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FSAE Turbocharger Design and Implementation

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FSAE Turbocharger Design and Implementation

A Major Qualifying Project Report
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by

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Approved:

Professor Eben Cobb, Advisor

1. Turbocharger
2. FSAE
3. Engine System

Abstract

The goal of this project was to produce a turbocharged engine system that maximized the system’s horsepower-to-weight ratio and could be installed on the 2008 FSAE racecar. In order to accomplish this goal, a complete turbocharged engine system was designed and optimized based on limitations such as cost, FSAE rules and chassis size. Most necessary components were purchased or machined. Ultimately, the design was completed but the implementation could only be advanced as far as the budget would permit.

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Introduction

Each year, a group of WPI students designs, assembles and tests a race car to be entered in the Formula Society of Automotive Engineers (FSAE) Competition. That project team must develop every aspect of the car from the chassis to the seatbelts. This project, however, is a supplement to the main FSAE project that focuses entirely on the engine of the FSAE car. In particular, this project involves the development of a turbocharged engine for the FSAE team to use in their car.

The primary goal of the project is to produce a working turbocharged system that meets all FSAE rules and can be implemented on the 2008 WPI FSAE car. This involves fully designing the system, acquiring all the necessary parts and assembling the system. The secondary goal of the project is to maximize the horsepower-to-weight ratio of the engine. This can be accomplished by maximizing the horsepower of the engine throughout the power band while minimizing the overall weight of the system's components.

At the start of this project, the team collectively had no experience working with engines or turbochargers. The MQP report from last year's FSAE turbocharger team (*Turbocharging the FORMULA SAE Race Car* by Jahnke, Locke and Shuman) was provided to serve as a starting point for this year's project. Along with the report, the team was given several components from last year's project, including a turbocharger, an intercooler and boost control assembly. The team was also provided with an undocumented pile of disassembled engine components. A bench was provided in the Washburn Shops to serve as the team's work area.

Given these starting conditions, the following plan was developed to accomplish the stated goals. The first step was to acquire a running naturally aspirated engine and test it for necessary data. The next step was to use this data along with the results of the team's research to completely design a turbocharged system that met the stated objectives. Once the design was complete, all the necessary components were to be purchased or machined. The actual system would then be assembled and tested. Finally, the operational system would be installed on the FSAE car.

1 – Background Information

This section of the paper contains the results of the background research conducted by the team. These findings include general information about internal combustion engines, forced induction and turbochargers. A summary of FSAE rules relevant to turbocharging is also included since the turbocharged system being developed must be designed to comply with all FSAE rules so it could be entered into the FSAE competition.

1.1 – Internal Combustion Engine

The internal combustion engine is the powerhouse of a variety of machines and equipment ranging from small lawn equipment to large aircraft or boats. Given the focus of this paper, the most important machine powered by an internal combustion engine is the automobile. The engine literally provides the driving force of the car while also directly or indirectly powering just about every other mechanical and electrical system in the modern automobile. While there are several types of internal combustion engines that cover the aforementioned large range of applications, they all basically do the same thing. They all convert the chemical energy stored in a fuel of some kind into mechanical energy, which can then be converted into electrical energy. The three most common types of internal combustion are the 4-stroke gasoline engine, the 2-stroke gasoline engine, and the diesel engine. A brief description of each the common types of internal combustion engine is provided below.

The 4-stroke gasoline engine is the most frequently used engine in cars and light trucks as well as in large boats and small aircraft. The major components of the cylinder of a 4-stroke gasoline engine are shown in Fig. 1.1. While the arrangement and number of the cylinders in an engine tends to vary, the parts that make up an individual cylinder remain pretty constant. The most significant component is the piston which is connected to the crankshaft via a connecting rod. The motions of the piston and crankshaft are always related, with one always forcing the other to move. The two valves, intake and exhaust, at the top of the cylinder are opened and closed by separate camshafts that precisely control the timing of each valve's movement. The spark plug at the top of the cylinder is powered by the engine battery and activated by the engine computer at the appropriate time. Finally, the entire cylinder is surrounded by coolant channels that run through the engine block to remove the massive amount of heat generated by the running engine.

