder head. A multi-layer steel (MLS) head gasket from Cometic was installed in place of the stock gasket.

Along those same lines, the higher cylinder pressures produce a greater force acting downward on the piston. Newton's first law states that every action has an equal and opposite reaction, and in this case that reaction is trying to separate the cylinder head from the rest of the engine. The WR450 comes with 4 long bolts holding the cylinder head in place. These bolts are adequate for stock power levels but they cannot withstand much more. They are designed to be torqued to their yield point, so tightening them beyond factory specifications will not produce any extra clamping force. The solution to this was to install stronger studs from Automotive Racing Products (ARP). In addition to being stronger and able to provide a greater clamping force, studs offer another benefit over bolts: only one mode of stress. When tightened, bolts are stressed both in tension and in torsion because the torque from the head has to be transmitted to the threads. Studs do not transfer the torque along their length because it is immediately transformed into a tensile stress by the threads at the top of the stud. Since the studs are capable of producing a greater clamping force than the original bolts without having to be torqued to their yield point, they do not have to be replaced every time they are removed. Torque recommendations from the professional engine builders were followed and the studs were tightened to 40 ft-lb.

7.2.4 CLUTCH AND TRANSMISSION

The transmission gears are capable of holding up to the increased power level, but the clutch needed to be strengthened to handle the torque without slipping. The clutch itself consists of 15 plates; 8 are mechanically connected to the crankshaft and 7 are connected to the transmission. Engine torque is transferred to the transmission through friction contact between these plates and the friction is directly proportional to the force clamping the plates together. The stock WR450 clutch can transfer a maximum of 39 ft-lb of torque before slipping. The most effective way to increase the torque capacity of the clutch is to install stiffer clutch springs to increase the force clamping the clutch plates together. Aftermarket springs were installed which are 30% stiffer than the stock springs so the clutch can hold the torque that the turbocharged engine will produce.

Side effects from stiffer clutch springs include greater clutch pedal effort required by the driver and pressure plate flexing. The clutch pedal effort will be evaluated by a driver in the FSAE car during testing but the pressure plate needed to be replaced with a stronger unit. A billet aluminum pressure plate from Hinson Racing was installed and is shown in Figure 46.

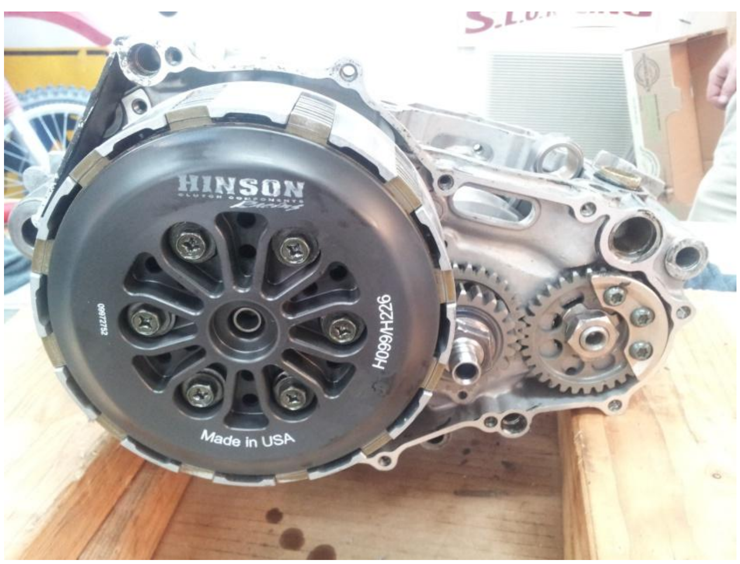


Figure 46: Heavy duty clutch springs and Hinson clutch plate installed

7.3 Turbocharger

Another support bracket had to be made and attached to the dynamometer in order to mount the turbocharger without placing any of its weight on the exhaust head pipe. This bracket bolts to a mounting boss already on the turbine housing. The turbocharger was mounted with the turbine oriented in the right direction and the other sections were clocked so that the oil path was within 20° of vertical and the compressor outlet pointed in the right direction.

Figure 47: Turbocharger location and turbine nozzle internals

In order to rotate the turbine housing relative to the center section, the entire variable vane mechanism had to be moved. Figure 47 (above, right) shows this mechanism along with the three black bolt heads that had to be relocated. Unfortunately only two of the three bolts holes lined up with existing holes in the turbine housing once the mechanism had been rotated as necessary. The solution was to drill and tap the turbine housing for the last bolt. Once this was done, the center section of the turbocharger fell within the required tolerance. The last section to be rotated was the compressor housing because it simply required loosening the bolts holding it on, rotating the housing, and then retightening those bolts.

Figure 48: Final turbocharger position and orientation

7.6 Boost Control

As previously explained, the variable vanes can be used to control the boost on the Garrett GT15V turbocharger. The boost pressure of the turbocharger is proportional to the rotational speed of the shaft, therefore the boost pressure will increase when the vanes are closed because that increases the rotational speed of the shaft. Conversely, the boost pressure can be decreased by opening the vanes and allowing the rotational speed to decrease. Now, to actuate the vanes we chose to use the Bosch 204103 Dual Port actuator instead of the stock actuator that was provided on the turbocharger. The main reason this was done was because the Bosch actuator has both pressure and vacuum ports that can control the vanes while the stock actuator can only be controlled by vacuum. This reduced the backpressure during light load by opening the vanes slightly and allowed the engine to slow down without boost pressure by holding the vanes at their full open position right after the driver lets off of the throttle, when a vacuum is created in the intake. A bracket was designed and built to adapt the Bosch actuator to the turbocharger. Since the motion of the actuator arms of the stock and Bosch actuators were in opposite directions, a rocker arm was used with the Bosch actuator to reverse the motion of the arm in relation to the vanes. Figure 49 shows the bracket with the center post for the rocker arm, holes to mount it to the turbo, and slots to mount and adjust the actuator. Figure 50 shows the actuator and bracket installed on the turbocharger, as well as the heat shield that was built to deflect the extreme exhaust temperatures.

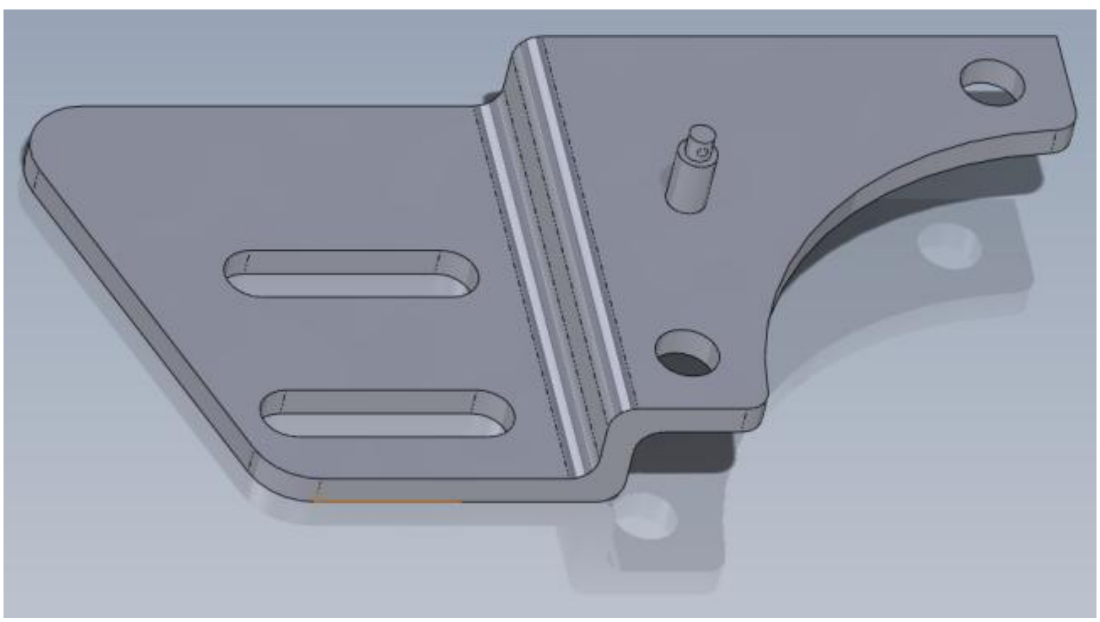


Figure 49. Turbo-mounted bracket for Bosch actuator

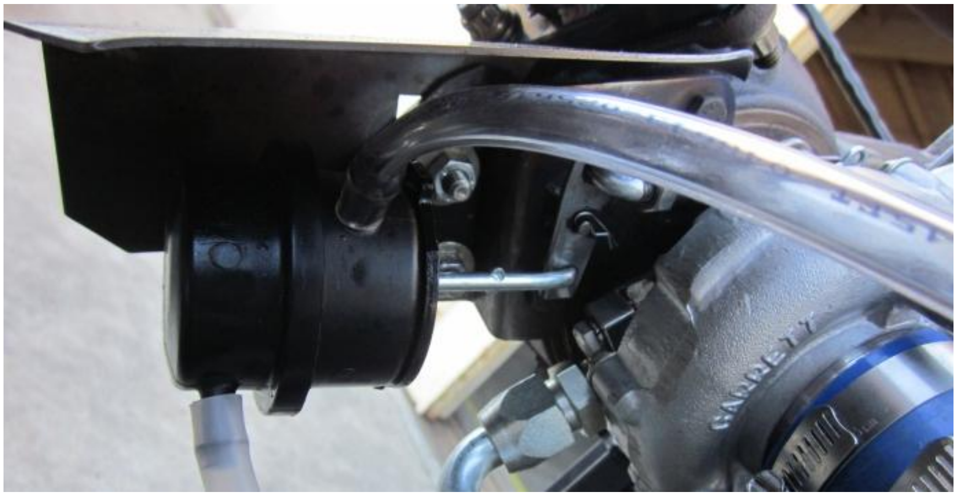


Figure 50. Actuator mounted with heat shield and pressure references attached

To properly setup the actuator so that the vanes were controlled correctly, we used the correlation between the actuator arm displacement and the pressure or vacuum inside the diaphragm chamber. Figure 51 is a chart that shows this correlation. It can be seen that to target a maximum boost pressure of 4 psi (to begin testing) the actuator must be setup so that the vanes are at their full open position when the actuator reaches 0.550 inches of displacement. This is the reason that the slots were put on the actuator bracket, allowing the actuator to be positioned correctly to target a maximum boost pressure. It can also be seen that the actuator arm can be controlled by vacuum, which is how we reduced backpressure during light engine load and held the vanes open momentarily when the throttle was closed.

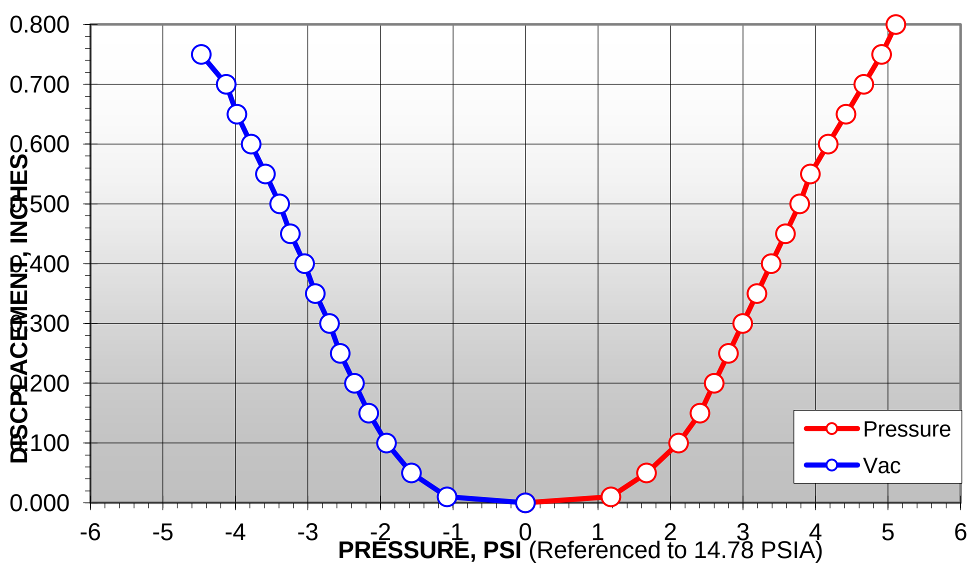


Figure 51. Pressure vs. displacement for actuator

(Bosch 204103 Dual Port actuator)

7.5 Intake System

The final dimensions for the critical components of the intake system are listed below in Table 9. The intake system was constructed out of aluminum and connected using rubber hose and hose clamps. The plenum was created by cutting, shaping, and welding aluminum sheet to create a cylinder 1.75L in volume. The bellmouth entrance was machined out of a solid piece of aluminum and then welded together with the plenum. The plenum also has a port for the Manifold Absolute Pressure/Intake Air Temperature (MAP/IAT) sensor necessary for controlling fuel delivery to the engine as well as a port to act as a pressure signal for the vane actuator.

Table 9: Final dimensions of intake system

Feature	Size	Units
Throttle Body	40	mm
Plenum	1750	cc
Runner Length	295	mm
Compressor Inlet	110	mm
Plenum Inlet	150	mm
Diffusion Angle	5	deg

The restrictor was machined out of a piece of Delrin in order to give the best surface finish possible and keep friction to a minimum so more air can flow through. The diameter of the throat needed to be as close as possible to 19 mm without being any larger. After machining, the final measurement came out to be 18.97 mm. In order for the restrictor to work with the rest of the intake system it was glued inside an aluminum tube which was then connected to the compressor inlet.

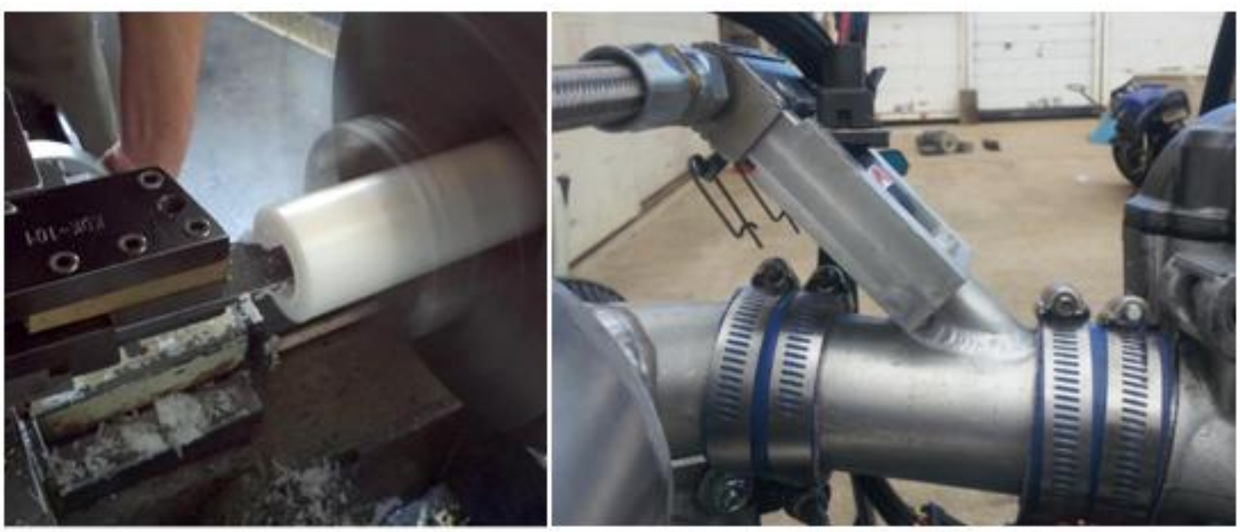


Figure 52: Machining restrictor and fuel injector location

Figure 53 shows the intake system as it was installed on the dyno. The air filter in the upper right corner is connected directly to the throttle body which is partially hidden by the fuel injector. The restrictor is in the long aluminum tube between the throttle body and the compressor. After the compressor is the plenum with the sensors attached. The bellmouth is hidden inside of the plenum and it connects to the elbow on the right side. The fuel injector is placed as close as possible to the cylinder head and angled so it injects fuel in the direction of the flow.

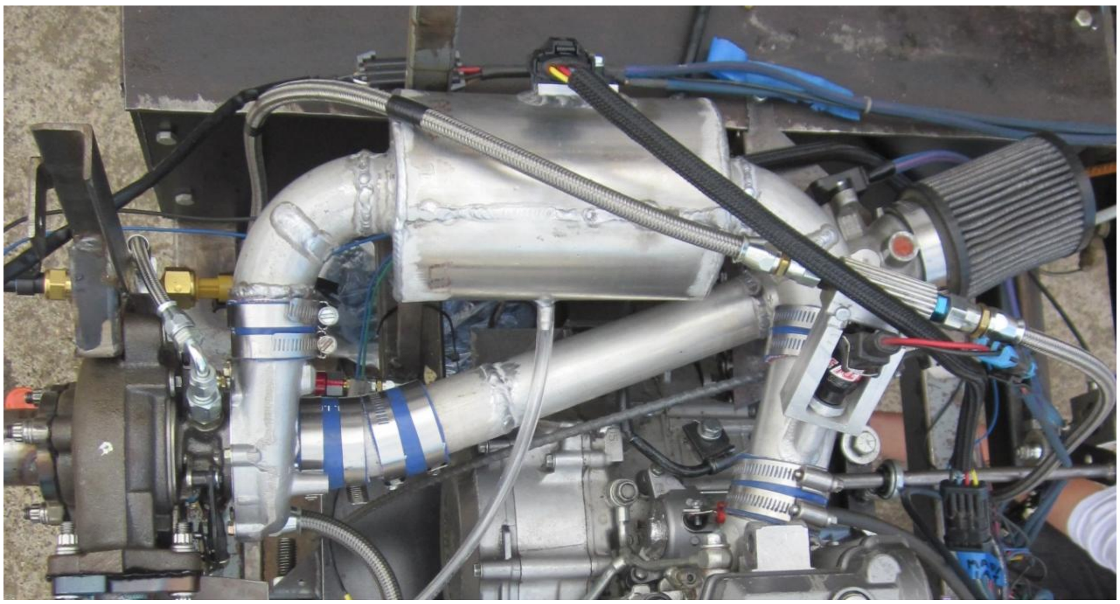


Figure 53: Mounted intake system

7.6 Exhaust System

The initial goals of the exhaust system included:

1. Minimize energy loss between engine and turbo
    - Address with smooth plumbing and large radii bends
2. Tuned length to take advantage of resonance tuning effects
    - Dictated by simulation results and applied to dyno.
3. Choose material that could handle intense thermal cycles
    - Mild steel chosen
4. Handle the stresses induced from partially supporting turbo and thermal expansion
    - Stainless steel flex pipe incorporated
5. Keep sound output below 110 dB at 7000 RPM.

Since then, there were many compromises and modifications that made our exhaust system differ from our initial design thoughts, and those are summarized in the next section.

7.6.1 SUMMARY OF FINAL MODIFICATIONS

Smooth bends were not necessarily implemented because with tighter bend radius was much easier to work with since the turbo location on the dyno was already defined. Although this is not ideal, it allowed for a shorter exhaust route.

We discovered that resonance tuning the exhaust did not play a large role in the exhaust because pressure difference between the combustion chamber and the exhaust is much smaller. From there, we determined that the exhaust would be primarily driven by dyno packaging.

Mild steel was still chosen due to availability and low cost. A stainless steel flex pipe was not introduced due to the unavailability with our exhaust pipe diameter. The flexible mount for the turbo alleviated that problem, allowing for a completely solid exhaust with fewer internal discontinuities to disrupt exhaust flow.

It was difficult to estimate what the sound level of the final system would be at, but research indicated that a turbocharger acts as roughly 1/3 of a muffler. While not very conclusive, we decided to go with a straight exhaust dump after the turbo to simplify manufacturing and reduce time.

16 gauge steel (.065") was still implemented, and although there is no analytical support, we believed that this would be an adequate wall stiffness to constrain 1 degree of freedom of turbocharger movement even during heat cycles.

Joining all of our connections were mild steel flanges CNC machined by Matt Bezkrovny. After machining they were within .001" flatness. Also, a high temperature carbon-steel gasket was implemented, with ARP stainless steel bolts connecting the flange to the turbine inlet.

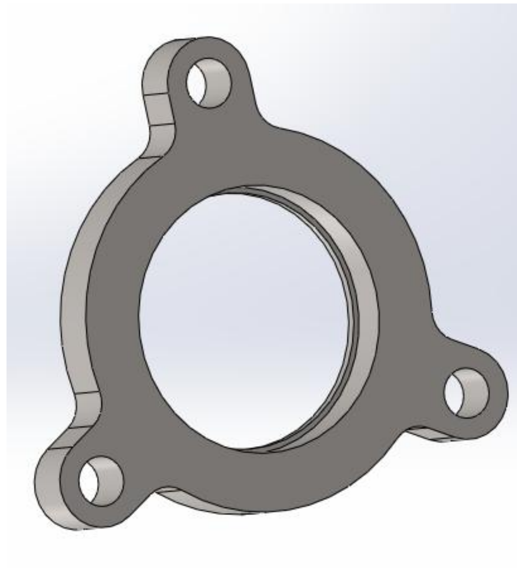


Figure 54: Turbine Exit Flange Model

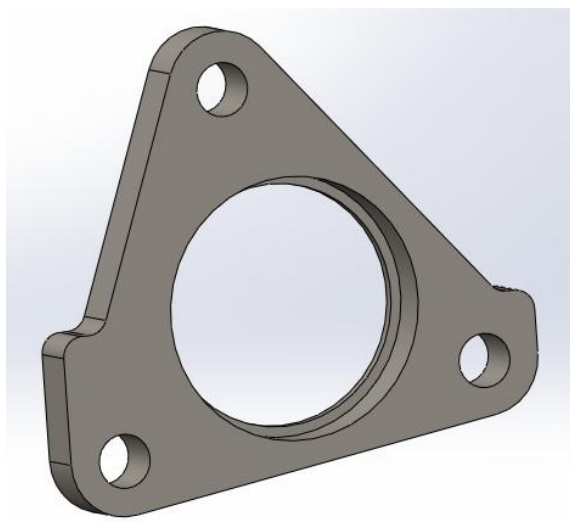


Figure 55: Turbine Inlet Flange Model

Data acquisition on the exhaust system consisted of a lambda sensor immediately after the turbine exit and an exhaust gas temperature sensor immediately before the turbine inlet. The location of the lambda sensor was recommended by *Maximum Boost*.