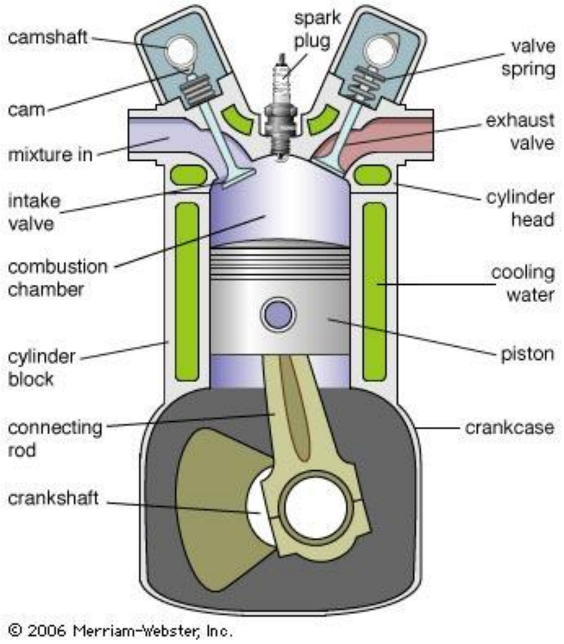

Figure 1.1 Components of a 4-stroke gasoline engine cylinder. (Retrieved from http://cache.eb.com/eb/image?id=72180&rendTypeId=35 on April 14, 2008)

The four strokes of a 4-stroke gasoline engine, illustrated in Fig. 1.2, are intake, compression, power and exhaust. During the intake stroke, the camshaft opens the intake valve as the crankshaft lowers the piston, which allows the cylinder to be filled with a precise mixture of air and gasoline. Once the piston reaches the bottom of the cylinder, the camshaft closes the intake valve. The piston is now at what is known as bottom dead center, and the cylinder is completely filled with the air/fuel mixture.

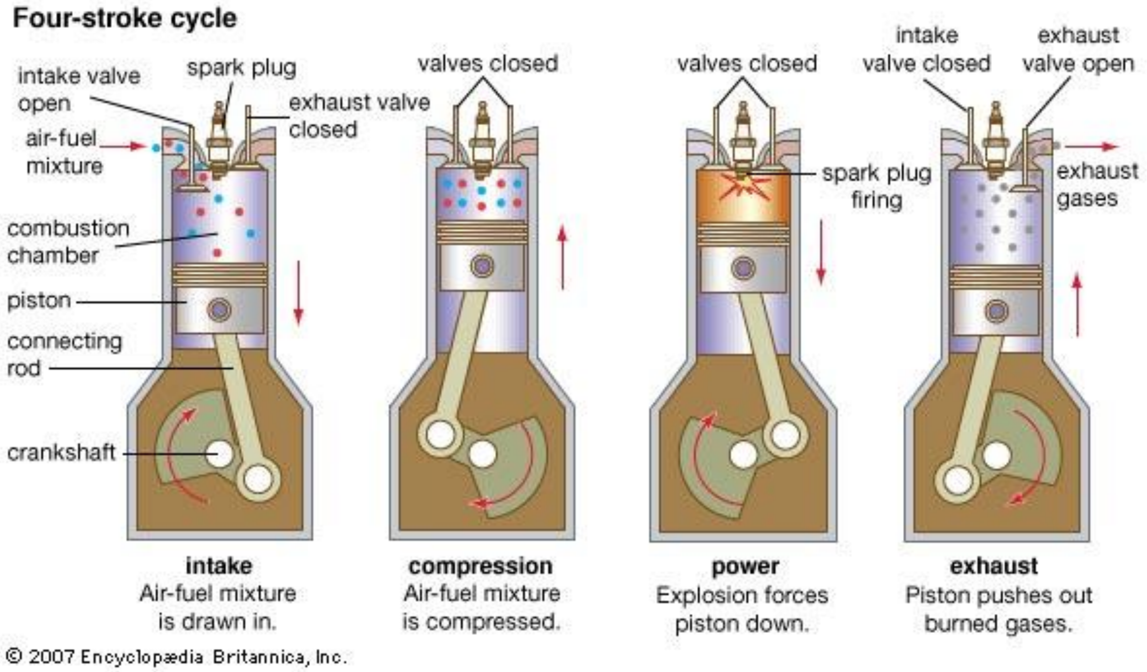

Figure 1.2 Engine cycle of a 4-stroke gasoline engine. (Retrieved from http://cache.eb.com/eb/image?id=93572&rendTypeId=34 on April 14, 2008)

The compression stroke comes next. With both intake and exhaust valves closed, the crankshaft raises the piston, compressing the air/fuel mixture. When the piston has been raised to the top of the cylinder, it is said to be at top dead center. Once the cylinder has reached top dead center, the air/fuel mixture has been compressed as much as possible.

The power stroke is next up. With the piston still at top dead center and both valves closed, the spark plug fires, igniting the compressed air/fuel mixture. Once ignited, a flame begins to move through the mixture, causing it to expand downward smoothly. This expansion downward forces the piston to move down. This means that the piston is rotating to the crankshaft, whereas the rotation of the crankshaft moves the piston in the other three strokes. The fact that the piston is driving the crankshaft means that energy is being transferred to the crankshaft. This is how an internal combustion engine transforms chemical energy in the fuel into mechanical energy. The power stroke is completed once the expanding gases have forced the piston to bottom dead center.

The final stroke is the exhaust stroke. The camshaft opens the exhaust valve as the crankshaft raises the piston, which pushes the exhaust gases out of the cylinder. Once the piston has reached top dead center, all of the exhaust gases have been removed from the cylinder. The cylinder is now ready to start the cycle over again with another intake stroke.

The 2-stroke gasoline engine accomplishes the same thing as the 4-stroke gasoline engine but with half as many strokes. Since they can produce a good amount of power for their relatively small size, 2-stroke gasoline engines are found on a variety of lawn care and recreational equipment like lawnmowers, chainsaws, snowmobiles and small boat engines. They are generally not used for larger application because they are less efficient and dirtier than their 4-stroke counterparts. Aside from having fewer strokes, 2-stroke engines differ from 4-stroke engines in their fuel mixtures and cylinder components. The two strokes, upstroke and downstroke, of a 2-stroke engine along with the cylinder setup can be seen in Fig. 1.3.