7.6.2 SCHEMATIC

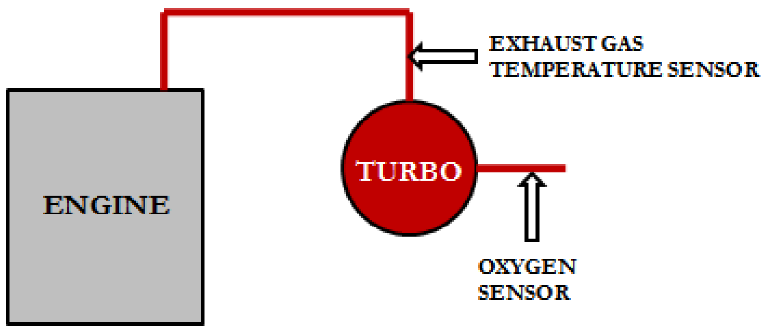


Figure 56: Exhaust system schematic

7.6.3 MANUFACTURING

The flanges were the first part of the exhaust system to be machined. The profile of these flanges was determined by "convert entities" feature on Solidworks, where it converted the outer profile of the turbocharger inlet and exit to a sketch. From there, the holes were expanded for ease of machining and to allow for thermal expansion. Each flange was 3/8" thick, based off of the recommendation from *Maximum Boost*. To ensure flatness, Matt Bezkrovny insisted on starting with ½" mild steel plate. From there, the flanges were machined.

Figure 57: Turbine outlet and inlet flanges machined.

Courtesy of Matt Bezkrovny

Another design tip from *Maximum Boost* that we implemented was the inner lip on the flanges. This was to allow the exhaust tubing to sit inside and rest on that lip. This was so the joint could maintain strength when weakened with welding.

For the turbo inlet manifold, we started with 1.625" OD, 16 gauge mandrel J bend, cut it, and mocked it up until it eventually connected both points. Connecting two points on different axis and at different angles proved to be challenging, but after a few tries it lined up without much tolerance.

From there, Simon Rowe fully welded our exhaust, alternating sides as he welded to minimize warpage. The exhaust was then mocked up again with the turbo flanges to ensure alignment – then they were welded on.

Figure 58: Partially welded outside of flange

Figure 59: Fully welded inside flange

An EGT sensor was chosen to be implemented after the manifold was fully welded, so the next step was creating the bung for the EGT sensor and drilling a hole for it. From there, it was welded in place. Using a straight pipe for our turbine exit greatly simplified our manufacturing process. With a finished flange, we started with a 1.75" OD mandrel bend, and cut a 90° bend. The exhaust bung for the lambda sensor was then manufactured while a hole to allow it was cut in the tubing. Soon thereafter, it was fully welded to create our final exhaust piece.

7.6.4 FINAL ASSEMBLY

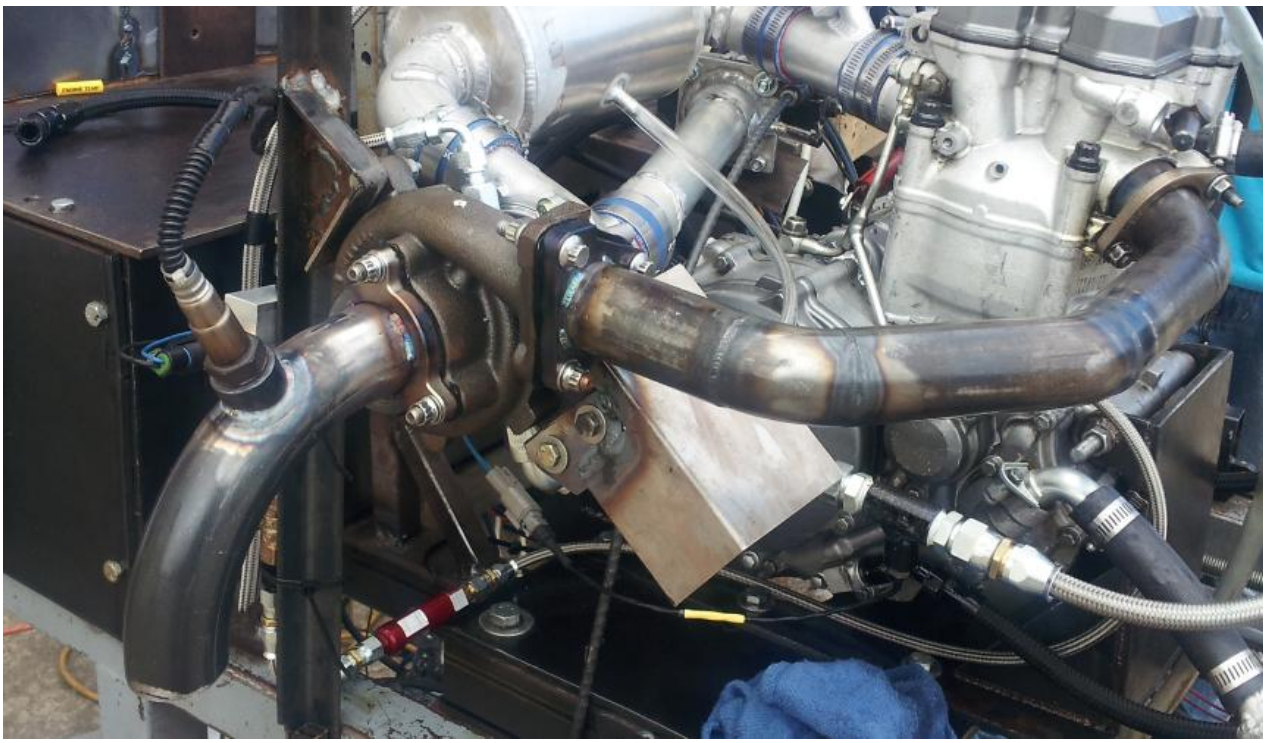


Figure 60: Full Exhaust Assembly

(EGT sensor not shown)

Although the flanges were still flat to .002-.005" after full welding, we implemented a high temperature gasket made of carbon and steel to prevent exhaust leaks. ARP stainless steel hardware was also used to clamp the flanges to the turbo.

7.7 Fuel System

For our fuel choice, we implemented E-85 to take advantage of the benefits outlined in section 4.4. After the fuel was chosen, the main goals of the fuel system were to provide adequate fuel pressure, filter the fuel properly, and prevent corrosion.

7.7.1 SUMMARY OF FINAL MODIFICATIONS

To provide adequate fuel pressure, we chose the same Aeromotive fuel pump (MN #11109). We didn't change the injector, which was an RC Engineering SH4-750 injector. The main changes we made from our initial design included plumbing details, filters, and fuel tank.

Our previous fuel tank was full of gasoline, made of stainless steel, and dirty. FSAE was also interested in replacing the tank, so we purchased a new fuel tank made of polyethylene for excellent corrosion resistance.

After looking into the E-85 community, and with personal recommendations from Matt Bezkrovny, the best fuel filtering setup was a 100 micron filter before the fuel pump, and 10 micron afterwards.

To plumb the lines, we were initially going to use JEGS Pro-Flo Nylon braided hose, but research indicated that ethanol dried and cracked nitrile rubber lines to the point where the fuel would bleed through the walls of the hose. For that reason, we decided to use Aeroquip's Teflon stainless steel braided hose (PTFE hose), and their hose ends.

7.7.2 SCHEMATIC

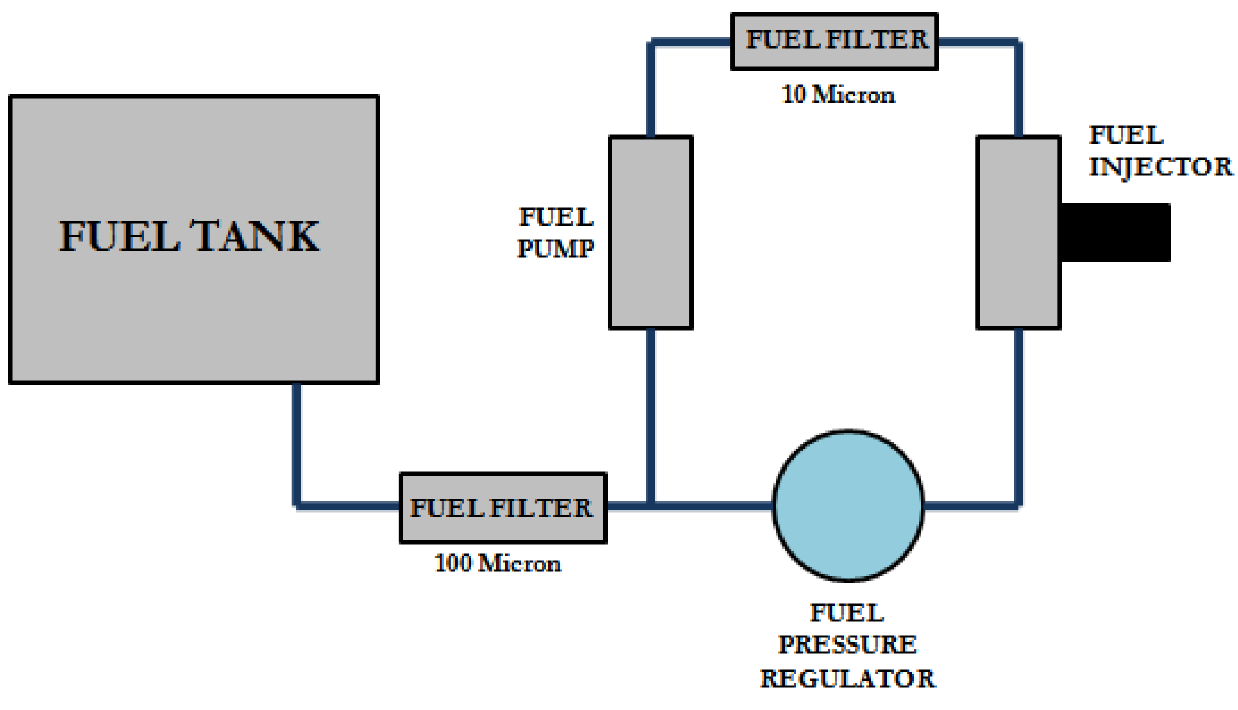


Figure 61: Fuel System Schematic

7.7.3 FINAL ASSEMBLY

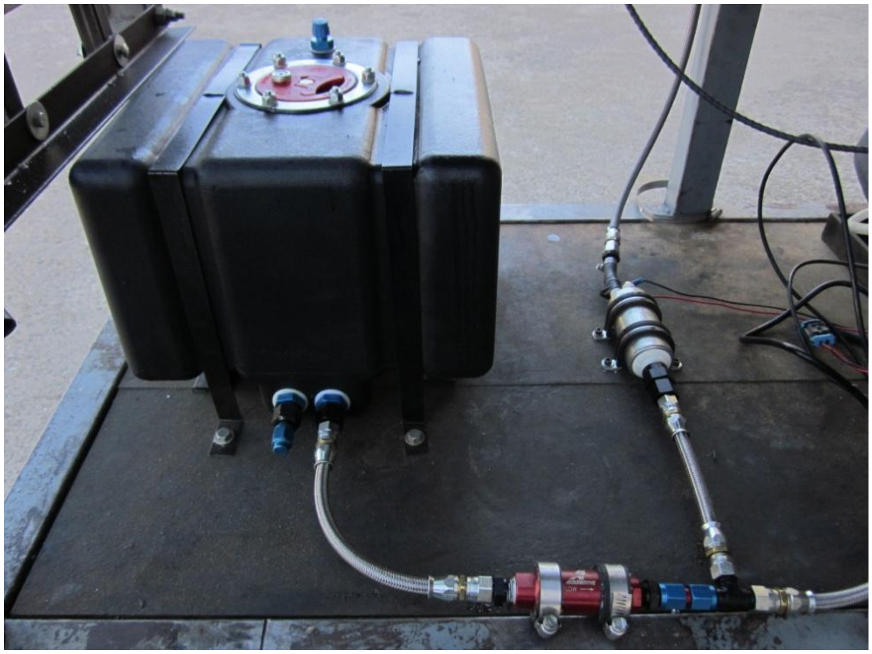


Figure 62: Plumbed Fuel System

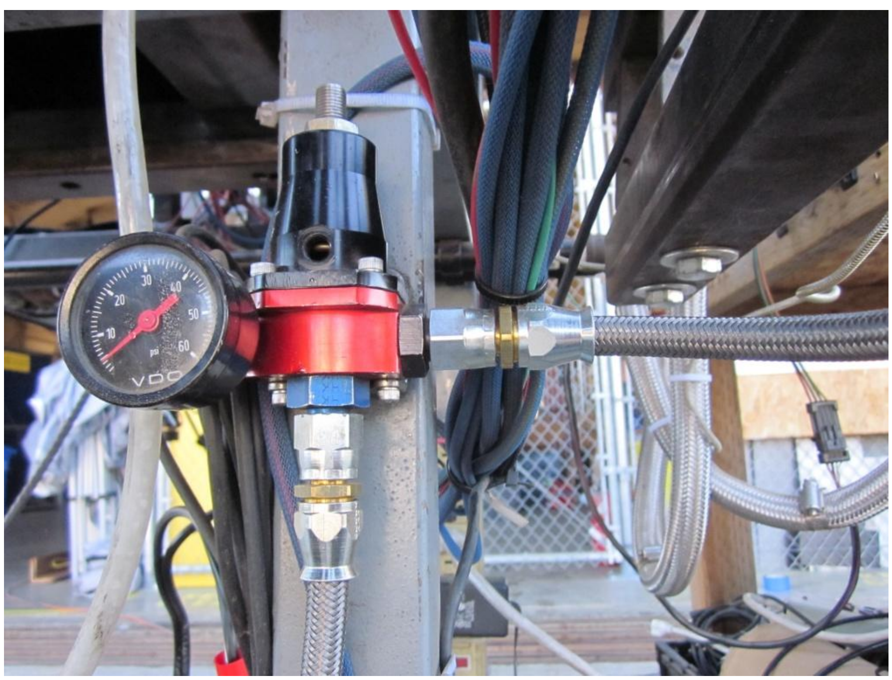


Figure 63: Fuel Pressure Regulator

Figure 64: 10 Micron fuel filter installed

7.8 Oiling System

The oiling system proved, by far, to be the most difficult sub-system to successfully integrate with the turbocharger. The original goals of the oiling system included:

- Provide adequate oil pressure to the turbocharger
- Provide adequate oil return system from the turbocharger
- Maintain safe oil temperatures
- Filter oil adequately
- Minimize oil burned through the engine

In our attempt to address these issues, we went through multiple iterations and found that the most adequate system was one where we ran an oiling system in parallel with the engine. The oil supply was teed off of the oil pump, fed to the turbocharger, and drained to a scavenge pumping system which returned the oil to the oil sump. We also implemented an oil cooler before the engine supply to keep temperatures lower.

7.8.1 SCHEMATIC

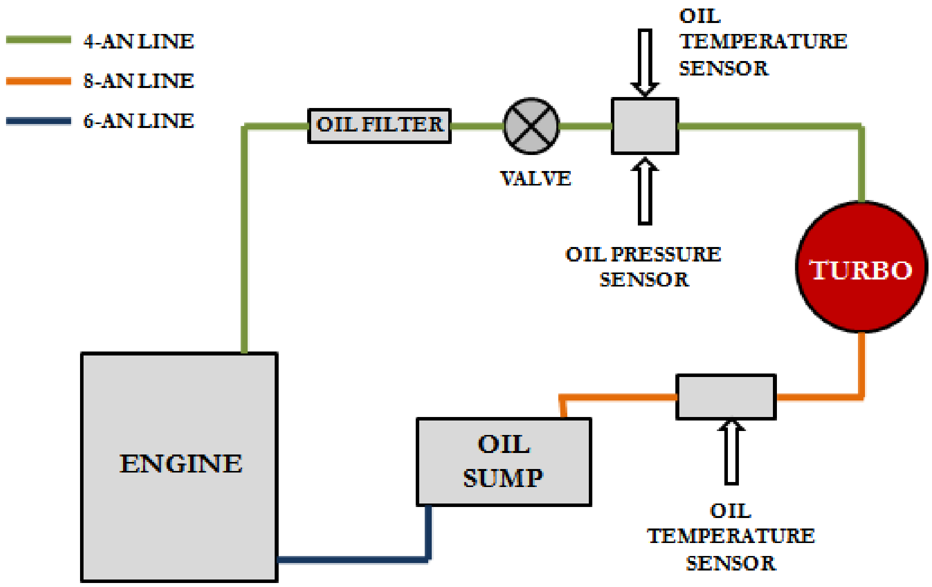


Figure 65: Initial Oiling System Schematic

7.8.2 MANUFACTURING

Similar to the fuel system, the oil system consisted mainly of routing line, assembling hose and fittings, and wiring pumps and sensors. The only manufactured part in the oil system was the oil return flange, which was designed with the same methodology as the turbo flanges. The material was mild carbon steel and 3/8" thick based on recommendations from *Maximum Boost*.

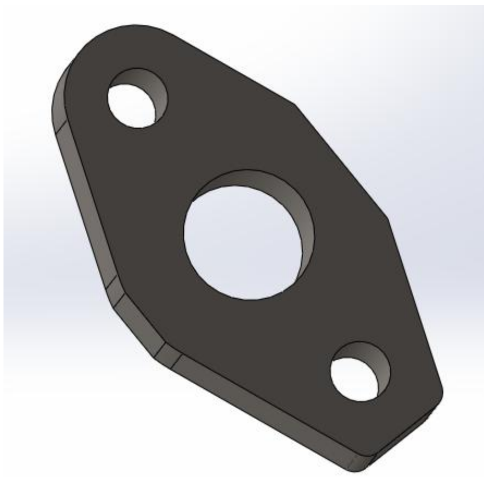


Figure 66: CAD Model of Oil Outlet Flange

Figure 67: Machined Turbo Oil Outlet Flanges

Courtesy of Matt Bezkrovny

7.8.3. FINAL ASSEMBLY

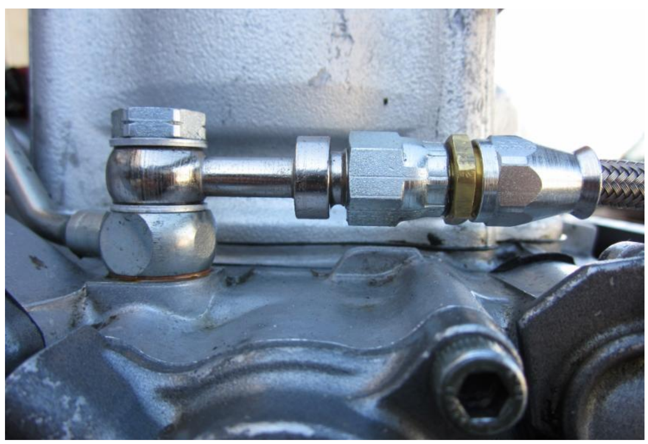


Figure 68: Turbo oil supply line tee from WR450 engine

Figure 69: Oil filter, valve, oil pressure, oil temperature sensors.

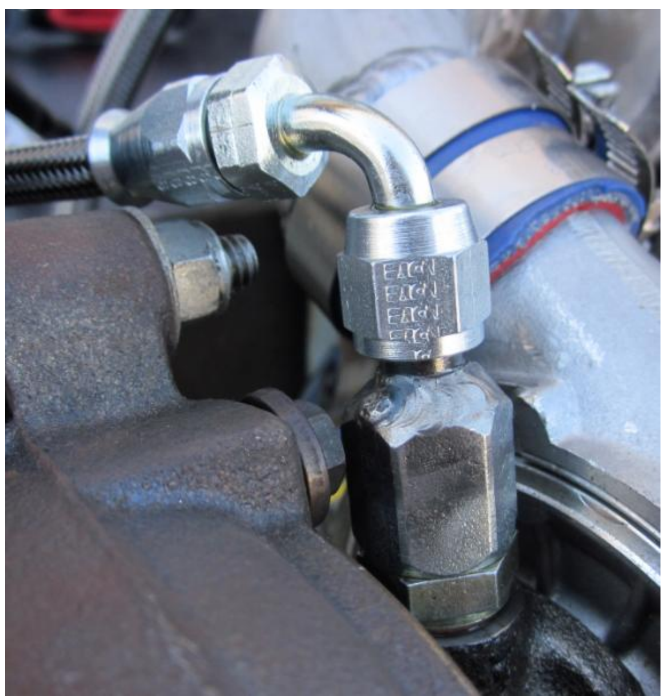


Figure 70: Turbo oil inlet

Figure 71: Turbo oil drain to 8-AN line

Figure 72: Oil return temperature sensor and bung

7.9 Dynamometer Safety

When the dynamometer was first built in 2008, time ran out before safety could be addressed. The only protection came from ¼ in. steel plate surrounding the chain drive and a small sheet metal flap that deflected the exhaust gasses.

The introduction of the turbocharger system carried with it a significant safety risk. Although turbochargers usually don't fail, the added stress to the engine can cause the engine and subsystems to fail catastrophically. In a worst case scenario, the connecting rod can fail or an oil leak can contact the hot exhaust and combust. The turbocharger system also uses higher temperatures and pressure in the oiling exhaust system, increasing the probability of gasket failure and fluid leakage. This leakage is a serious concern to bystanders because exhaust temperatures exceed 1600°F, hot coolant can exceed 180°F, and oil can exceed 300°F.

It was determined that there needs to be some form of transparent material that will allow us to see the engine while it is running. In most industrial applications, there are really only two choices: acrylic glass (also known as plexiglass) or polycarbonate sheets.

7.9.1 ACRYLIC VS POLYCARBONATE

Acrylic glass and polycarbonate are very similar materials in that they are completely transparent. Both are often used as an alternative to glass for their light weight. The difference between them lies in their properties and common applications.

Acrylic glass is brittle, has a lower strength, and doesn't handle heat as well compared to polycarbonate. Hence, acrylic is mainly used for low impact and low heat applications, such as protecting museum displays, aquariums, and CD/DVDs.

Polycarbonate on the other hand is not as brittle and can withstand more heat. For these reasons it is used for motorcycle visors, bulletproof glass, and even storm panels to protect windows during hurricanes. Based on these properties, polycarbonate was used in making the shield.

7.9.2 SAFETY SHIELD

For the safety shield it would be best to be able to have a full 360° view of the entire dyno with polycarbonate panels all around. Figure 73 below is a good example of what these shields would look like. Unfortunately, polycarbonate is very expensive and our budget did not allow for this type of construction.