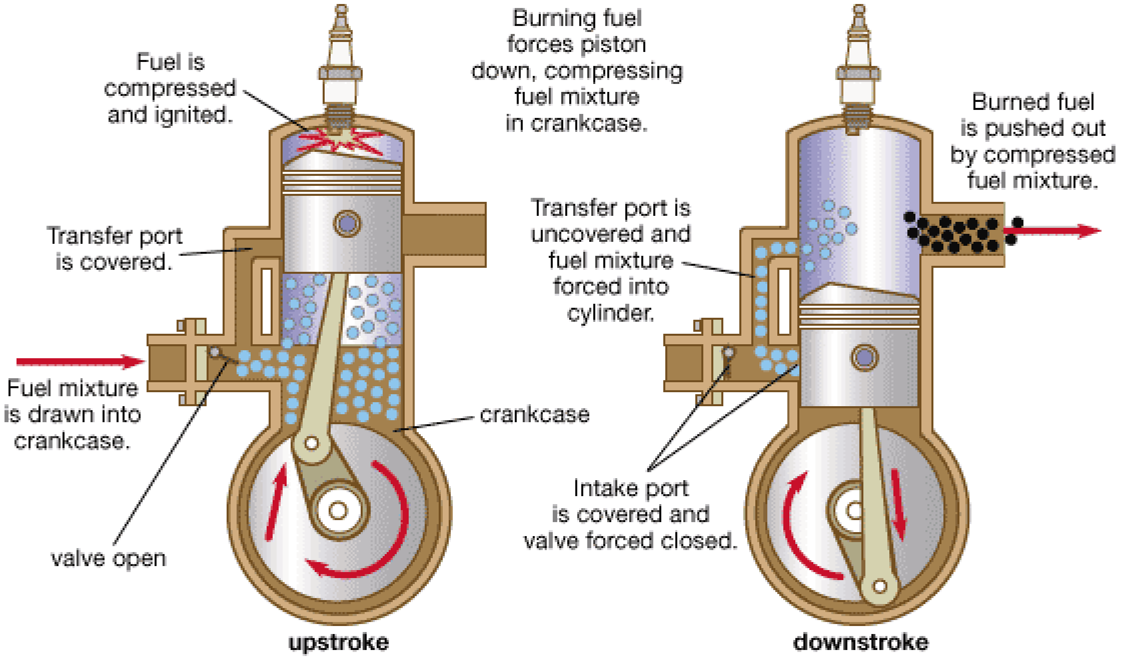

Figure 1.3 Engine cycle of a 2-stroke gasoline engine. (Retrieved from http://cache.eb.com/eb/image?id=310&rendTypeId=4 on April 14, 2008)

There are no camshafts or complicated valve trains involved here, meaning the piston basically has to perform more diverse functions that in 4-stroke engines. Furthermore, special two cycle oil is mixed in with the gasoline to help lubricate the piston, so the air/fuel mixture in a 2-stroke engine includes oil. When the piston is at the bottom of the cylinder, the already compressed air/fuel mixture has moved via the transfer port into the top of the cylinder. On the upstroke, the piston further compresses the air/fuel mixture, creates a vacuum in the crankcase and uncovers the intake port. The vacuum opens the intake valve and draws more air/fuel mixture into the crankcase. The spark plug fires and ignites the mixture, which forces the piston down just like in the 4-stroke cycle. On the downstroke, the piston transfers energy to the crankshaft while compressing the air/fuel mixture in the crankcase and uncovering the exhaust port. As the piston reaches the bottom of the cylinder, the compressed air/fuel mixture is again forced into the top of the cylinder via the transfer port, which forces the remaining exhaust out of the cylinder. The cycle can now begin again.

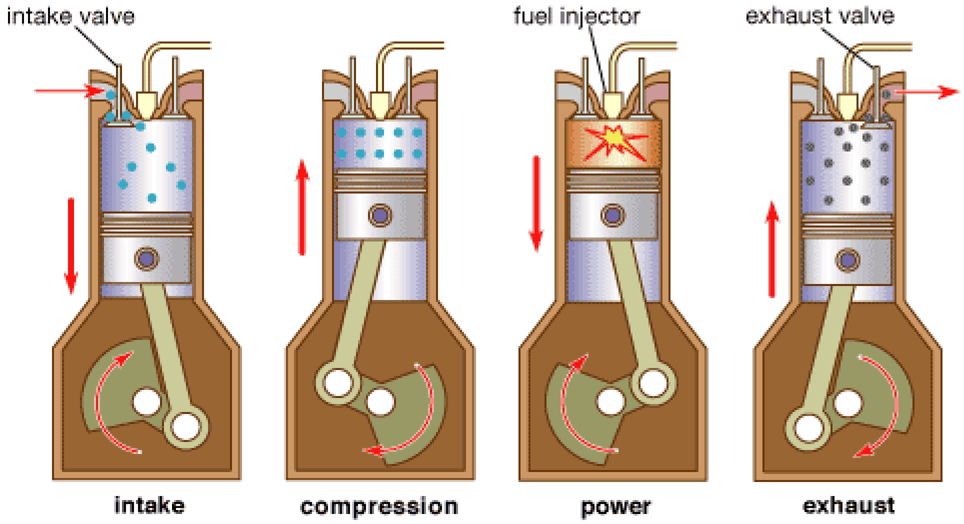

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Figure 1.4 Engine cycle of a 4-stroke diesel engine. (Retrieved from http://cache.eb.com/eb/image?id=24075&rendTypeId=4 on April 14, 2008)