Figure 73: Polycarbonate acoustic drum chamber

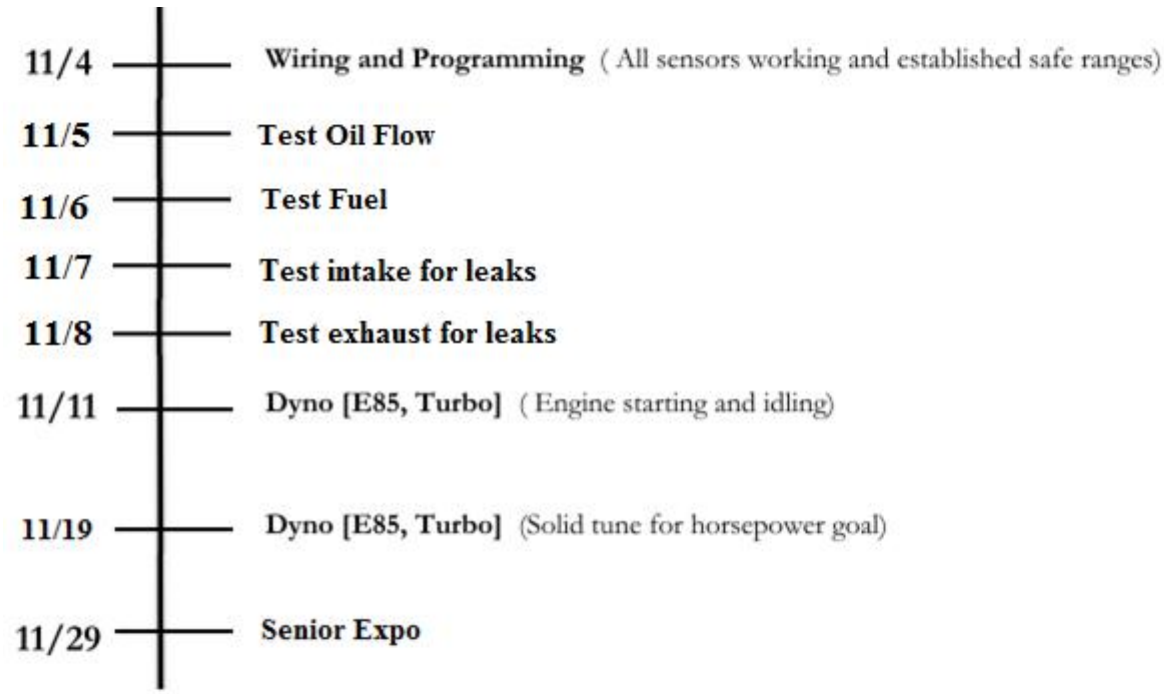
From audiocircle.com

To reduce the cost of the shield it was decided that instead of large panels of polycarbonate, most of the shielding would be constructed with OSD plywood sandwiched between two sheets of 22 gauge mild steel sheet metal. There would be one viewing window constructed from three sheets of .093 in. thick polycarbonate and quick release hinges at all four corners. Refer to Appendix H for safety shield plans. Figure 74 below shows the safety shield encasing the dyno test stand on a rainy test day.

Figure 74: Covered dyno safety shield

8.0 Testing

8.1 Test Plan Timeline



8.2 Problems Encountered/Solutions

Over the course of the project, we encountered numerous problems that slowed the progression of our project. Many of these issues were minor and we were able to solve them quickly. Some small issues included such things as our battery charger not charging the battery, slipping out of 5th gear, and spark blow out. However, there were several major problems that plagued the progression of the project, specifically during engine testing and tuning, which played a major role in the final outcome of the turbocharger system. These problems consisted of insufficient oil pressures feeding the turbocharger, oil being forced past the turbocharger seals, inadequate starter motors, boost pressure causing intake plenum mounts to come loose, clogged injectors, and the turbocharger vanes not actuating properly. Each of these problems and their solutions are discussed below.

8.2.1 ELECTRICAL

We had battery issues during the early stages of testing and determined that the old charger we had finally stopped working. We purchased a new charger to solve this problem.

A common problem with turbocharged engines is spark blowout. This is when the spark plug cannot arc because of the increased pressure and density of the fluid in the cylinder. Decreasing the spark plug gap from 0.030 inches to 0.025 inches solved this problem.

Starter motors have been a consistent issue for Cal Poly FSAE. This is because when testing and tuning the engine on the dynamometer, the engine is constantly being started, ran, and then shut down only to repeat seconds later. The Yamaha WR450 starter motors were not designed for this consistent of loading. Once we had the engine converted to E-85 and fitted with the turbocharger, tuning began and the first starter motor did not last long. After replacing it with another motor that we had which was in better condition, we got the same result. The starter motors were only lasting through a couple days of testing before they would stop working. To further assess the severity of the problem, we took one of the starter motors apart. We found no evidence of a major problem, however.

In an effort to reduce the load on the starter motor, the bushing that supports the motor's shaft was carefully drilled out and replaced with a needle roller bearing that was press fit into the housing. This significantly reduced the load on the motor judging by the sound of the motor at startup. It no longer sounded like it was struggling to turn over the engine. Unfortunately, this starter did not last much longer than the others. Again, we took the starter apart to see what had gone wrong and it had no major damage. However, it looked like all the internals of the motor all had some small defects and we thought that the sum of these defects were enough to stress the motor more than usual.

With no other options, we ordered an aftermarket starter motor. After installing it, we were able to resume testing. Everything seemed to be going well, until the engine stopped turning over once again. Believing that there was no way that our brand new started had already been damaged; we inspected the started clutch since we had seen a similar problem before. The starter clutch was fine which left only one possible explanation to the problem: the starter motor had to be damaged. We took out the starter and found the evidence immediately. The splined shaft on the motor that engages the gear on the engine was stripped. Figure 75 below shows the stripped shaft on the new aftermarket starter motor.

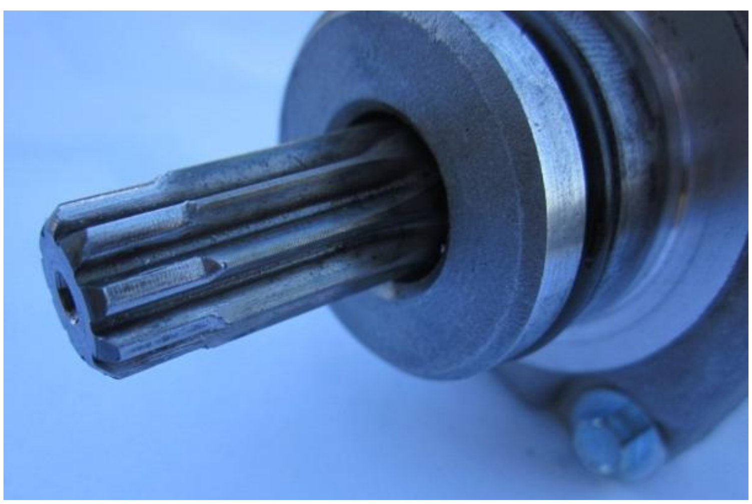


Figure 75: Failed aftermarket starter

Without any more WR450 starter motors and not able to pay $400 for an OEM starter that would have to be ordered directly from Yamaha, we had to get creative. We found a Yamaha R6 starter motor and surprisingly it fit on the WR450 engine. However, the R6 starter motor only has a single mounting point, where the WR450 engine has two and neither lined up with the point of the starter. An adaptor was machined so that we could utilize the R6 starter on our WR450 engine. We quickly discovered that the R6 starter is higher torque and lower speed than the WR450 starter, and it was not able to turn the engine over fast enough. In order to make the motor spin faster, we connected two batteries in series to provide the starter with 24 volts. The R6 starter worked incredibly well, despite the increased electrical load placed on it. Testing continued but it was not long until starter issues brought it to a halt. Thinking that the 24 volts had fried the motor, we began to inspect it. Fortunately, we found that the motor had just come loose because of the single mounting point design. After tightening the motor down, testing was resumed. Our starter problem was finally solved. As for the future design that will be used in the car, a new WR450 motor will be used and because the engine will not be consistently started like it is when it is being tested on the dynamometer there should be far fewer issues. Another option is to look into using a starter from a YFZ450 ATV, specifically the 2004-2006 models. The engines from these years were based on the WR450 dirt bike engine but came equipped with a stronger starter motor designed for more frequent use.

8.2.2 ENGINE

Another small problem that we encountered was at about 25 ft-lb of torque the engine would slip out of 5th gear. Since the engine had been run at higher torque in 5th gear last year, we decided that this is a sign of a worn transmission. Since rebuilding the transmission was not part of the scope of our project and LapSim results show that we will only use gears two through four, we decided to recalibrate the dynamometer and complete the testing in 4th gear.

8.2.3 LUBRICATION

The manufacturer of the turbocharger that we used, Garrett, specified the oil pressures that the turbocharger requires at idle and at full engine speed. These specifications were 5 psi at idle and 25 psi at our top engine speed, which was 10,000 rpm. This means that the oil pressure at the turbocharger must be proportional to the engine speed and that we could not simply use a constant oil flow rate. While this sounds more difficult, it is actually easier to accomplish as long as the turbocharger can be supplied with oil from the existing mechanical oil pump in the engine which already pumps proportionally to the engine speed. However, if a separate oil system is used to supply oil to the turbocharger than it must be controlled so that it will allow a flow rate proportional to the engine speed. To avoid having to create a complex control system for a separate oiling system, we decided to integrate the turbocharger supply and return lines into the existing system. Initially, our system sufficiently supplied the turbocharger with enough oil, but the post-throttle vacuum in the intake forced oil past the seals of the turbocharger's center section and into the engine. To alleviate this issue, we attempted to equalize the vacuum by creating a vent line just past the throttle. Because

SAE rules prohibit venting to atmosphere after the throttle body, we vented the intake to an oil catch can which was connected to the crank case. Figure 76 shows the configuration that we used.

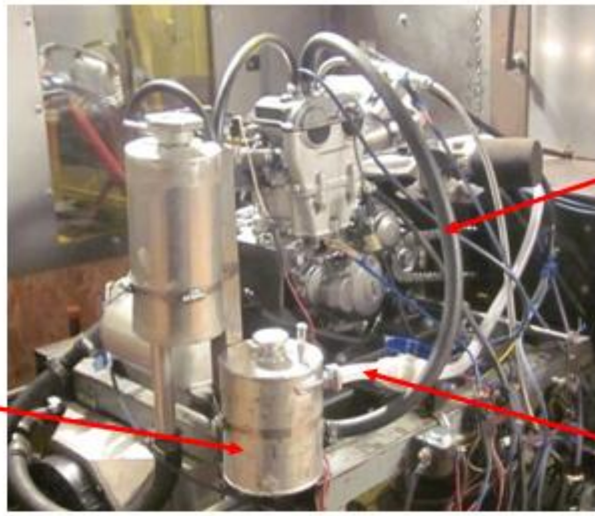


Figure 76: Oiling system plumbing for equalized pressure across seals

However, this created a vacuum in the crankcase which pulled more oil into the engine and away from the turbocharger. By solving one issue, another one was created. After failed attempts at several different vent configurations trying to equalize pressure and trying to increase the capacity of the mechanical oil pump, we determined that a separate oil pump was needed to help supply sufficient oil to the turbocharger. Out of time and options, we used an electronic fuel pump as an oil scavenge pump to draw oil through the turbocharger. However, using the fuel pump created a separate oil circuit which allowed the vacuum in the intake to pull oil past the turbochargers seals once again. Our research showed that only one Formula SAE team has been able to solve this problem by custom machining the center section of the turbocharger to accommodate better seals.

Using the fuel pump solved the oil drain issue until we tuned the upper range of engine speed. In this range where much higher oil flow rates are required, the converted fuel pump running at a constant speed was not able to pump enough oil away from the turbo. In order to compensate, another fuel pump was connected in parallel to increase the oil flow rate. This allowed us to finish testing and tuning the engine, but is in no way a proper solution. To actually solve the problem, an aftermarket electronic oil pump would have to be used to pull the oil through the turbocharger and it would have to be controlled by the ECU so that the flow rate of the pump could be varied with engine speed. The final oil schematic is shown in Figure 77.

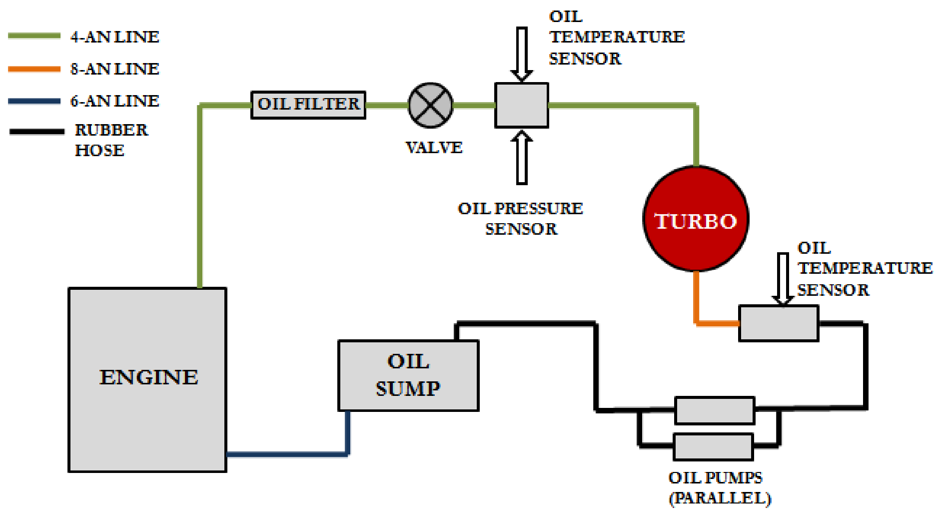


Figure 77: Oil system schematic used during testing

8.2.6 INTAKE

Another issue that slowed the testing was the intake plenum coming off because of the boost pressure. The plenum would slip out of the silicon hose that we used to connect it to the engine and the turbocharger. We would reposition the intake plenum and tighten the hose clamp that was holding it in place, only to have the other end come off on the next run. Figure 78 shows the intake plenum and the blue silicon hose that we used.

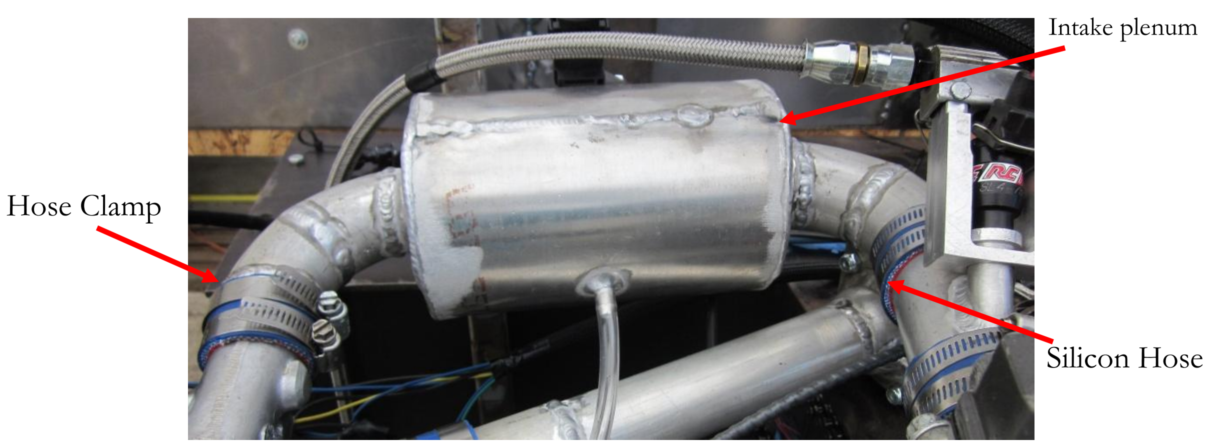


Figure 78: Initial intake plenum setup and attachment method

To solve this problem, radiator hose was used instead of silicon hose. We did this because the silicon hose was slipping out from under the clamp and the radiator hose is not as smooth as the silicon hose. This allowed the hose clamps to gain more grip on the hose and kept the intake from coming off, even when we hit our maximum boost pressure.

8.2.5 FUEL

Due to the characteristics of E-85 fuel, clogged injectors are a common problem. Out project was no exception. We had our injectors clog several times during testing and we had to clean them periodically to avoid it. To clean the injector, we removed it from the engine and connected it to a hose filled with carburetor cleaner. Using a bike pump, the carburetor cleaner was pressurized and the injector was then connect to the ECU so that the injector's valve could be opened, allowing the cleaner to flow through the injector. Figure 79 shows how this was setup.

Figure 79: Cleaning injector with pressurized hose and carb cleaner.

Eventually, a new injector was purchased when there was too much buildup in the injector to be cleaned sufficiently by our method. After we researched the issue, we found an E-85 fuel treatment that is designed to target this problem. Unfortunately, we did not discover this until the closing days of the project so we cannot report on the effectiveness.

8.2.6 TURBOCHARGER

One final problem, although minor, was the turbocharger's variable vanes not actuating smoothly like they should. This problem could become very severe because if the vanes do not open as the engine speed increases, the turbocharger's rotational speed will increase until some kind of catastrophic event. Fortunately, the vanes on our turbocharger were opening but it was not smooth and they seemed to be sticking at different points through their range of motion. Seeing no external obstruction that would cause such a problem, the turbocharger was taken apart and thoroughly cleaned. We believe that the new materials used for the exhaust system produced unusual exhaust gases that buildup inside the turbine housing, because we never encountered this problem again.

8.3 Dyno Testing Procedure/Calibration

In order to ensure accurate results from the dynamometer, FSAE had previously calibrated the water brake load cell. Although Formula had performed this, we'd like to mention that the dyno was indeed calibrated before our testing began. This procedure involved hanging weights from the load cell, which had a 12" moment arm, and changing gain values until the torque value was accurate.

As mentioned in 8.2.2, the dyno was re-calibrated for 4^{th} gear operation due to problems with the engine's 5^{th} gear. This consisted of shifting into 4^{th} gear, releasing the clutch and allowing the dyno to spin. We changed drive ratios until the RPM from the dyno and the engine's RPM were nearly identical. This method is used because calculating the drive ratio would leave us with RPM values farther apart then calibrating it by visually matching RPM.

8.4 Dyno Test Results

The results from dyno testing gave valuable insight into how the subsystems performed. A major observation was that the oil system pulling oil off of the engine's system did not supply the turbocharger with enough pressure. The values are shown below in Table 10.

Table 10: Oil Pressure Values

Engine Speed	Required	Actual
Idle	9psi	3psi
Peak Torque	25psi	17psi

One design flaw noted during engine testing was that the variable vane design of the turbine was not capable of controlling the boost levels that we wanted. The vanes were set to be fully open when the intake reached 4 psi of boost, but intake pressure continued to climb all the way to 15 psi of boost. In order to better control the boost a wastegate would have to be installed.

The engine ran for a combined total of 7 hours, partially on gasoline. Over 3 hours total were spent under load and more than 15 gallons of E-85 were burned during testing. Figure 80 shows the engine running with the turbocharger under load. The fuel map was tuned at every point above 3000 rpm from vacuum (no load) to 7 psi of boost (part load). Ignition timing was tuned with the goal of reducing exhaust gas temperatures (EGTs) to below 1600°F in order to prevent damage to the turbine vanes. While there have been unofficial claims on the Formula SAE forums of the GT15V surviving EGTs in excess of 2000°F, we decided it would be better to follow recommendations from Honeywell and keep them lower. Constant problems such as starter motors, oil consumption, excess boost pressure, and injector clogging prevented further run time and tuning.

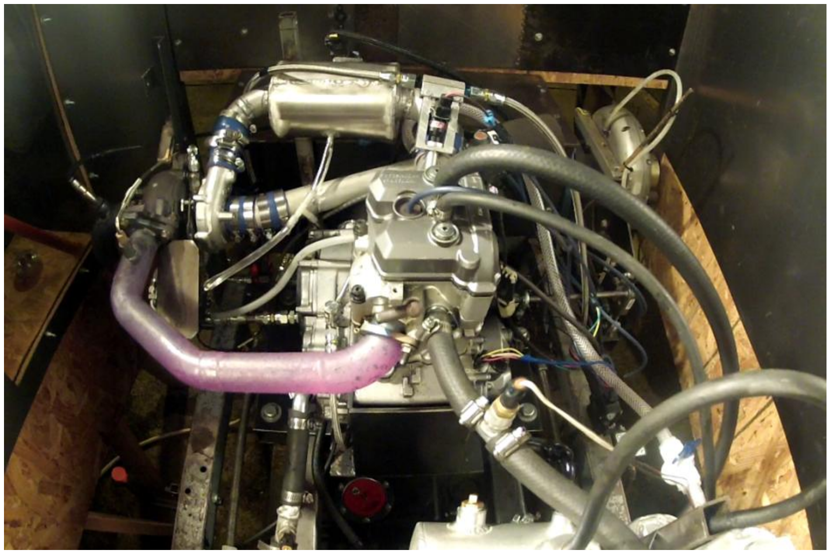


Figure 80. Turbocharged Engine Running Under Load.

While not enough time was available to fully tune the engine and generate an entire torque curve, the engine did provide a single point proving that it is capable of producing high levels of power: it produced 40 ft-lb of torque and 55 horsepower at 7200 RPM. It produced this power with 15 lbs of boost and approximately 80% throttle. While the output is close to the predicted power output at 7200 RPM (39 ft-lb and 54 hp), the operating conditions were very different than those in the simulation which ran at 8 psi of boost. In order to see if the model is still valid, another simulation was run with 15 psi of boost. The results are shown in Figure 81.

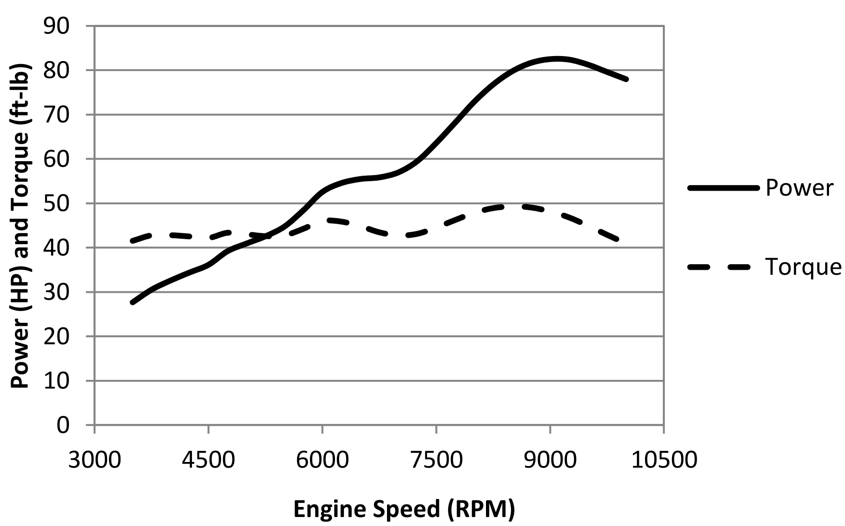


Figure 81: Simulated engine output with 15 psi of boost

According to the simulation, the engine should produce 42 ft-lb and 58 hp at 7200 RPM with 15 psi of boost. The engine produced 40 hp and 55 hp at 7200 RPM on the dyno with the same amount of boost. This is only one point, not an entire data range, but the correlation is still present between the simulation and the test results which suggest the simulation model is a valid approximation.

10.0 Cost

10.1 Projected vs. Actual Cost

The actual cost is higher than the projected cost because the scope of the project was expanded beyond its original bounds. Originally, the project involved building a turbocharger system around one engine. Since then the scope has expanded to include the purchase and subsequent strengthening of a second engine as well as the construction of a safety shield around the dynamometer. Figure 82 shows the cost breakdown and where the money was spent.

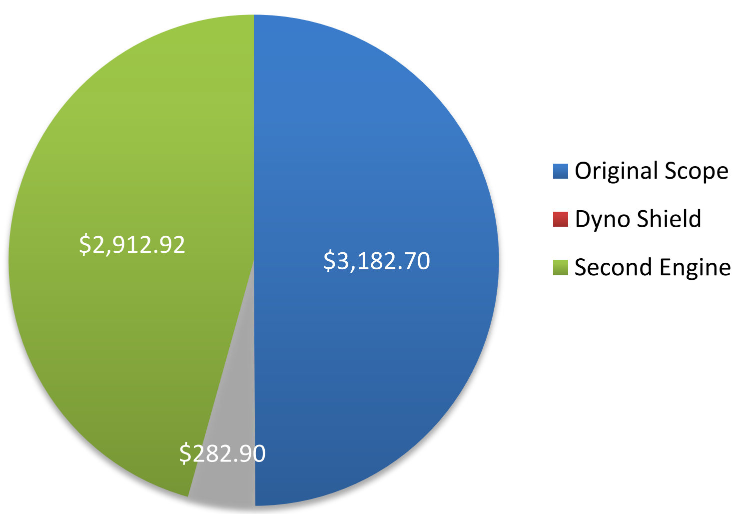


Figure 82. Division of Project Final Cost.

Within the scope of the original project, the total cost of to FSAE was $2032.70 after funding from MESFAC. Out of a total $6378.52 spent, the total cost to FSAE was $3538.52 after funding form MESFAC. The original goal was to keep the cost of this system to FSAE under $1000, so even when only the original scope is taken into consideration, the project cost FSAE more than twice as much as initially projected. Each purchase was approved by the team lead of FSAE and the cost is partially offset in the budget by not having to spend this money in the future. The cost of this senior project is the development cost for the FSAE powertrain that can be used for years to come.

11.0 Conclusion

The turbocharger system that we built around the WR450 engine proved to be capable of producing large increases in power output relative to the naturally aspirated engine. Even though full torque and power curves were not obtained from testing, the engine still produced power comparable with projections. With this, we are confident in the validity of the WAVE simulation model and therefore the ability of this system to match projected power output.

The GT15V compressor proved to be a good match for the engine, but the vanes in the turbine could not regulate the boost at the desired levels. Since an external wastegate is necessary for it to operate at the correct level, we recommend that the Formula SAE team switch to a GT12-41 turbocharger for continued refinement. The GT12-41 offers an integrated wastegate built into the turbine housing to manage boost. This would greatly simplify the control system in relation to adding an external wastegate to the GT15V, which would require control systems for both the vanes and the wastegate. Additionally, the GT12-41 would be several pounds lighter, partially because the unit itself weighs 2 pounds less and partially because it would not require the added weight of an external wastegate.

A potential downside with switching to the GT12-41 is the possibility for longer time for the turbocharger to spool up and build boost. Variable vanes are designed to make the turbine react as quickly as possible instead of dumping potential energy past the turbine through a wastegate. Theoretically this sounds like the perfect scenario for an engine, but the reality is that the GT12-41 is so small anyway that it will still spool up very quickly even without variable vanes. The GT15V was set up so that its vanes would be fully open at 4 psi of boost. This means that at every operating point above 4 psi of boost the vanes were at a constant position and no longer acting like a variable vane turbine. Boost response was still very quick in spite of this fact so we feel that switching to the GT12-41 will not cause any significant increase in turbo lag.

The largest unresolved problems with this system revolved around the oiling system. The first was the insufficient oil pressure feeding the turbocharger. Further research has shown that there is an oil bypass valve located in the right outer engine case cover which can be shimmed to produce more oil pressure. Additionally, a high output oil pump from a 2007-2009 YFZ450 is a direct bolt on replacement for the WR450 oil pump and is capable of supplying much more oil. The Yamaha part number for this oil pump is 5D3-13300-00-00.

The next issue was oil leaking past the compressor seal. Large volumes of oil would be sucked into the intake tract and burned by the engine regardless of what we tried. This issue needs to be investigated further as it presents a potentially serious problem if so much oil is burned that none is left to lubricate the engine. Potential solutions are outlined in reference [13] but their compliance with FSAE rules must first be determined.

Further testing time is required to refine crucial turbocharger subsystems before the turbocharged engine can become a reliable powerplant for the Formula car. This engine is capable of serving as the solid base for future iterations of turbocharger development at Cal Poly FSAE.

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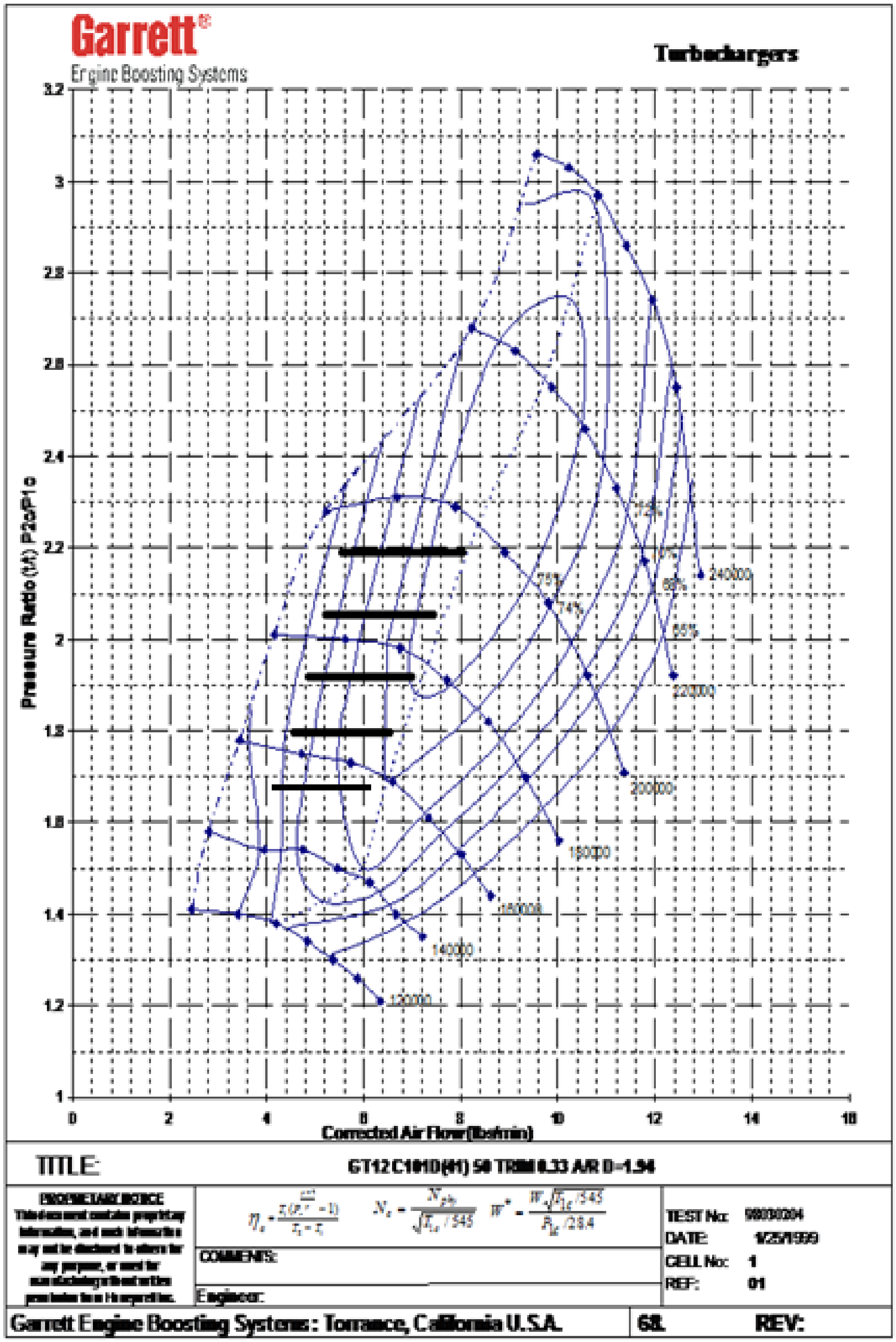
Appendix A: Quality Function Deployment (QFD)

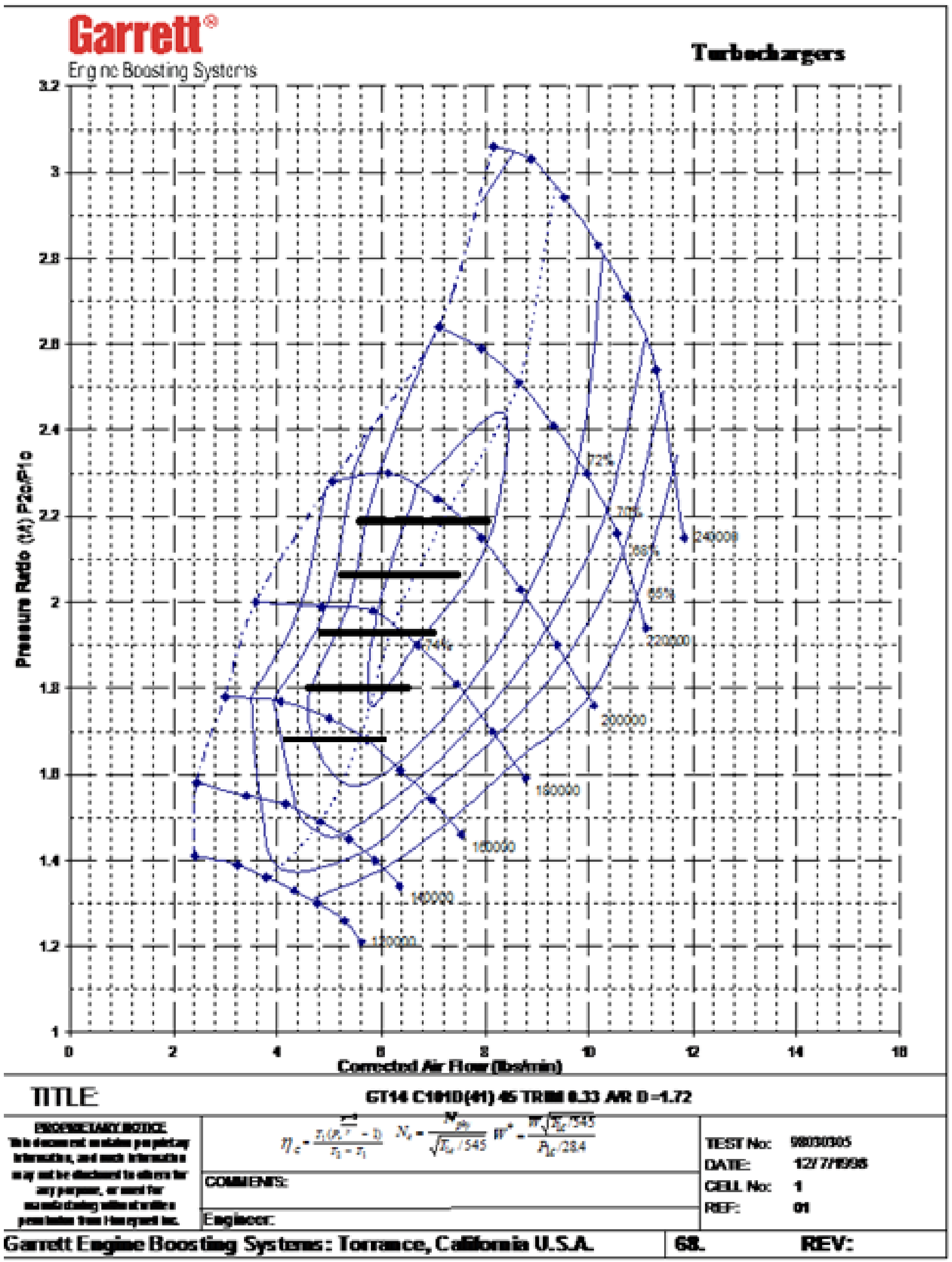
			Measures												
			Peak Power (HP)	Peak Torque (lb*ft)	Weight (lb)	Time to start (sec)	Starts/Start attempts	Installation time (min)	testing time (hr)	Exhaust Noise (dB)					
			A	B	C	D	E	F	G	H	1	2	3	4	5
Meet FSAE Rules	1	5								9					
Cost	2	4			3				1	1					
Overall Reliability	3	5			3		9		9						
Power Output	4	4	9	9					9						
Low Weight	5	3			9										
Dynamic Events	6	4	3	3	9	3		1	9						
Predictable Throttle Response	7	4	3	3				3							
Starts every time	8	4				9									
Ease of maintenance	9	3			1			9							
	10														
	11														
	12														
	13														
	14														
	15														
Strong - 9	Good	5													
Medium-3		4													
Weak - 1	Company Ratings	3													
		2													
Relationship Strength	Bad	1													
Targets			>55 hp	>35 lb*ft	<20 lbs	3 sec	100%	<10 min	>40 hours	<110 dB @ 7000 rpm					
Weighted Importance			60	60	93	48	45	43	121	49		519			
% Importance			12	12	18	9.2	8.7	8.3	23	9.4					

Appendix B: Compressor Map Results

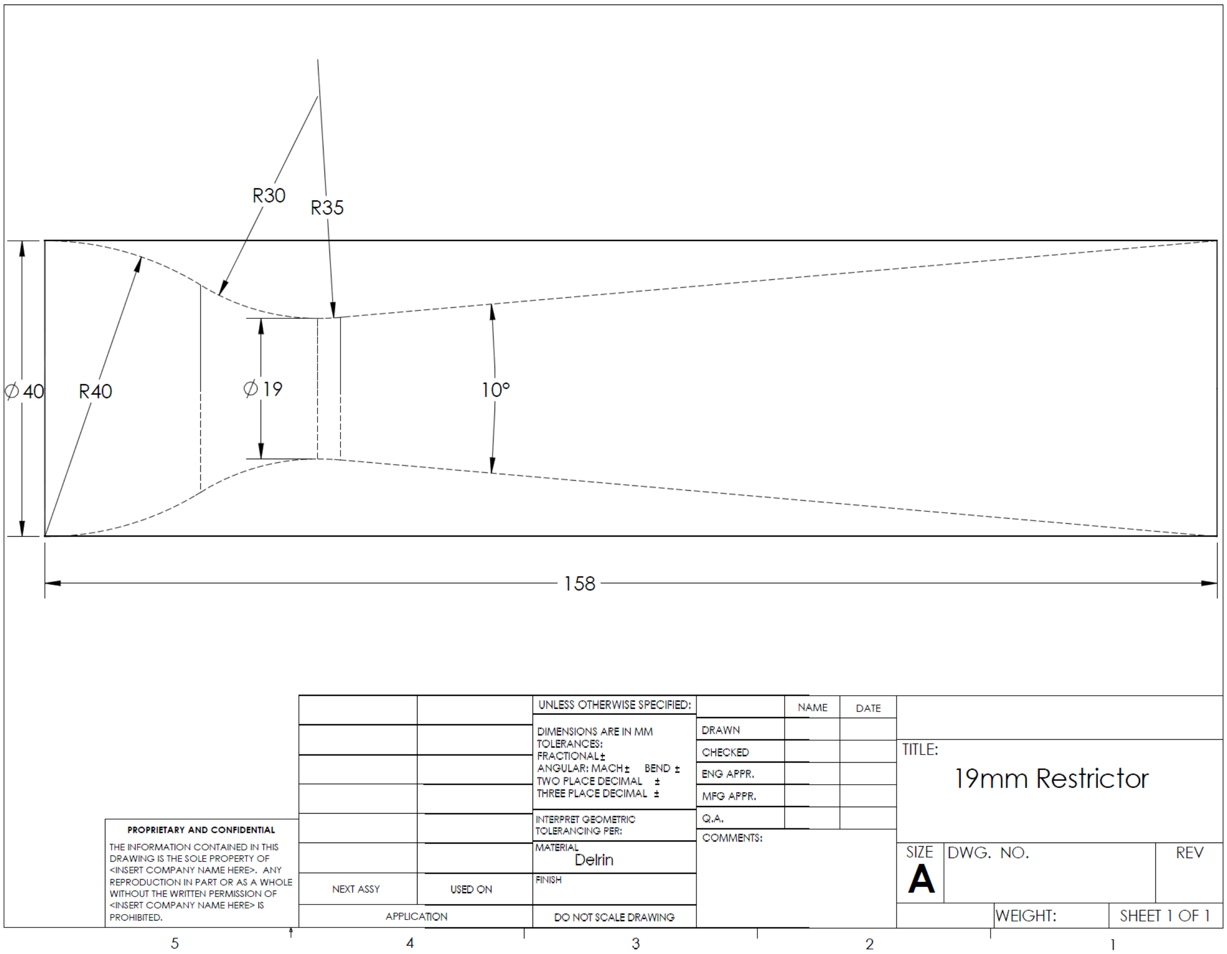
GT12-41 Compressor:



GT15V Compressor:



Appendix C: Restrictor Dimensions



Appendix D: Turbocharger Selection Calculations

Calculate Pressure Ratio + Airflow

Horsepower Target: 70

Engine Displacement, V_d = 449cc / 27.4 in³

Max Engine Speed, N = 9000 RPM

BSFC = 0.75 lb/hp·hr (estimate with E85)

Desired AFR = 8:1 (E85)

Actual Airflow, $W_a = HP \cdot AFR \cdot \frac{BSFC}{60}$ (Garrett 2009 catalog, p.9)

$$= 70 \cdot 8 \cdot \frac{.75}{60}$$

$$W_a = 7.0 \text{ lb/min}$$

Gas constant, R = 639.6 in/°R

Intake Manifold Temp, T_m = 100°F (no intercooler, cooling from fuel spray)

Volumetric Efficiency: 0.95 (5-valve head)

Required Manifold Pressure, $MAP_{req} = \frac{W_a \cdot R \cdot (460 + T_m)}{VE \cdot N/2 \cdot V_d}$ (p. 10)

$$= \frac{7.0 \text{ lb/min} \cdot 639.6 \text{ in/°R } (460+100) \text{ °R}}{0.95 \cdot \frac{9000}{2} \frac{rev}{min} \cdot 27.4 \text{ in}^3/\text{rev}}$$

$$MAP_{req} = 21.4 \text{ psi}$$

No intercooler – assume 1 psi drop from compressor outlet to manifold. ΔP_{loss} = 1 psi

Compressor Discharge Pressure, $P_{2c} = MAP_{req} + \Delta P_{loss}$ (p. 10)

$$P_{2c} = 22.4 \text{ psi}$$

1/2

Ambient Pressure, $P_{atm} = 14.1$ psi (Lincoln, Nebraska)

- Assume 0.5 psi pressure drop due to air filter and throttle body. $\Delta P_{intake} = 0.5$ psi
- Assume 15% loss in pressure due to 19mm restrictor. $\eta_r = 0.85$ (Data ranges from 10-20%)

Compressor Inlet Pressure, $P_{1c} = (P_{atm} - \Delta P_{intake})\,\eta_r$

$$P_{1c} = (14.1\text{ psi} - 0.5\text{ psi})\,0.85$$

$$P_{1c} = 11.6\text{ psi}$$

Pressure Ratio, $\pi_c = \frac{P_{2c}}{P_{1c}}$

$$\pi_c = \frac{22.4\text{ psi}}{11.6\text{ psi}}$$

$$\pi_c = 1.93$$

Now Calculate W_a for $N = 6000$ RPM (max torque)

$$(W_a)_e = \frac{MAP_{req} \cdot VE \cdot \frac{N}{2} \cdot V_d}{R \cdot (460 + T_m)}$$

$$(W_a)_e = \frac{21.4\text{ psi} \cdot 0.95 \cdot \frac{6000}{2}\frac{\text{rev}}{\text{min}} \cdot 27.4\,\text{in}^3/\text{rev}}{639.6\text{ in}/{}^\circ R\,(460 + 100)^\circ R}$$

$$(W_a)_e = 4.88\text{ lb/min}$$

Various HP targets:

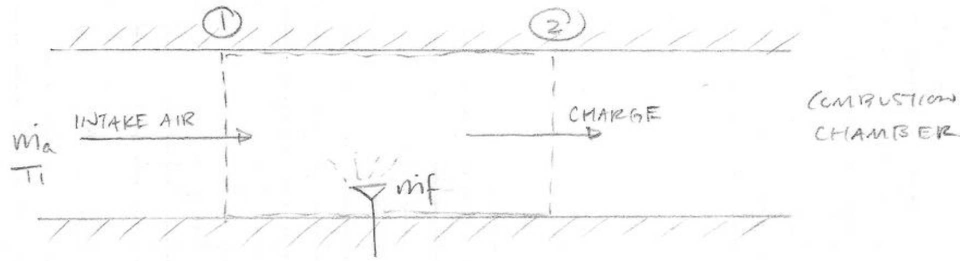
	65 HP	70 HP	75 HP	80 HP
π_c	1.80	1.93	2.06	2.19
W_a	6.5 lb/min	7.0 lb/min	7.5 lb/min	8.0 lb/min
$(W_a)_e$	4.55 lb/min	4.88 lb/min	5.22 lb/min	5.55 lb/min

Appendix E: Ideal Temperature Drop Across Injector Calculations

SLO RACING ①

CALCULATION | TEMPERATURE DROP OF INTAKE CHARGE

SCHEMATIC



GIVEN |

Compressed intake air @ STATE 1, fuel injected ($\dot{m}_f$), intake charge enters combustion chamber.

FIND | Theoretical temperature drop of intake air when both a) E85 and b) GASOLINE is injected (under same cond.)

ASSUMPTIONS |

[A_1] ⟶ All the fuel is vaporized.

[A_2] ⟶ A/F ratio is 8 for E85, 12 for gasoline (recommended for turbocharged applications)

[A_3] ⟶ No heat exchange from surroundings

[A_4] ⟶ Same conditions for both fuels

ANALYSIS |

→ 1st law on control volume

$$\dot{E}_{in} - \dot{E}_{out} + \dot{E}_{gen} \begin{cases} \dot{H}_1 - \dot{H}_2 = \Delta\dot{H} = \dot{m}_f (h_{fg}) \\ \Delta\dot{E}_{st} = [\dot{m}_a C_p \Delta T] \end{cases}$$

$$\dot{m}_f (h_{fg})_e = \dot{m}_a C_p \Delta T$$

$$\Delta T = \frac{\dot{m}_f (h_{fg})}{\dot{m}_a C_p} \quad \left\{ \frac{\dot{m}_f}{\dot{m}_a} = \frac{1}{AFR} \longrightarrow \begin{matrix} \left(\frac{1}{8}\right) - E85 \\ \left(\frac{1}{12}\right) - gas \end{matrix} \right.$$

→ h_{fg} — latent heat of vaporization

$$(h_{fg})_{E85} = \frac{766.5\ kJ}{kg}$$

$$(h_{fg})_{gasoline} = \frac{350\ kJ}{kg}$$

NOTE: Rough weighted approximation because specific E85 values could not be found.