Diesel engines can have a 2-stroke cycle, but most have a 4-stroke cycle, particularly those used by large trucks. The 4-stroke diesel engine shown in Fig. 1.4 will thus be focused on. The 4-stroke diesel cycle is very similar to the 4-stroke gasoline cycle with the big differences, aside from fuel type, being the fuel injection timing and ignition method. During the intake stroke, the intake valve opens as the crankshaft lowers the piston, drawing in pure air. Once the piston reaches bottom dead center, the intake valve closes. The crankshaft then raises the piston, compressing the air. Diesel engines compress the air to much higher compression ratios than their gasoline counterparts do the air/fuel mixture, which means the compressed air reaches scorching temperatures. The compression stroke is complete when the piston reaches top dead center and the air is completely compressed. It is at this moment that the fuel is injected into the compressed air. The extreme temperature of the compressed air immediately ignites the fuel. The expanding gases then force the piston down, transferring energy to the crankshaft. The power stroke is complete when the piston reaches bottom dead center. During the exhaust stroke, the crankshaft raises the piston, forcing the exhaust gases out of the now open exhaust valve. Once the piston reaches top dead center, the cycle is ready to begin again.

1.2 – Regarding Automobile Engines

Since it was known early on that the engine to be turbocharged in this project was going to be a 4-stroke gasoline engine from an automobile, extra attention was paid to this particular engine and its application in cars and motorcycles. Though the 4-stroke gasoline cycle and the components of an individual cylinder have been covered, there are some other general findings that should be noted.

It was previously stated that engines can have different numbers and arrangements of cylinders. Most automobile engines have four, six or eight cylinders arranged vertically in a line (inline), horizontally opposed (flat) or in a V-shape (V) as shown in Fig. 1.5. The total engine displacement is the total volume, measured in liters, cubic centimeters or cubic inches, of all the cylinders combined. Generally speaking, higher engine displacements lead to higher engine power output, measure in horsepower. The volume of an individual cylinder is determined by its bore (cylinder diameter) and stroke (distance the piston moves from bottom dead center to top dead center). There are thus three ways to increase engine displacement: increase cylinder bore, increase cylinder stroke, or add more cylinders.

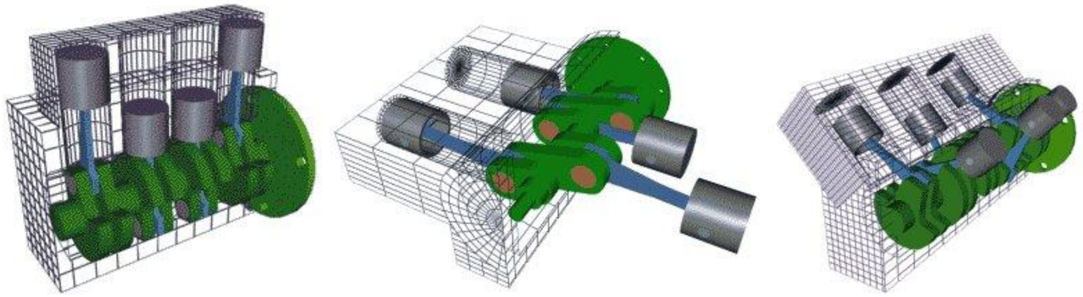

Figure 1.5 Inline, flat and V cylinder arrangements. (Retrieved from http://auto.howstuffworks.com/question366.htm on April 14, 2008)