E85 |

$$\Delta T)_{E85} = \frac{(h_{fg})_{E85}}{(AFR)_{E85} C_p} = \frac{\left(766.5 \frac{kJ}{kg}\right)}{8\left(1.0 \frac{kJ}{kg\,K}\right)} = \boxed{95.8K = (\Delta T)_{E85}}$$

(172.8 °F)

GASOLINE |

$$\Delta T)_{GAS} = \frac{(h_{fg})_{GAS}}{(AFR)_{E85} C_p} = \frac{350 \frac{kJ}{kg}}{12\left(1.0 \frac{kJ}{kg\,K}\right)} = \boxed{29.17K = (\Delta T)_{gas}}$$

(52.5 °F)

Appendix F: Restrictor Maximum Flow Rate Calculation

CALCULATING MAX AIRFLOW THROUGH RESTRICTOR

RESTRICTOR DIAMETER = 19 mm

FOR CHOKED FLOW:

$$\dot{m}_{choke} = A_t P_o \left(\frac{k}{R T_o}\right)^{0.5} \left(\frac{2}{k+1}\right)^{\frac{(k+1)}{2(k-1)}}$$

WHERE: $A_t \equiv$ Throat Area
$P_o \equiv$ initial pressure
$T_o \equiv$ initial temp.
$R \equiv$ Gas Constant
$k \equiv$ Specific Heat Ratio

$A_t = \frac{\pi}{4} D^2 = \frac{\pi}{4}(0.019\,m)^2 = 2.835 \times 10^{-4}\,m^2$

$P_o = 97$ kPa (Pressure for Lincoln, NE at 1219ft)

$T_o = 302.6$ K (Average Temp for Lincoln, NE in June)

$R = 287\ J/kg \cdot K$ (FOR AIR)

$k = 1.4$ (FOR AIR)

$$\dot{m}_{choke} = 2.835 \times 10^{-4} m^2 (97.0\,kPa) \left(\frac{1.4}{287\,J/kg \cdot K\,(302.6K)}\right)^{0.5} \left(\frac{2}{1.4+1}\right)^{\frac{(1.4+1)}{2(1.4-1)}}$$

$$\dot{m}_{choke} = 0.0639\ kg/s$$

$$\dot{m}_{choke} = 0.0639 \frac{kg}{s} \left(\frac{2.205\,lb}{1\,kg}\right)\left(\frac{60\,s}{1\,min}\right)$$

$$\boxed{\dot{m}_{choke} = 8.45\ lb/min}$$

FOR AVERAGE ALTITUDE & TEMP (IN JUNE) IN LINCOLN, NE

Appendix G: Cost Breakdown of Project Budget

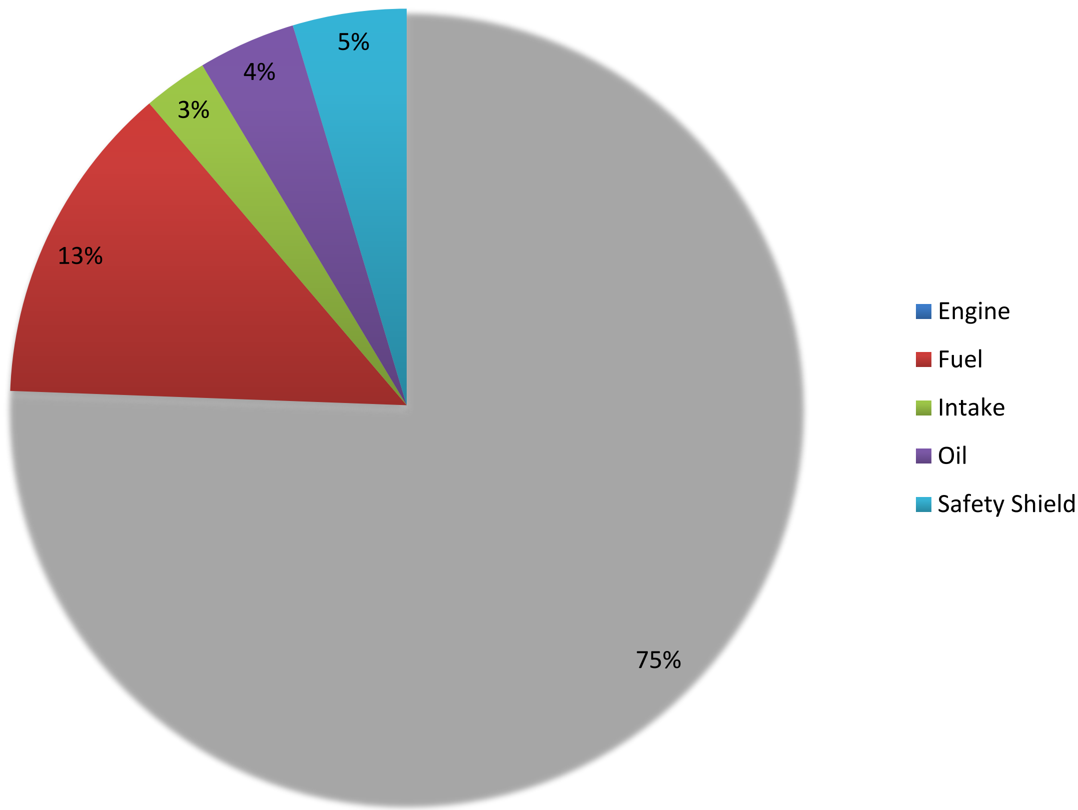


Table 11-G: Engine Cost Breakdown

Description	Supplier	Part #	Price	Quantity	Subtotal	Total
Ron Woods HD Clutch Springs	rotax.net	call	39.95	2	79.90	85.69
Kibblewhite Valve Kit	RM ATV	1168330008	233.99	2	467.98	467.98
Gasket Set	RM ATV	1321050037	39.99	4	159.96	159.96
Countershaft Seal	RM ATV	1348810018	14.99	2	29.98	29.98
Yamaha OEM Main Bearings	RM ATV	93306-30633-00	40.81	4	163.24	163.24
Transmission Bearing 1	CCP	93306-20532-00	21.26	1	21.26	21.26
Transmission Bearing 2	CCP	93306-00431-00	14.40	1	14.40	14.40
Transmission Bearing 3	CCP	93306-00426-00	14.40	1	14.40	14.40
Transmission Bearing 4	CCP	93306-20593-00	21.26	1	21.26	21.26
Transmission Oil Seal	CCP	93102-32480-00	7.85	1	7.85	7.85
Water Pump Seal 1	CCP	93109-11073-00	4.85	2	9.70	9.70
Water Pump Seal 2	CCP	93102-12321-00	5.20	2	10.40	10.40
Oil Filter O-Ring	CCP	93210-07135-00	2.97	6	17.82	17.82
Oil Filter	CCP	5TA1344000	4.43	5	22.15	22.15
Hinson Pressure Plate	Hinson	H099	189.99	2	379.98	407.53
Carrillo Connecting Rod	CP-Carrillo	Custom # 61949	250.00	2	500.00	500.00
CP Piston	CP-Carrillo	Order # 170187	180.00	2	360.00	360.00
Weld and Balance Crank	Q and E	n/a	722.00	1	722.00	722.00
ARP Studs	Hillco Performance	ARP107500125B	18.99	4	75.96	81.47
ARP Studs	Hillco Performance	ARP106450125B	9.98	4	39.92	42.81
ARP Nuts	Hillco Performance	ARP10125LB	5.50	8	44.00	47.19
ARP Washers	Hillco Performance	ARP200-8705	1.44	8	11.52	12.36
Spark Plugs	Cycle Gear	CR9EIX		4	0.00	0.00
Engine #2	ebay	n/a	891.99	1	891.99	891.99
OEM Main Bearing	Mission Motorsports	93310-63417-00	63.25	2	126.50	135.67

Starter Clutch	Mission Motorsports	5TJ-15590-00-00	108.58	1	108.58	116.45
Starter Clutch Bolt	Mission Motorsports	91317-06014-00	1.88	6	11.28	12.10
Balance Shaft Bearing	Mission Motorsports	93306-20335-00	14.81	1	14.81	15.88
Valve Spring Compressor	RMATV	1168670001	34.99	1	34.99	34.99
Clutch Plate	RMATV	131204000320	8.71	4	34.84	34.84
Clutch Plate	RMATV	131204004755	7.90	3	23.70	23.70
O-Ring	RMATV	131204003797	2.58	2	5.16	5.16
Flywheel Case Cover	RMATV	131204002202	140.56	1	140.56	140.56
Oil Squirter O-Ring	RMATV	131204003833	1.98	1	1.98	1.98
Timing Plug	RMATV	131204003282	5.58	2	11.16	11.16
Crank Cover Plug	RMATV	131204003282	7.58	1	7.58	7.58
Flywheel Nut Washer	RMATV	131204003141	2.43	1	2.43	2.43
Flywheel Nut	RMATV	131204002997	2.66	2	5.32	5.32
Oil Squirter	RMATV	131204001486	23.56	1	23.56	23.56
HotCams Valve Shim Set	Private Seller	n/a	35.00	1	35.00	35.00
Case Saver	Motorcycle Superstore	203115	42.95	2	85.90	85.90
Valve Guide	RMATV	131204002058	24.44	1	24.44	24.44
Valve Guide Clip	RMATV	131204004035	0.90	1	0.90	0.90
Head Gasket	RMATV	1009120143	36.99	2	73.98	73.98
					Total:	4903.04

Table 12-G: Fuel Cost Breakdown

Description	Supplier	Part #	Price	Quantity	Subtotal	Total
Fuel Hose	JEGS	555110912	59.99	1	59.99	59.99
6-AN 90 deg. Fitting	JEGS	555110021	13.99	4	55.96	55.96
NPT to 6-AN Fitting	JEGS	555110104	2.99	4	11.96	11.96

Fuel Regulator	Aeromotive		131.33	1	131.33	140.85
Fuel Pump	Aeromotive		228.41	1	228.41	244.97
Fuel Injector	RC		0.00	1	0.00	0.00
100 Micron Fuel Filter	Summit	AEI-12316	73.95	1	73.95	73.95
6AN to 5/16 Barb	Summit	AEI-15635	21.95	1	21.95	21.95
6AN Fitting	Summit	AER-FBM1103	6.95	17	118.15	118.15
PTFE Fuel Line	Summit	AER-FCC0615	81.95	1	81.95	81.95
5 Gal Fuel Cell	Summit	SUM-290101	94.95	1	94.95	94.95
Shipping/Handling	Summit	n/a	12.95	1	12.95	12.95
					Total:	917.63

Table 13-G: Oiling System Cost Breakdown

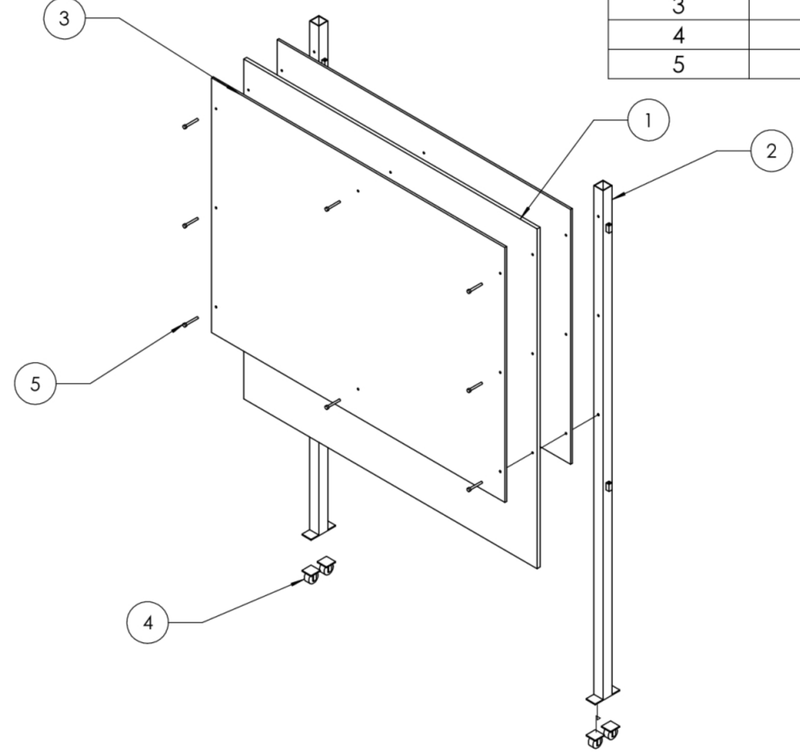
Description	Supplier	Part #	Price	Quantity	Subtotal	Total
In-Line Oil Filter	forced performance		77.27	1	77.27	82.87
Oil Lines	Summit	AER-FCC0406	21.79	1	21.79	21.79
Oil Lines	Summit	AER-FCC0606	35.95	1	35.95	35.95
PTFE Hose End	Summit	AER-FBM1101	5.75	6	34.50	34.50
PTFE Hose End	Summit	AER-FBM1103	6.95	4	27.80	27.80
6AN to 4AN Reducer	Summit	AER-FBM2408	6.25	1	6.25	6.25
6AN to 3/8 NPT	Summit	AER-FBM2521	2.75	1	2.75	2.75
4AN to 1/8 NPT	Summit	AER-FBM5001	2.75	3	8.25	8.25
6AN to 1/4 NPT	Summit	AER-FBM5004	3.95	3	11.85	11.85
4AN Male to male	Summit	AER-FBM5051	4.05	1	4.05	4.05
Double Banjo Bolt	Summit	EAR-977518ERL	8.99	1	8.99	8.99
Banjo Fitting	Summit	RUS-640481	10.99	1	10.99	10.99
4AN Coupler	Summit	SUM-220441B	6.95	1	6.95	6.95
Shipping/Handling	Summit	n/a	11.95	1	11.95	11.95
					Total:	274.94

Table 14-G: Safety Shield Cost Breakdown

Description	Supplier	Part #	Price	Quantity	Subtotal	Total
OSD Plywood	Home Depot		19.95	2	39.90	42.795
Lexan - Polycarbonate	Home Depot		19.98	3	59.94	64.28
Hardware	Home Depot				19.99	21.44
Caster Wheels	Home Depot		9.99	3	29.97	32.14
Mild Steel Square Tubing	B&B Specialty Metals		14.00	2	28.00	30.03
Gage Mild Steel	B&B Specialty Metals		42.99	2	85.98	92.21
					Total:	282.90

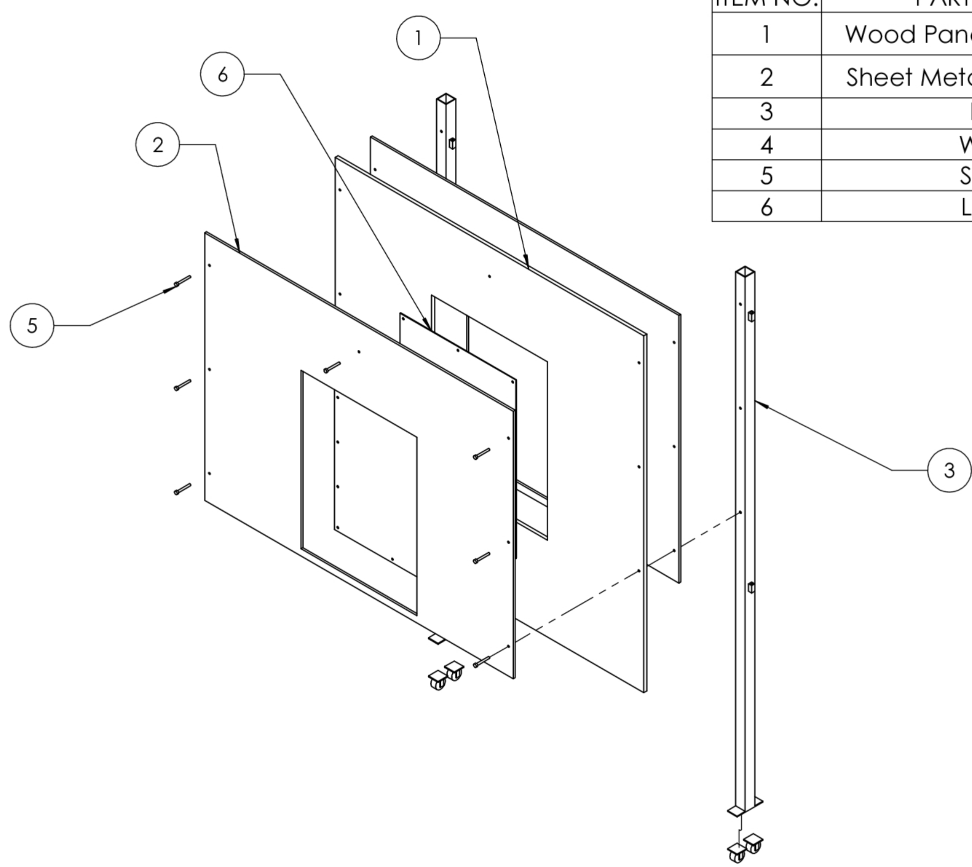
Appendix H: Safety Shield Drawings

ITEM NO.	PART NUMBER	QTY.
1	wood panel	1
2	Legs	2
3	Sheet Metal	2
4	Wheel	4
5	Screw	8



ME 430 Fall 2012	SCALE: 1:16	PART: Front with no window
Mechanical Engineering CAL POLY	UNITS: Inches	MATERIAL:
	TOLERANCE:	DATE: November 12, 2012
	DWG: 1.1.0	Team: Slo Racing

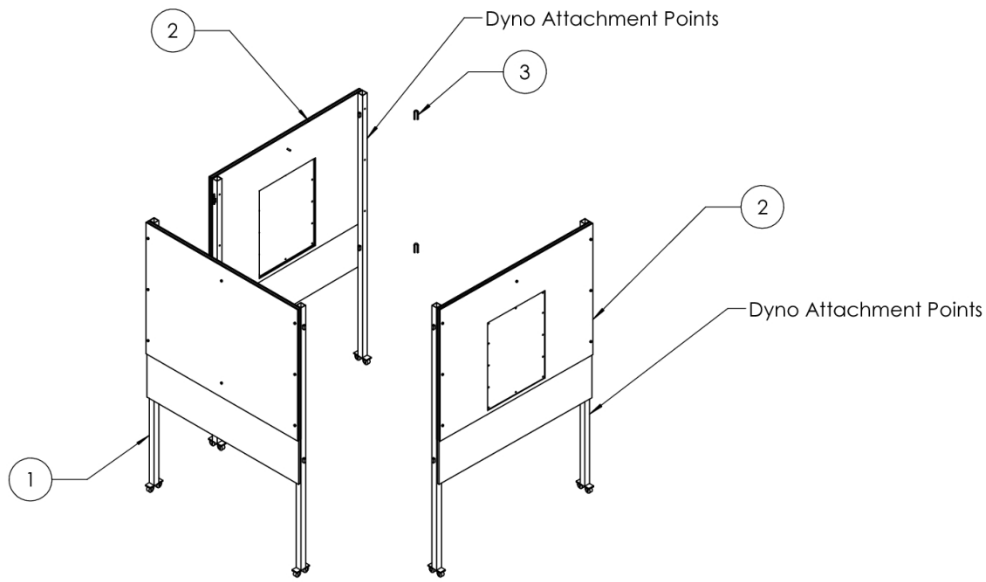
ITEM NO.	PART NUMBER	QTY.
1	Wood Panel with Window	1
2	Sheet Metal with Window	2
3	Legs	2
4	Wheel	4
5	Screw	7
6	Lexan	3



ME 430 Fall 2012	SCALE: 1:16	Sides with Windows
Mechanical Engineering CAL POLY	UNITS: Inches	MATERIAL:
	TOLERANCE:	DATE: November 12, 2012
	DWG: 1.2.0	Team: Slo Racing

5 4 3 2 1

ITEM NO.	PART NUMBER	QTY.
1	Front with no Window	1
2	Sides with Window	2
3	U bolt	4



ME 430 Fall 2012	SCALE: 1:32	PART: Safety Shield
Mechanical Engineering CAL POLY	UNITS: Inches	MATERIAL:
	TOLERANCE:	DATE: November 12, 2012
	DWG: 1.0.0	Team: Slo Racing

5 4 3 2 1

Appendix I: Dynamic Pressure Ratio Calculations and 11:1 CP Piston Spec. Sheet

CALCULATION OF DYNAMIC COMPRESSION RATIO

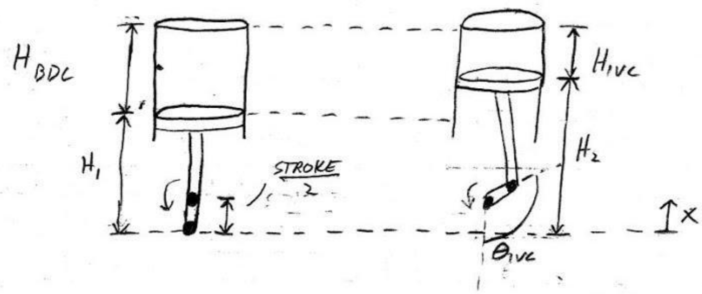
TERMINOLOGY: CR → COMPRESSION RATIO
TDC → TOP DEAD CENTER
BDC → BOTTOM DEAD CENTER
IVC → INTAKE VALVE CLOSED
NA → NATURALLY ASPIRATED
ATM → ATMOSPHERIC

STATIC $CR = \frac{V_{BDC}}{V_{TDC}}$ DISPLACEMENT $= V_{BDC} - V_{TDC}$

FOR STATIC CR = 11
AND
DISPLACEMENT = 449 cc (95 mm BORE × 63.4 mm STROKE)

$V_{BDC} = 493.9$ cc $V_{TDC} = 44.9$ cc

DYNAMIC $CR = \frac{V_{IVC}}{V_{TDC}}$ $V_{IVC} = H_{IVC} \cdot \frac{\pi (BORE)^2}{4}$



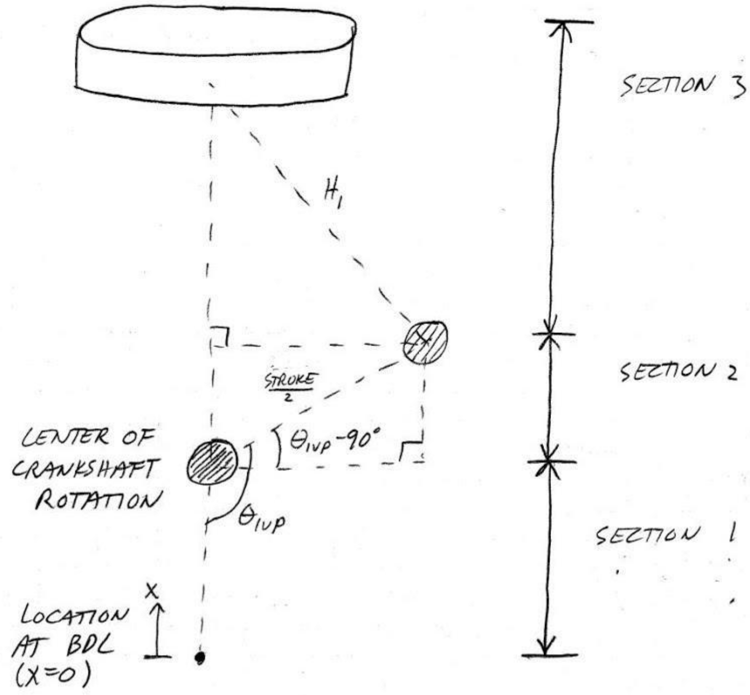
$H_{BDC} + H_1 = H_{IVC} + H_2$

$H_1 = 103.5$ mm (LENGTH OF CONNECTING ROD)

ACCORDING TO MEASUREMENTS, THE INTAKE VALVE CLOSES AT 100° AFTER BDC ⇒ $\theta_{IVC} = 100°$

1/3

H_2 CAN BE BROKEN DOWN INTO 3 SECTIONS:



$$\text{SECTION } 1 = \frac{STROKE}{2}$$

$$\text{SECTION } 2 = \frac{STROKE}{2} \sin\left(\theta_{IVC} - 90°\right)$$

$$\text{SECTION } 3 = \left[H_1^{\,2} - \left(\frac{STROKE}{2} \cos(\theta_{IVC} - 90°) \right)^2 \right]^{1/2}$$

SECTION 1 = 31.7 mm SECTION 2 = 5.5 mm

SECTION 3 = 98.7 mm

$H_2 = 135.9$ mm $H_{BDC} = V_{BDC} \cdot \frac{4}{\pi (BORE)^2}$

$H_{IVC} + H_2 = H_{BDC} + H_1$ $H_{BDC} = 69.7$ mm

$H_{IVC} = 37.3$ mm $V_{IVC} = 264.2$ cc

2/3

DYNAMIC CR = 5.88

- THIS IS ASSUMING 100% CYLINDER FILL AND NATURALLY ASPIRATED
- TO CALCULATE TURBOCHARGED DYNAMIC CR MULTIPLY VOLUME AT IVC BY CYLINDER FILL AND INTAKE PRESSURE

$$\text{TURBO DCR} = \frac{(\text{CYL FILL})\left(\frac{\text{INT. P}}{\text{ATM. P}}\right) V_{IVC}}{V_{TDC}}$$

- 5-VALVE CYLINDER HEADS SUCH AS THE ONE ON THE VR450 TYPICALLY ACHIEVE 95% CYLINDER FILL
- 8 PSI OF BOOST IS 22.7 PSI ABSOLUTE

$$\text{TURBO DCR} = \frac{(.95)\left(\frac{22.7\ \text{PSI}}{14.7\ \text{PSI}}\right)(264.2\text{cc})}{44.9\ \text{cc}}$$

$$\boxed{\text{TURBO DCR} = 8.63}$$

3/3

AMPAD

Specification Sheet

Date Printed: *6/6/2012*

Customer:	**Pistons (Out of California)**	Job Number:	**170187**
Customer ID:	**1008**	P.O. Number:	

Quantity:	**2**	Forging:	**MX94**	Shelf Part
Quantity:	**0**	Forging:		

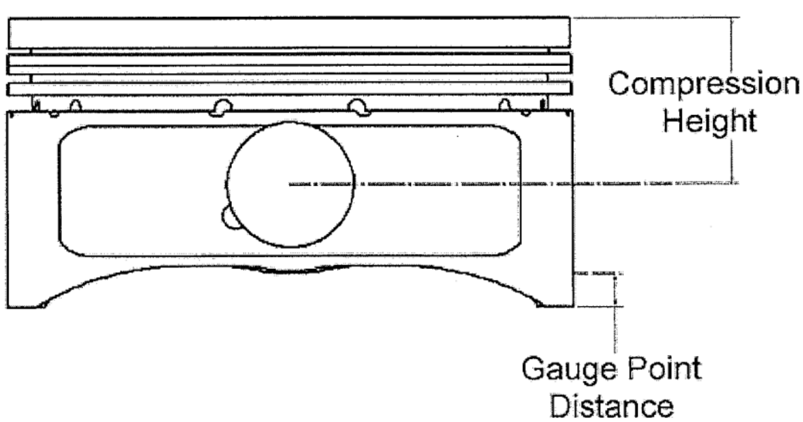
Bore Diameter:	**3.74**	Piston Size at Gauge Point 0.4 " up from Bottom of Skirt	**3.738**
Clearance:	**0.0020**		
Compression Height:	**0.918**		

Top Groove Width:	**0.04**	Top Groove Diameter:	**3.422**
2nd Groove Width:	**0.048**	2nd Groove Diameter:	**3.394**
3rd Groove Width:	**0.141**	3rd Groove Diameter:	**3.436**

Intake Valve Relief:	**-0.165**	Exhaust Valve Relief:	**-.185**

Profile:	·	Pin Diameter:	**0.708**
Dish Depth:	**-0.096**	Pin Length:	**2**
Dome Height:	**0**	Lock Type:	**.050 Wire**

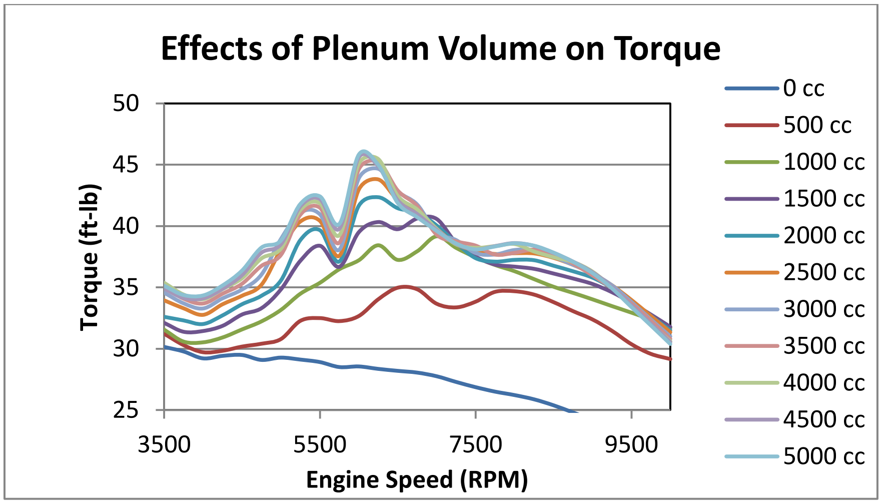
Gram Weight:	**295**	Cam Profile:	**10MC45M11**

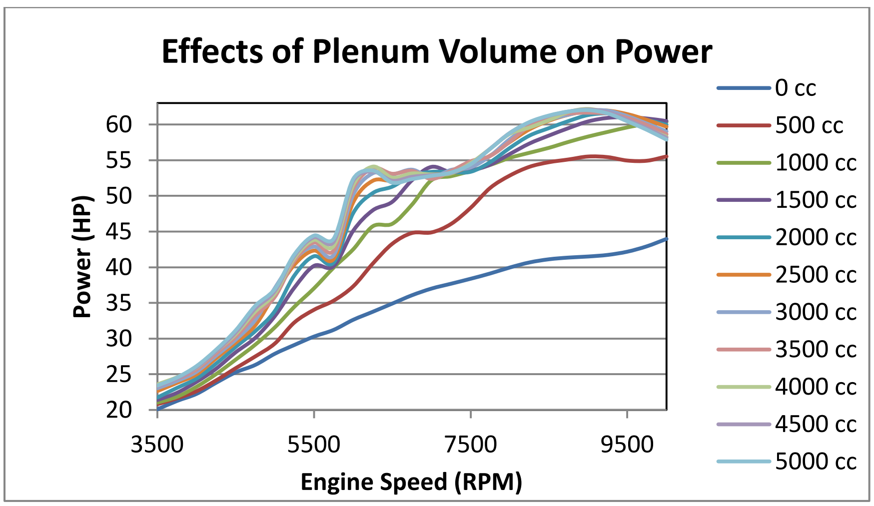


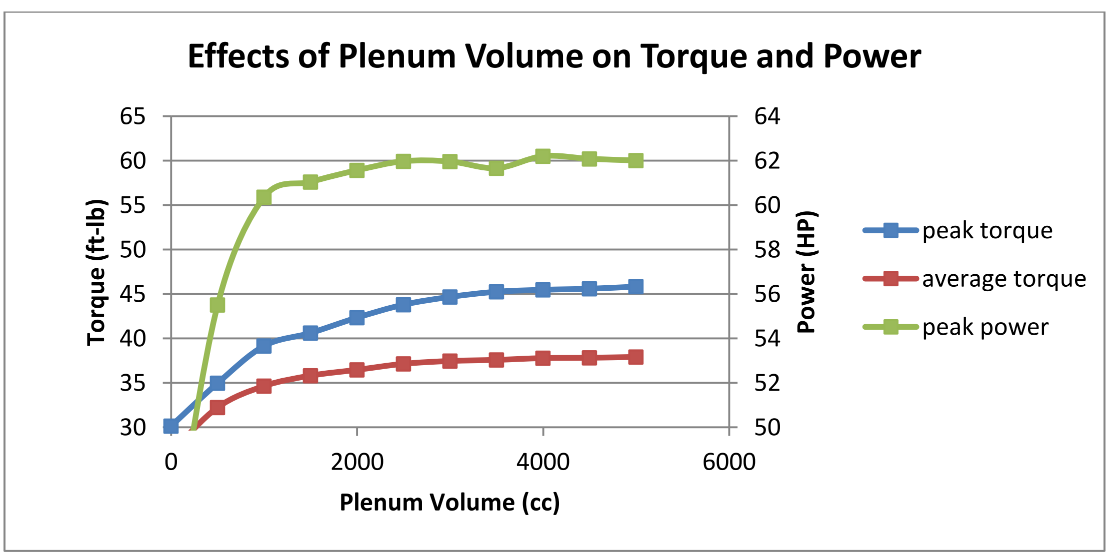
CP Pistons, LLC

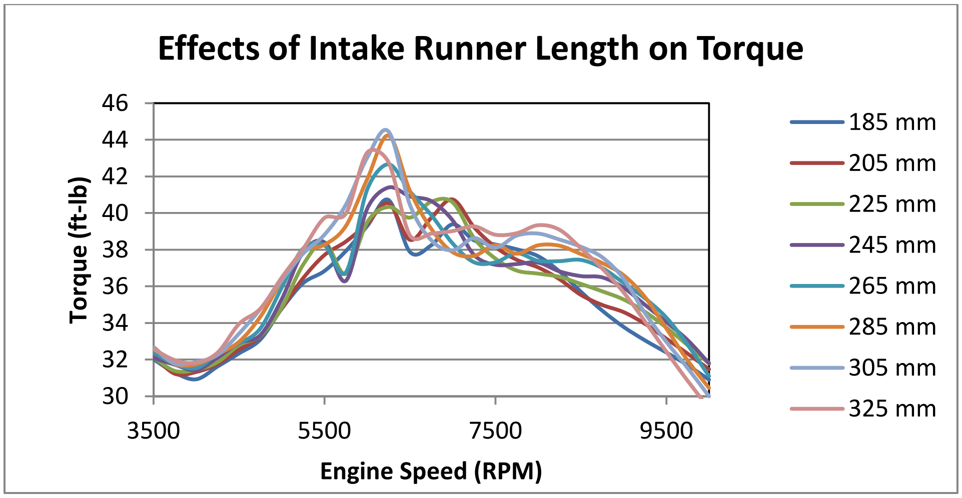
1902 McGaw Ave. - Irvine, CA 92614 949-567-9000 Fax 949-567-9010 www.cppistons.com

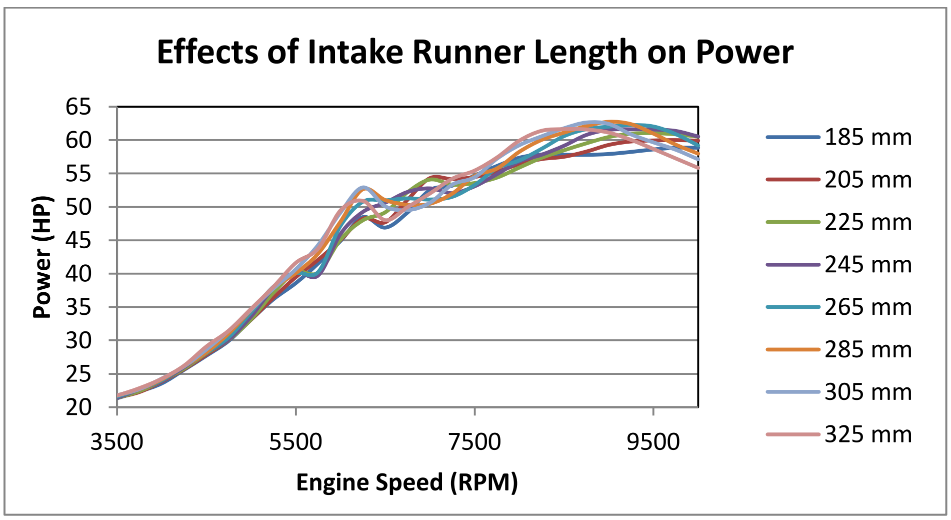
Appendix J: Ricardo WAVE Results and Values

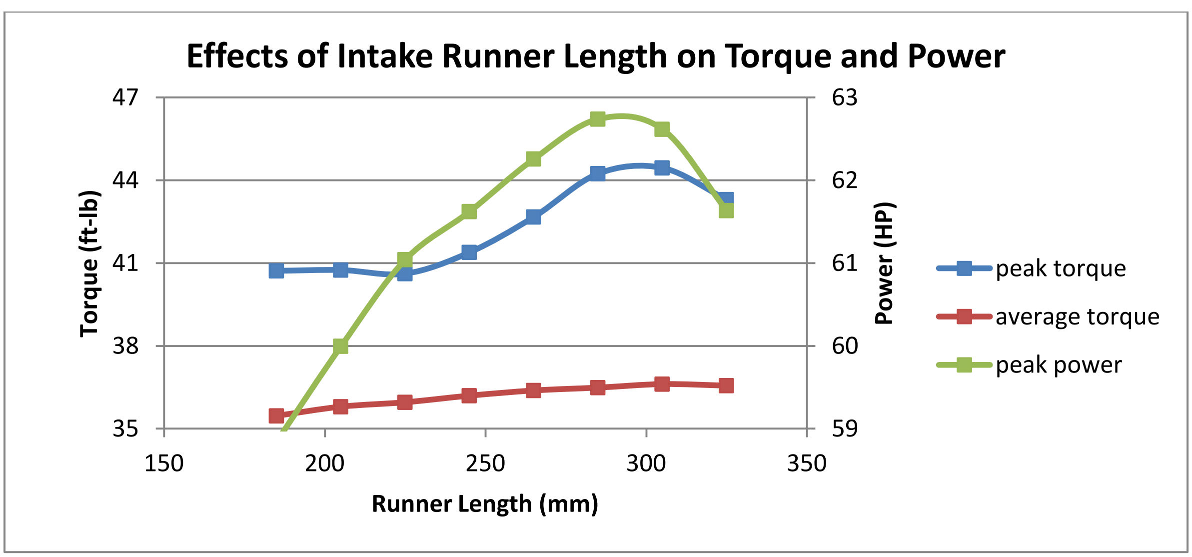


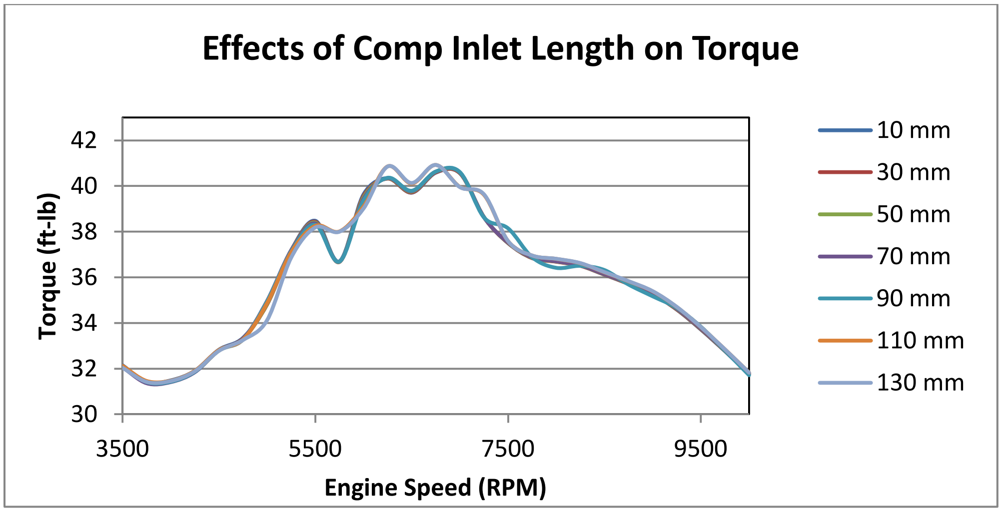


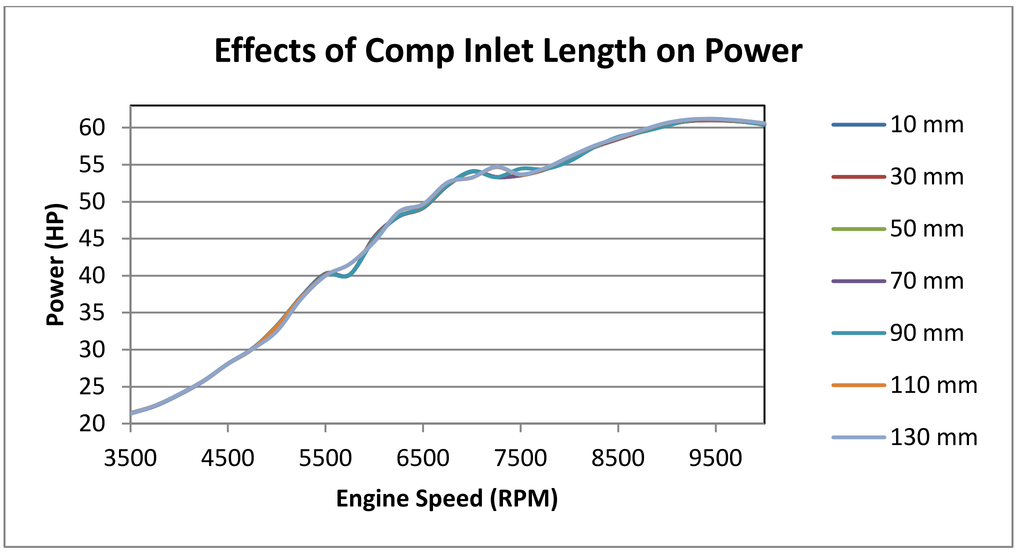


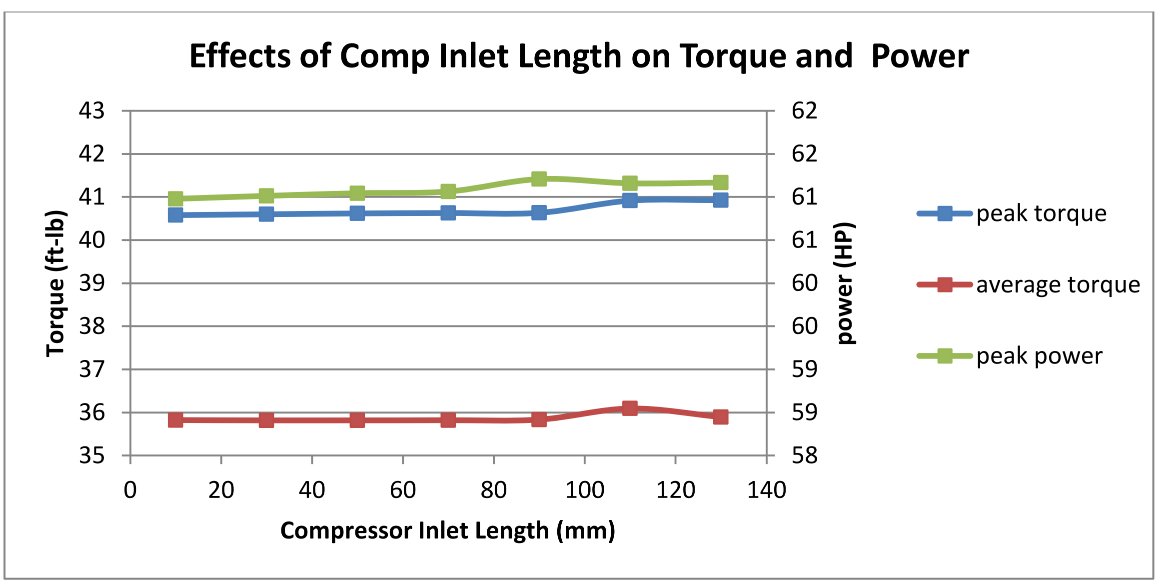

Effects of Intake Runner Length on Torque
Torque (ft-lb)
46
44
42
40
38
36
34
32
30
3500
5500
7500
9500
Engine Speed (RPM)
185 mm
205 mm
225 mm
245 mm
265 mm
285 mm
305 mm
325 mm

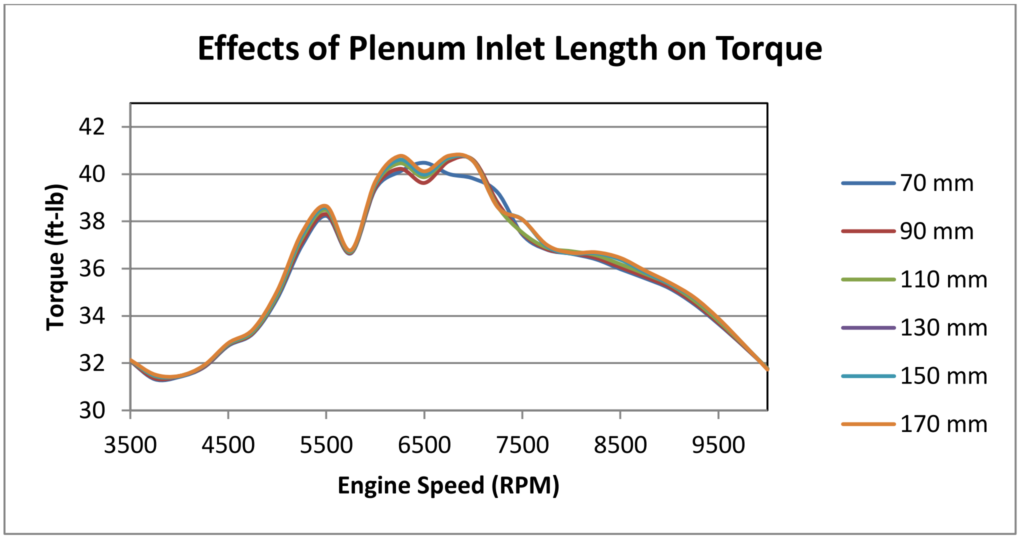

Effects of Intake Runner Length on Power
Power (HP)
65
60
55
50
45
40
35
30
25
20
3500
5500
7500
9500
Engine Speed (RPM)
185 mm
205 mm
225 mm
245 mm
265 mm
285 mm
305 mm
325 mm