While the events in the cylinder have been discussed, the means by which the air/fuel mixture arrives at the intake valve and where the exhaust goes after being forced out the valve have not been talked about. Although engines tend to vary from vehicle to vehicle, the following is generally true. Ambient air enters the engine through an air filter that keeps nasty stuff like dirt and bugs out of the engine. The air then enters the plenum which is a large thin walled chamber that gathers and distributes the air to the runners. The runners are simply pipes that connect the plenum to the intake ports. At the base of the runners, just prior to the intake ports, the throttle bodies and fuel injectors are located. The throttle bodies are valves that control the amount of air that enters the engine. Each engine has a specific air/fuel ratio at which is it designed to operate. The engine computer calculates how much air the throttle bodies let into the engine and then activates the fuel injectors to spray just the right amount of fuel into the air. The air/fuel mixture then enters the cylinder during the intake stroke of the engine cycle. After being expelled during the exhaust stroke, the exhaust gases enter pipes known as primaries. The primaries, one per cylinder, are then merged into a single pipe before entering the catalytic converter, which cleans the exhaust of some its worse pollutants. The exhaust then goes through the muffler, which silences that noise of the thousands of explosions taking place in the cylinders each minute, before leaving the tailpipe.

1.3 – Forced Induction

In automotive applications, forced induction quite literally means to force air into the engine. Under standard atmospheric conditions, the engine will naturally consume a volume of air equal to its engine displacement each time it completes its 4-stroke cycle and is said to be naturally aspirated. When some form of forced induction is added, the engine will be forced to consume a volume of air greater than its engine displacement each time it completes its 4-stroke cycle. While this may seem trivial, it is very significant and can result in large power gains for the engine.

The method by which a 4-stroke gasoline engine converts chemical energy into mechanical energy is discussed in Section 1.1. It should be stated plainly that the chemical energy is found entirely within the fuel, and thus the power generated by the engine is directly related to how much fuel is in the cylinders during the power stroke. However, simply flooding the cylinder with gasoline will not result in more power but will manage to seriously damage the engine. Section 1.2 mentions that the engine is designed to operate at a specific air/fuel ratio (AFR). What this means is that for every particle of fuel there needs to be a corresponding number of air molecules. If this ratio is disturbed, the engine will run lean (too little fuel) or rich (too much fuel), either one of which is bad for engine. The engine control unit (ECU) monitors the airflow into the engine and adjusts fuel injection accordingly to maintain a proper AFR. Thus the only way to really get more fuel into the cylinders, and enjoy the added power, is to increase the airflow into the engine, hence the importance of forced induction.

Forced induction systems make use of a compressor to force more air into the cylinders of the engine. In order to maintain a proper AFR, the ECU tells the fuel injectors to spray more fuel into the cylinders, resulting in more power. The compressor is able to force more air into the cylinders by increasing the pressure of the ambient air before it enters the intake ports. With a constant cylinder volume, a lot more air can fit into the cylinder at 20psi than at atmospheric pressure, 14.7psi. An unavoidable thermodynamic result of increasing the air's pressure is to also increase its temperature. The compressor thus raises both the pressure and the temperature of the air.

The two most common types of forced induction are turbocharging and supercharging. Turbochargers and superchargers both use compressors to raise the pressure of the intake air as described above. These devices differ in the way by which they power the compressor. A turbocharger uses exhaust gases expelled from the cylinders to spin a turbine, which in turn powers the compressor. Figure 1.6 shows a cutaway view of a typical turbocharger. The black housing with red highlights on the left is the turbine. The gray housing with blue highlights on the right is the compressor. The yellow and green section in the middle contains the connecting shaft between the turbine and compressor with its bearings.