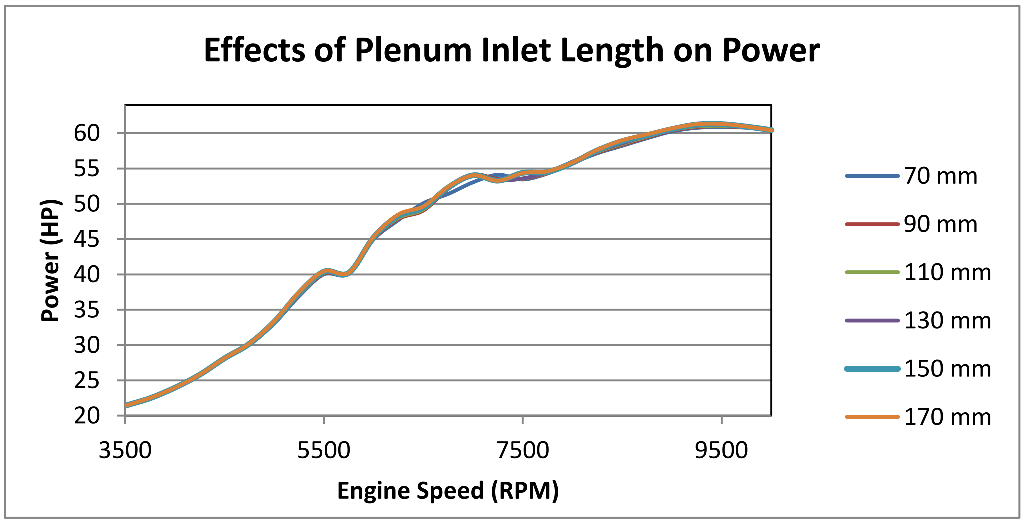

Effects of Intake Runner Length on Torque and Power
Torque (ft-lb)
47
44
41
38
35
150
200
250
300
350
Runner Length (mm)
Power (HP)
63
62
61
60
59
peak torque
average torque
peak power

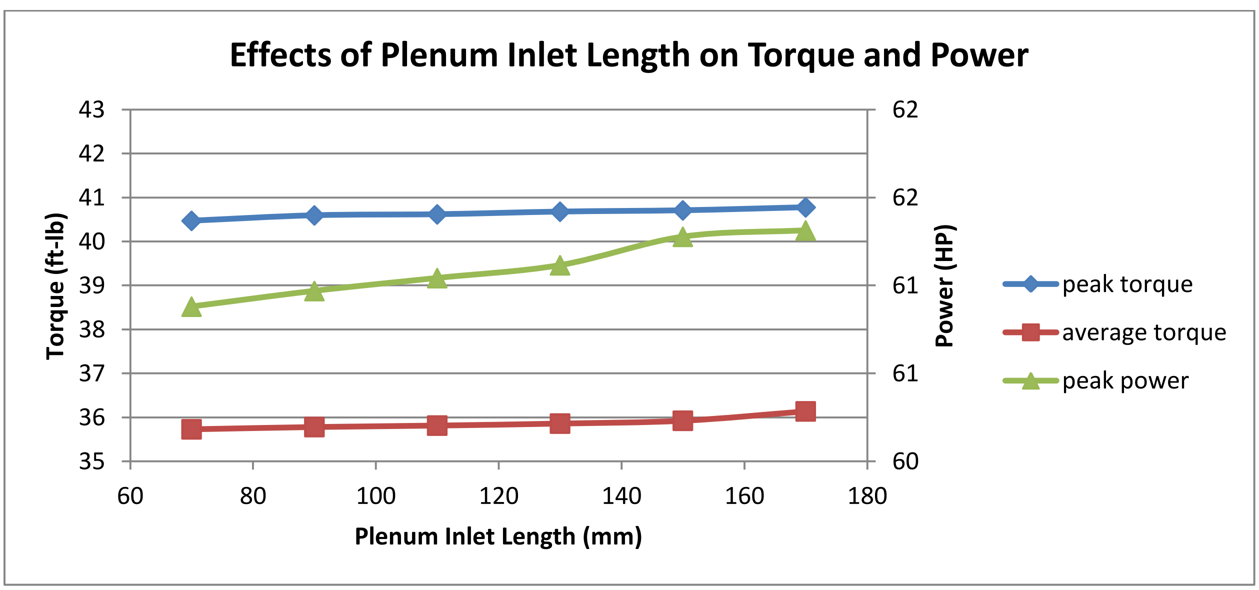

Effects of Comp Inlet Length on Torque
Torque (ft-lb)
42
40
38
36
34
32
30
3500
5500
7500
9500
Engine Speed (RPM)
10 mm
30 mm
50 mm
70 mm
90 mm
110 mm
130 mm

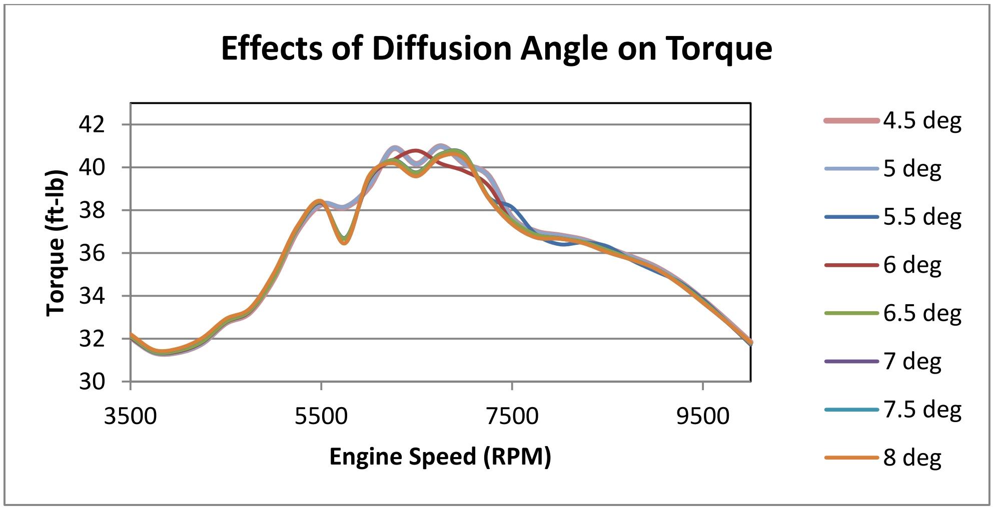

Effects of Comp Inlet Length on Power
Power (HP)
60
55
50
45
40
35
30
25
20
3500
4500
5500
6500
7500
8500
9500
Engine Speed (RPM)
10 mm
30 mm
50 mm
70 mm
90 mm
110 mm
130 mm


Effects of Comp Inlet Length on Torque and Power
Torque (ft-lb)
43
42
41
40
39
38
37
36
35
power (HP)
62
62
61
61
60
60
59
59
58
0
20
40
60
80
100
120
140
Compressor Inlet Length (mm)
peak torque
average torque
peak power

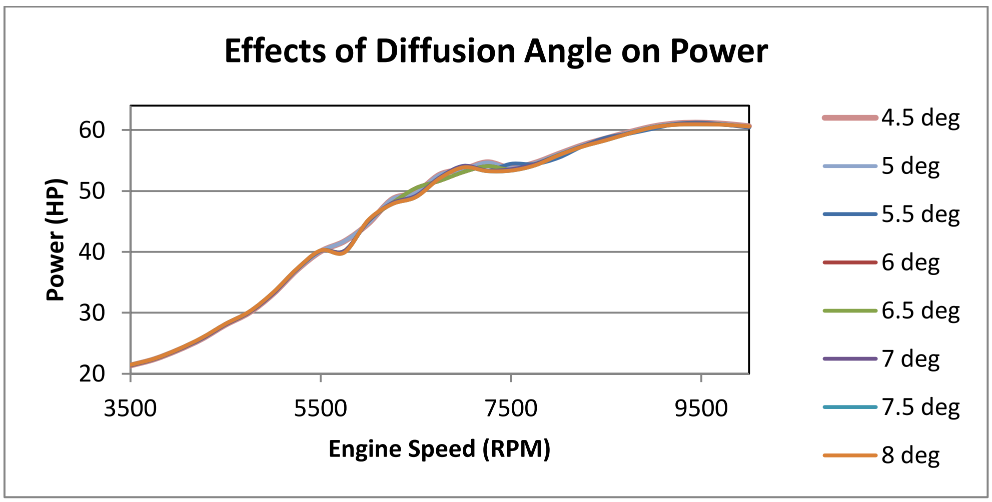

Effects of Plenum Inlet Length on Torque
Torque (ft-lb)
42
40
38
36
34
32
30
3500
4500
5500
6500
7500
8500
9500
Engine Speed (RPM)
70 mm
90 mm
110 mm
130 mm
150 mm
170 mm


Effects of Plenum Inlet Length on Power
Power (HP)
60
55
50
45
40
35
30
25
20
3500
5500
7500
9500
Engine Speed (RPM)
70 mm
90 mm
110 mm
130 mm
150 mm
170 mm

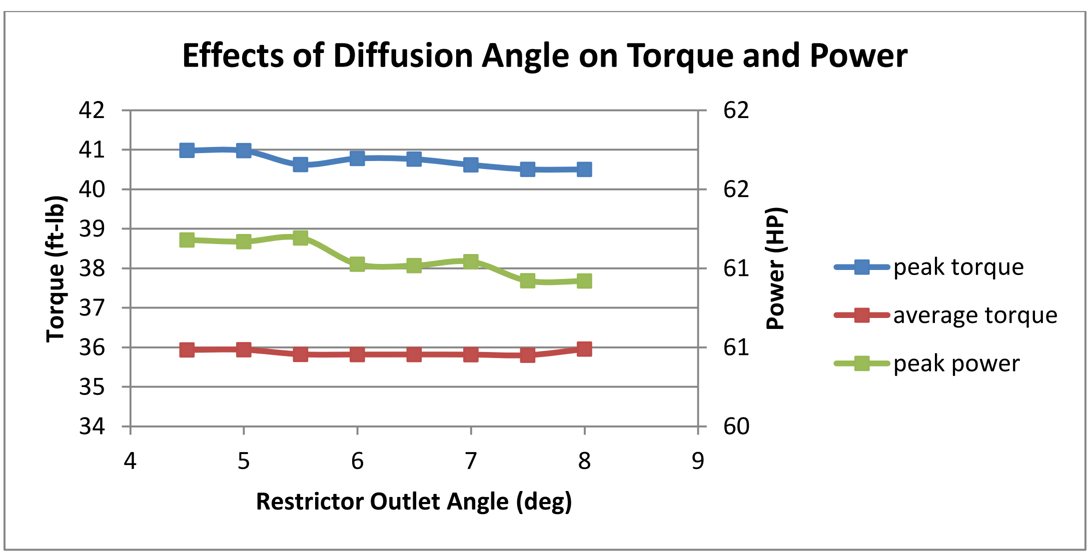

Effects of Plenum Inlet Length on Torque and Power
Torque (ft-lb)
43
42
41
40
39
38
37
36
35
Power (HP)
62
62
61
61
60
60
80
100
120
140
160
180
Plenum Inlet Length (mm)
peak torque
average torque
peak power



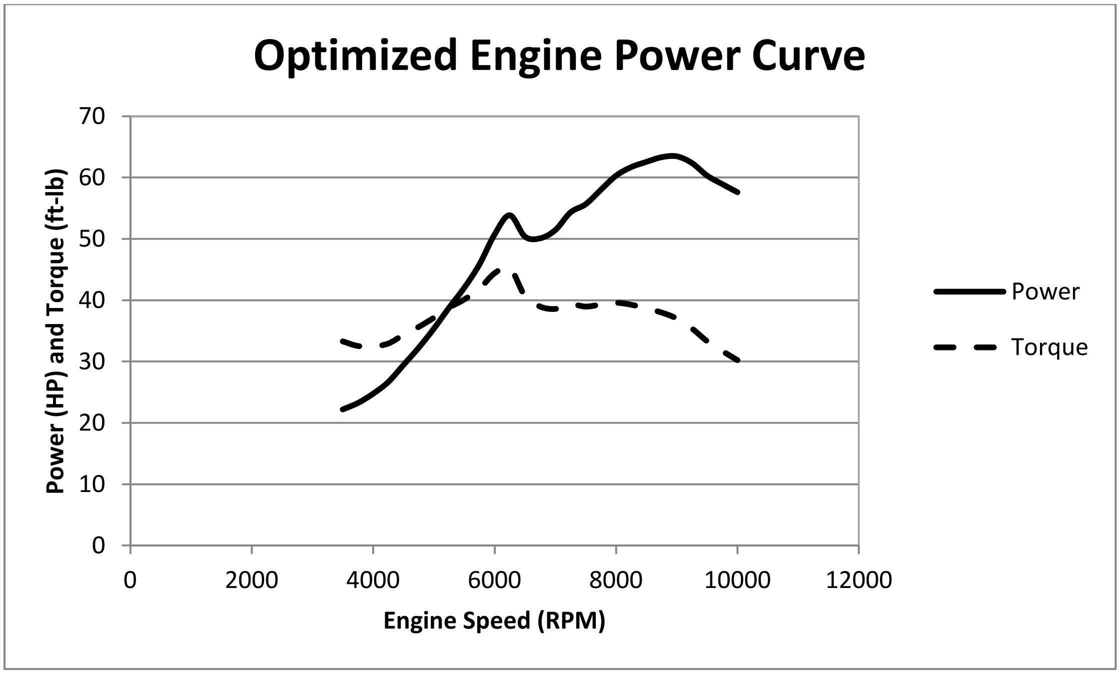
Effects of Diffusion Angle on Torque
Torque (ft-lb)
42
40
38
36
34
32
30
3500
5500
7500
9500
Engine Speed (RPM)
4.5 deg
5 deg
5.5 deg
6 deg
6.5 deg
7 deg
7.5 deg
8 deg



Effects of Diffusion Angle on Power
Power (HP)
60
50
40
30
20
3500
5500
7500
9500
Engine Speed (RPM)
4.5 deg
5 deg
5.5 deg
6 deg
6.5 deg
7 deg
7.5 deg
8 deg



Effects of Diffusion Angle on Torque and Power
Torque (ft-lb)
42
41
40
39
38
37
36
35
34
Power (HP)
62
62
61
61
60
4
5
6
7
8
9
Restrictor Outlet Angle (deg)
peak torque
average torque
peak power



	Naturally Aspirated		Turbocharged	
	Intake	Exhaust	Intake	Exhaust
# of valves	3	2	3	2
Valve diameter	27 mm	28 mm	27 mm	28 mm
Port into cylinder diameter	23 mm	24.5 mm	23 mm	24.5 mm
Port into cylinder length	50mm (outer)	34 mm	50mm (outer)	34 mm
	37mm (center)		37mm (center)	
Port into cylinder angle	80°	80°	80°	80°
Port into head diameter	37.5 mm	36 mm	37.5 mm	36 mm
Port into head length	40 mm	20 mm	40 mm	20 mm
Port into head angle	0°	0°	0°	0°
Max lift	8.71 mm	8.15 mm	8.71 mm	8.15 mm
Bore	95 mm		95 mm	
Stroke	63.5 mm		63.5 mm	
Displacement	449cc		449cc	
Compression Ratio	12.5:1		11:1	
Wrist Pin Offset	0.0 mm		0.0 mm	
Connecting Rod Length	103.5 mm		103.5 mm	
Clearance height	5.51 mm		6.33 mm	

		Run	Run	Run	Run	Run	Run	Run	Run	Run	Run	Run	Run	Run	Run
AFR		7.5	7.5	7.5	7.5	7.5	7.5	7.5	7.5	7.5	7.5	7.5	7.5	7.5	7.5
BMOUTH	mm	170	170	170	170	170	170	170	170	170	170	170	170	170	170
Boost	atm	1.2	1.2	1.2	1.3	1.3	1.3	1.4	1.4	1.4	1.4	1.5	1.5	1.5	1.5
Cin	mm	110	110	110	110	110	110	110	110	110	110	110	110	110	110
CPLEN	mm^3	250	250	250	250	250	250	250	250	250	250	250	250	250	250
DLE	mm	35	35	35	35	35	35	35	35	35	35	35	35	35	35
DLI	mm	25	25	25	25	25	25	25	25	25	25	25	25	25	25
EVAP		0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3
FMEP		0.45	0.4615	0.47307	0.48461	0.49615	0.5076	0.51923	0.53076	0.54230	0.55384	0.56538	0.57692	0.58846	0.6
HLENGTH	mm	200	200	200	200	200	200	200	200	200	200	200	200	200	200
Pin	mm	150	150	150	150	150	150	150	150	150	150	150	150	150	150
PLENVOL	cm^3	1900	1900	1900	1900	1900	1900	1900	1900	1900	1900	1900	1900	1900	1900
Rin	mm	22	22	22	22	22	22	22	22	22	22	22	22	22	22
Rout	mm	65	65	65	65	65	65	65	65	65	65	65	65	65	65
SPANGLE	deg	45	45	45	45	45	45	45	45	45	45	45	45	45	45
SPEED	rpm	3500	3750	4000	4250	4500	4750	5000	5250	5500	5750	6000	6250	6500	6750
SRPM	rpm	100000	102500	105000	107500	110000	112500	115000	117500	120000	122500	125000	127500	130000	132500
Throttle	deg	90	90	90	90	90	90	90	90	90	90	90	90	90	90
Tolerance	bar	0.2	0.2	0.2	0.2	0.2	0.2	0.1	0.1	0.1	0.05	0.05	0.05	0.03	0.03
VGR		0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.65	0.65	0.7	0.7
WBDUR	deg	41	41.5	42	42.3	42.6	42.8	43	43.3	43.7	44	44	44	44	44
WCA50	deg	8.85	8.6	8.4	8.2	8	7.8	7.5	7.3	7.1	7	6.9	6.8	6.65	6.55
WEXP		1.7	1.682	1.67	1.65	1.635	1.61	1.6	1.58	1.568	1.555	1.54	1.52	1.5	1.5

	Run	Run	Run	Run	Run	Run	Run	Run	Run	Run	Run	Run	Run
AFR	7.5	7.5	7.5	7.5	7.5	7.5	7.5	7.5	7.5	7.5	7.5	7.5	7.5
BMOUTH	170	170	170	170	170	170	170	170	170	170	170	170	170
Boost	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5
Cin	110	110	110	110	110	110	110	110	110	110	110	110	110
CPLEN	250	250	250	250	250	250	250	250	250	250	250	250	250
DLE	35	35	35	35	35	35	35	35	35	35	35	35	35
DLI	25	25	25	25	25	25	25	25	25	25	25	25	25
EVAP	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3
FMEP	0.611538	0.623077	0.634615	0.646154	0.657692	0.669231	0.680769	0.692308	0.703846	0.715385	0.726923	0.738462	0.75
HLENGTH	200	200	200	200	200	200	200	200	200	200	200	200	200
Pin	150	150	150	150	150	150	150	150	150	150	150	150	150
PLENVOL	1900	1900	1900	1900	1900	1900	1900	1900	1900	1900	1900	1900	1900
Rin	22	22	22	22	22	22	22	22	22	22	22	22	22
Rout	65	65	65	65	65	65	65	65	65	65	65	65	65
SPANGLE	45	45	45	45	45	45	45	45	45	45	45	45	45
SPEED	7000	7250	7500	7750	8000	8250	8500	8750	9000	9250	9500	9750	10000
SRPM	135000	137500	140000	142500	145000	147500	150000	152500	155000	157500	160000	162500	165000
Throttle	90	90	90	90	90	90	90	90	90	90	90	90	90
Tolerance	0.03	0.03	0.02	0.02	0.02	0.02	0.02	0.02	0.02	0.02	0.02	0.02	0.02
VGR	0.7	0.7	0.7	0.7	0.7	0.7	0.74	0.79	0.83	0.85	0.88	0.9	0.95
WBDUR	44	44	44	44	44	44	44	44	44	44	44	44	44
WCA50	6.5	6.5	6.5	6.5	6.5	6.5	6.5	6.5	6.5	6.5	6.5	6.5	6.5
WEXP	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5

INTAKE CAMSHAFT		
crank angle	Lift (in.)	Lift (mm)
0	0.000	0.000
180	0.000	0.000
320	0.000	0.000
340	0.010	0.254
347	0.020	0.508
351	0.030	0.762
355	0.040	1.016
358	0.050	1.270
362	0.060	1.524
365	0.070	1.778
368	0.080	2.032
370	0.090	2.286
373	0.100	2.540
375	0.110	2.794
377	0.120	3.048
380	0.130	3.302
383	0.140	3.556
385	0.150	3.810
387	0.160	4.064
390	0.170	4.318
393	0.180	4.572
396	0.191	4.851
399	0.200	5.080
402	0.210	5.334
405	0.220	5.588
409	0.230	5.842
411	0.240	6.096
415	0.250	6.350
419	0.260	6.604
423	0.270	6.858
426	0.280	7.112
430	0.290	7.366
435	0.300	7.620
440	0.310	7.874
446	0.320	8.128
455	0.330	8.382
467	0.340	8.636
475	0.343	8.712
494	0.340	8.636
505	0.330	8.382
512	0.320	8.128
519	0.310	7.874
523	0.300	7.620
527	0.290	7.366
530	0.280	7.112
534	0.270	6.858
538	0.252	6.401
544	0.235	5.969
549	0.220	5.588
552	0.210	5.334
555	0.200	5.080
558	0.187	4.750
560	0.180	4.572
564	0.169	4.293
566	0.160	4.064
570	0.150	3.810
573	0.139	3.531
576	0.130	3.302
579	0.120	3.048
582	0.110	2.794
585	0.100	2.540
588	0.082	2.083
593	0.070	1.778
597	0.059	1.499
600	0.049	1.245
602	0.040	1.016
606	0.029	0.737
610	0.020	0.508
630	0.010	0.254
640	0.000	0.000

EXHAUST CAMSHAFT		
crank angle	Lift (in.)	Lift (mm)
0	0.000	0.000
130	0.000	0.000
140	0.010	0.254
145	0.020	0.508
148	0.031	0.787
151	0.040	1.016
154	0.050	1.270
157	0.060	1.524
160	0.070	1.778
162	0.080	2.032
166	0.090	2.286
168	0.100	2.540
172	0.110	2.794
175	0.120	3.048
177	0.130	3.302
179	0.140	3.556
182	0.150	3.810
186	0.160	4.064
189	0.170	4.318
192	0.180	4.572
195	0.190	4.826
199	0.200	5.080
202	0.210	5.334
205	0.220	5.588
209	0.230	5.842
212	0.240	6.096
216	0.250	6.350
219	0.260	6.604
224	0.270	6.858
230	0.280	7.112
235	0.290	7.366
241	0.300	7.620
252	0.310	7.874
258	0.315	8.001
266	0.320	8.128
272	0.321	8.153
278	0.320	8.128
294	0.310	7.874
303	0.300	7.620
310	0.290	7.366
317	0.280	7.112
322	0.270	6.858
326	0.260	6.604
331	0.250	6.350
337	0.240	6.096
342	0.230	5.842
346	0.220	5.588
349	0.210	5.334
352	0.200	5.080
355	0.190	4.826
358	0.180	4.572
362	0.170	4.318
364	0.160	4.064
367	0.150	3.810
369	0.140	3.556
374	0.130	3.302
376	0.119	3.023
378	0.110	2.794
381	0.100	2.540
384	0.090	2.286
387	0.080	2.032
390	0.070	1.778
393	0.060	1.524
396	0.049	1.245
399	0.040	1.016
402	0.030	0.762
406	0.020	0.508
410	0.010	0.254
417	0.000	0.000

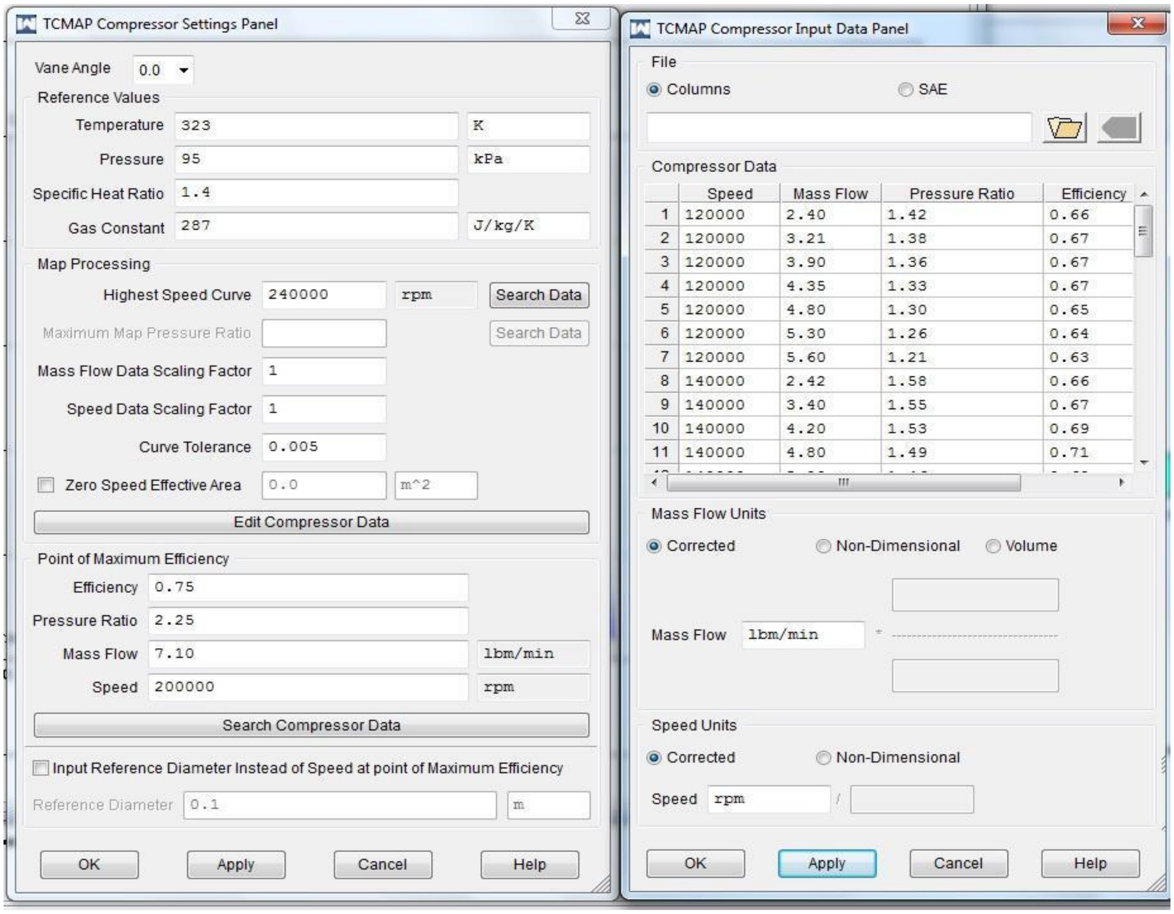
GT15V Turbine Values Input

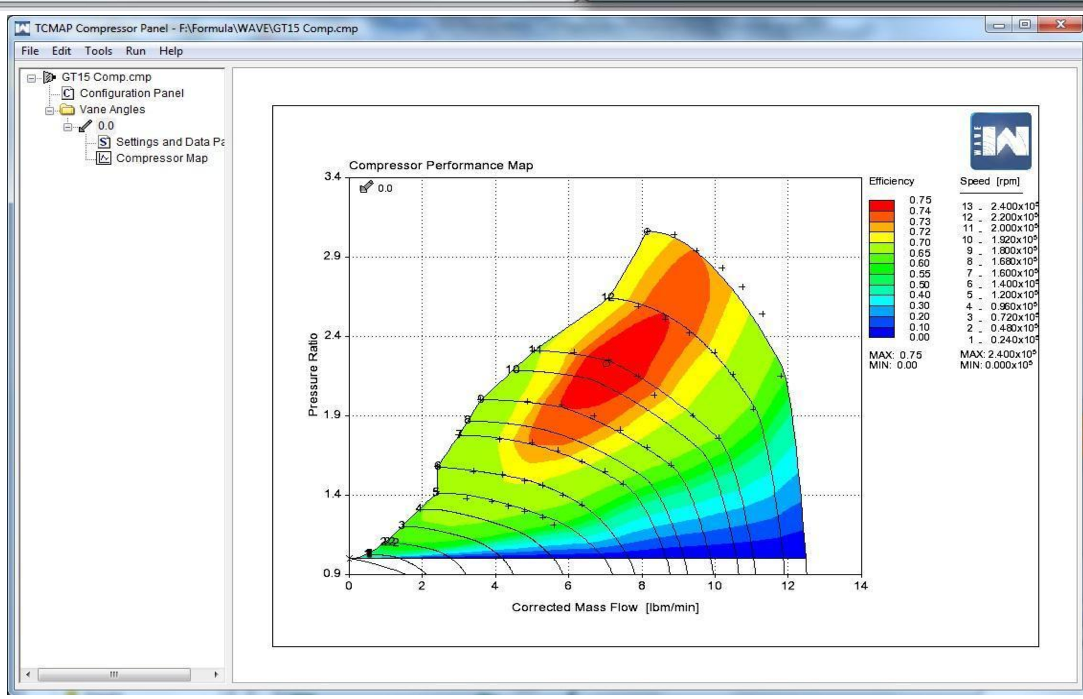
0% Open				
lb/min	speed	kg/s	PR	eff
1.85	55608	0.013986	1.82	0.378
2.07	66845	0.015649	2.16	0.369
2.22	78045	0.016783	2.6	0.375
2.4	89278	0.018144	2.93	0.38
2.6	100428	0.019656	3.45	0.38
10% Open				
lb/min	speed	kg/s	PR	eff
2.4	55608	0.018144	1.53	0.46
2.7	66845	0.020412	1.76	0.465
2.9	78045	0.021924	2.04	0.476
3.1	89278	0.023436	2.37	0.492
3.22	100428	0.024343	2.77	0.51
20% Open				
lb/min	speed	kg/s	PR	eff
2.75	55608	0.02079	1.44	0.514
3.02	66845	0.022831	1.63	0.515
3.25	78045	0.02457	1.86	0.52
3.52	89278	0.026611	2.15	0.535
3.65	100428	0.027594	2.48	0.553
3.8	111460	0.028728	2.91	0.575
3.85	122550	0.029106	3.38	0.581
40% Open				
lb/min	speed	kg/s	PR	eff
3.8	55608	0.028728	1.28	0.635
4.22	66845	0.031903	1.39	0.622
4.64	78045	0.035078	1.55	0.618
5	89278	0.0378	1.75	0.622
5.23	100428	0.039539	1.98	0.624
5.4	111460	0.040824	2.3	0.625
5.5	122550	0.04158	2.7	0.62
60% Open				
lb/min	speed	kg/s	PR	eff
4.6	55608	0.034776	1.23	0.612
5.16	66845	0.03901	1.33	0.586
5.69	78045	0.043016	1.48	0.58
6.14	89278	0.046418	1.66	0.574
6.4	100428	0.048384	1.9	0.565
6.6	111460	0.049896	2.21	0.552
6.6	122550	0.049896	2.62	0.537

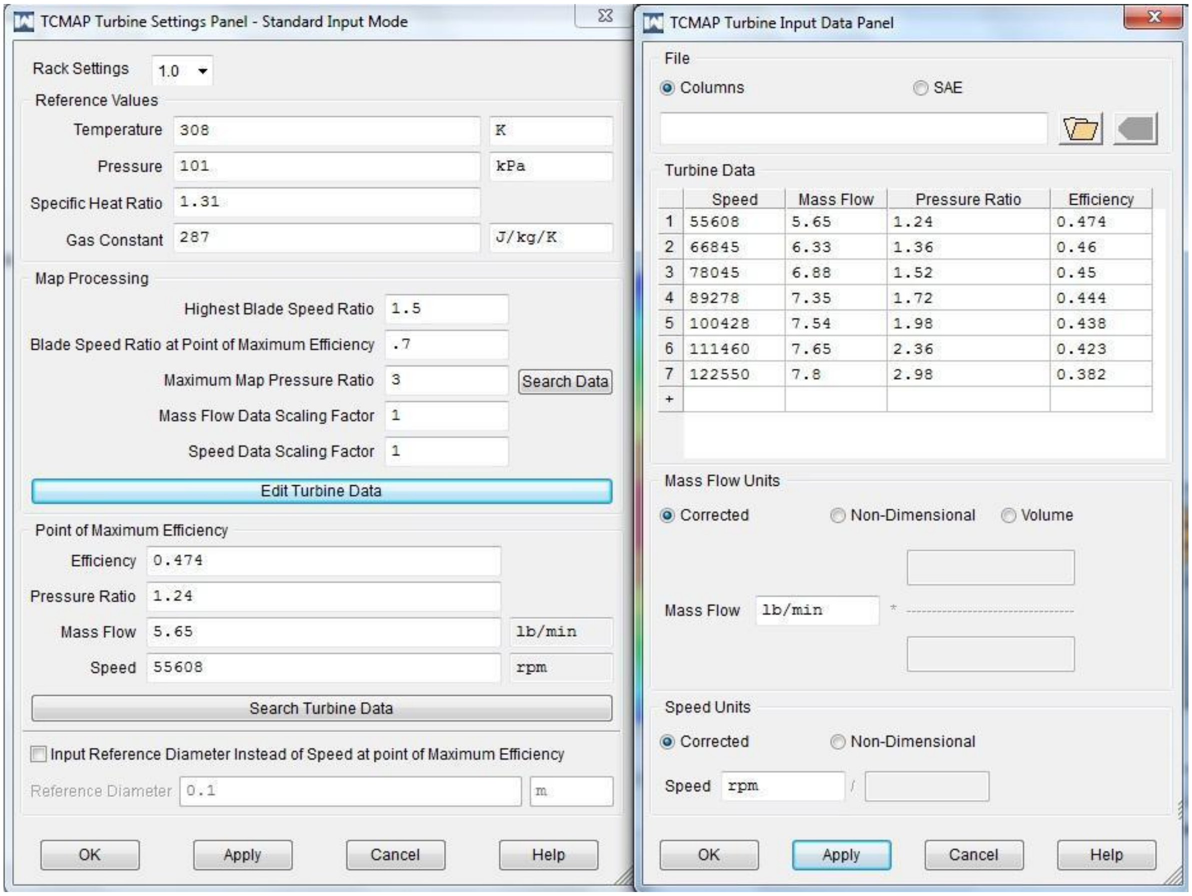
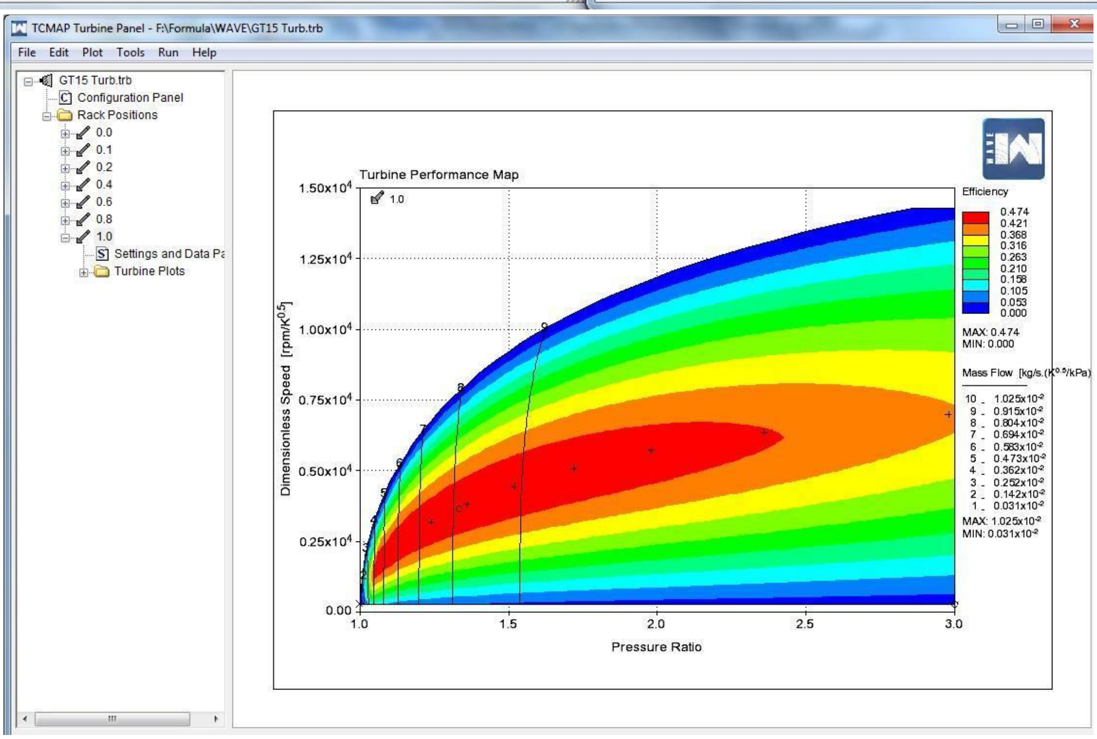
80% Open				
lb/min	speed	kg/s	PR	eff
5.2	55608	0.039312	1.22	0.54
5.84	66845	0.04415	1.34	0.518
6.48	78045	0.048989	1.48	0.51
6.9	89278	0.052164	1.67	0.5
7.2	100428	0.054432	1.92	0.492
7.3	111460	0.055188	2.25	0.482
7.2	122550	0.054432	2.72	0.459
100% Open				
lb/min	speed	kg/s	PR	eff
5.65	55608	0.042714	1.24	0.474
6.33	66845	0.047855	1.36	0.46
6.88	78045	0.052013	1.52	0.45
7.35	89278	0.055566	1.72	0.444
7.55	100428	0.057078	1.98	0.438
7.55	111460	0.057078	2.36	0.423
7.35	122550	0.055566	2.98	0.382

GT15V Compressor Values Input

speed	lb/min	PR	eff	kg/s
120000	2.40	1.42	0.68	0.018
120000	3.21	1.38	0.68	0.024
120000	3.90	1.60	0.68	0.029
120000	4.35	1.33	0.68	0.033
120000	4.80	1.30	0.65	0.036
120000	5.30	1.26	0.65	0.040
120000	5.60	1.21	0.65	0.042
140000	2.42	1.58	0.68	0.018
140000	3.40	1.55	0.68	0.026
140000	4.20	1.53	0.70	0.032
140000	4.80	1.49	0.72	0.036
140000	5.28	1.46	0.70	0.040
140000	5.85	1.40	0.68	0.044
140000	6.37	1.34	0.65	0.048
160000	3.00	1.78	0.68	0.023
160000	4.10	1.75	0.72	0.031
160000	5.00	1.73	0.74	0.038
160000	5.70	1.68	0.74	0.043
160000	6.37	1.61	0.72	0.048
160000	7.00	1.55	0.68	0.053
160000	7.50	1.47	0.65	0.057
180000	3.60	2.00	0.68	0.027
180000	4.88	1.99	0.70	0.037
180000	5.82	1.97	0.74	0.044
180000	6.70	1.90	0.76	0.051
180000	7.42	1.81	0.74	0.056
180000	8.15	1.70	0.70	0.062
180000	8.80	1.59	0.65	0.067
200000	5.05	2.28	0.70	0.038
200000	6.15	2.30	0.74	0.046
200000	7.10	2.25	0.76	0.054
200000	7.90	2.15	0.76	0.060
200000	8.35	2.03	0.74	0.063
200000	9.40	1.90	0.70	0.071
200000	10.10	1.76	0.65	0.076
220000	7.10	2.64	0.74	0.054
220000	7.90	2.59	0.74	0.060
220000	8.65	2.51	0.74	0.065
220000	10.00	2.30	0.72	0.076
220000	10.50	2.16	0.70	0.079
220000	11.05	1.94	0.65	0.084
240000	8.15	3.06	0.72	0.062
240000	8.90	3.04	0.74	0.067
240000	9.50	2.94	0.74	0.072
240000	10.20	2.83	0.74	0.077
240000	10.75	2.71	0.72	0.081
240000	11.30	2.54	0.70	0.085
240000	11.80	2.15	0.65	0.089


TCMAP Compressor Settings Panel
Vane Angle 0.0
Reference Values
Temperature 323 K
Pressure 95 kPa
Specific Heat Ratio 1.4
Gas Constant 287 J/kg/K
Map Processing
Highest Speed Curve 240000 rpm Search Data
Maximum Map Pressure Ratio Search Data
Mass Flow Data Scaling Factor 1
Speed Data Scaling Factor 1
Curve Tolerance 0.005
Zero Speed Effective Area 0.0 m^2
Edit Compressor Data
Point of Maximum Efficiency
Efficiency 0.75
Pressure Ratio 2.25
Mass Flow 7.10 lbm/min
Speed 200000 rpm
Search Compressor Data
Input Reference Diameter Instead of Speed at point of Maximum Efficiency
Reference Diameter 0.1 m
OK Apply Cancel Help
TCMAP Compressor Input Data Panel
File
Columns SAE
Compressor Data
Speed Mass Flow Pressure Ratio Efficiency
1 120000 2.40 1.42 0.66
2 120000 3.21 1.38 0.67
3 120000 3.90 1.36 0.67
4 120000 4.35 1.33 0.67
5 120000 4.80 1.30 0.65
6 120000 5.30 1.26 0.64
7 120000 5.60 1.21 0.63
8 140000 2.42 1.58 0.66
9 140000 3.40 1.55 0.67
10 140000 4.20 1.53 0.69
11 140000 4.80 1.49 0.71
Mass Flow Units
Corrected Non-Dimensional Volume
Mass Flow lbm/min
Speed Units
Corrected Non-Dimensional
Speed rpm
OK Apply Cancel Help


TCMAP Compressor Panel - F:\Formula\WAVE\GT15 Comp.cmp
File Edit Tools Run Help
GT15 Comp.cmp
Configuration Panel
Vane Angles
0.0
Settings and Data Pa
Compressor Map
Compressor Performance Map
Pressure Ratio
Corrected Mass Flow [lbm/min]
Efficiency
MAX: 0.75
MIN: 0.00
Speed [rpm]
MAX: 2.400x10^5
MIN: 0.000x10^5


TCMAP Turbine Settings Panel - Standard Input Mode
Rack Settings 1.0
Reference Values
Temperature 308 K
Pressure 101 kPa
Specific Heat Ratio 1.31
Gas Constant 287 J/kg/K
Map Processing
Highest Blade Speed Ratio 1.5
Blade Speed Ratio at Point of Maximum Efficiency .7
Maximum Map Pressure Ratio 3
Search Data
Mass Flow Data Scaling Factor 1
Speed Data Scaling Factor 1
Edit Turbine Data
Point of Maximum Efficiency
Efficiency 0.474
Pressure Ratio 1.24
Mass Flow 5.65 lb/min
Speed 55608 rpm
Search Turbine Data
Input Reference Diameter Instead of Speed at point of Maximum Efficiency
Reference Diameter 0.1 m
OK
Apply
Cancel
Help
TCMAP Turbine Input Data Panel
File
Columns
SAE
Turbine Data
Speed Mass Flow Pressure Ratio Efficiency
1 55608 5.65 1.24 0.474
2 66845 6.33 1.36 0.46
3 78045 6.88 1.52 0.45
4 89278 7.35 1.72 0.444
5 100428 7.54 1.98 0.438
6 111460 7.65 2.36 0.423
7 122550 7.8 2.98 0.382
Mass Flow Units
Corrected
Non-Dimensional
Volume
Mass Flow lb/min
Speed Units
Corrected
Non-Dimensional
Speed rpm
OK
Apply
Cancel
Help

TCMAP Turbine Panel - F:\Formula\WAVE\GT15 Turb.trb
File Edit Plot Tools Run Help
GT15 Turb.trb
Configuration Panel
Rack Positions
0.0
0.1
0.2
0.4
0.6
0.8
1.0
Settings and Data Pa
Turbine Plots
Turbine Performance Map
Dimensionless Speed [rpm/K^0.5]
Pressure Ratio
1.50x10^4
1.25x10^4
1.00x10^4
0.75x10^4
0.50x10^4
0.25x10^4
0.00
1.0
1.5
2.0
2.5
3.0
Efficiency
0.474
0.421
0.368
0.316
0.263
0.210
0.158
0.105
0.053
0.000
MAX: 0.474
MIN: 0.000
Mass Flow [kg/s.(K^0.5/kPa)]
10 1.025x10^-2
9 0.915x10^-2
8 0.804x10^-2
7 0.694x10^-2
6 0.583x10^-2
5 0.473x10^-2
4 0.362x10^-2
3 0.252x10^-2
2 0.142x10^-2
1 0.031x10^-2
MAX: 1.025x10^-2
MIN: 0.031x10^-2