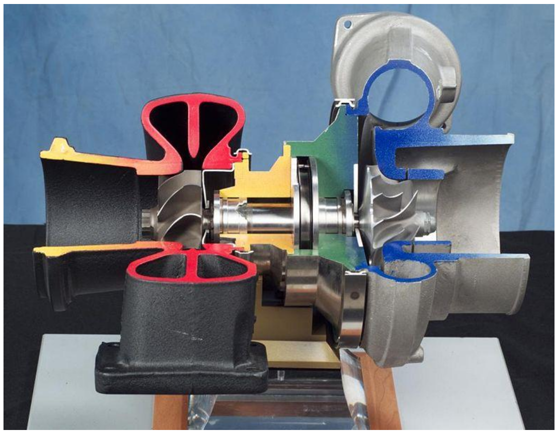

Figure 1.6 Cutaway view of a turbocharger. (Retrieved from http://commons.wikimedia.org/wiki/Image:Turbocharger.jpg on April 14, 2008)

A supercharger uses a belt or chain connection to the engine's crankshaft to power the compressor. Figure 1.7 shows a cutaway view of a typical supercharger. The belt is attached to the pulley on the left directly from the engine's crankshaft. As the crankshaft rotates, it causes the pulley to rotate, which in turn causes the internal gears in the supercharger to rotate. These gears power the compressor.

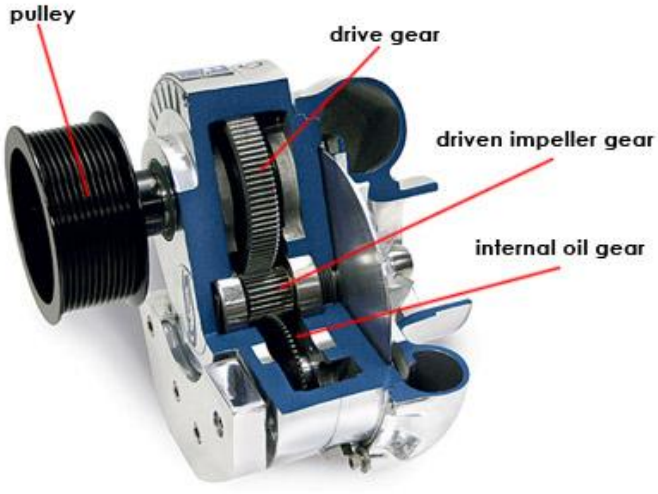

Figure 1.7 Cutaway view of a supercharger. (Retrieved from http://auto.howstuffworks.com/supercharger1.htm on April 14, 2008)

On the right side of Fig. 1.7, there is a centrifugal compressor like that found on a turbocharger. Whereas this is only type of compressor that is found on turbochargers, there are actually a couple other types of compressors that can be found on superchargers, shown in Fig. 1.8. Each of these compressor variants does the same thing. It draws in ambient air at atmospheric pressure, compresses said air and directs the high pressure air towards the engine. The differences in their detailed operations are not relevant to this project.

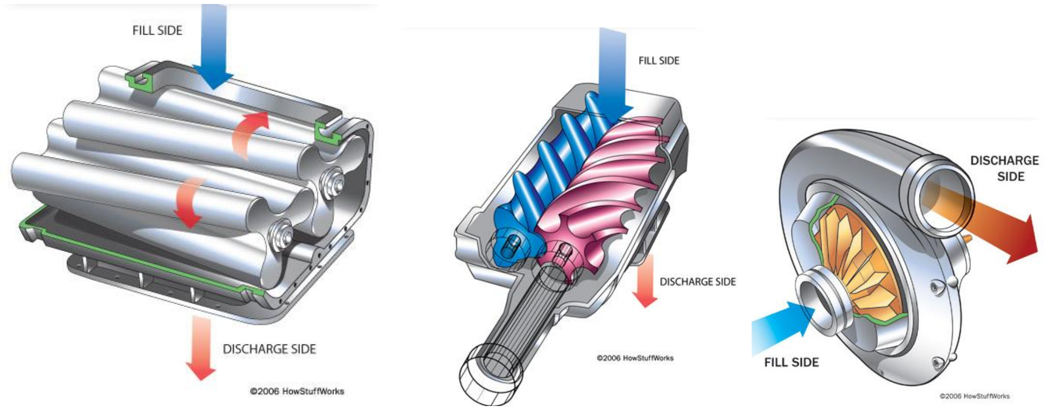

Figure 1.8 Types of supercharger compressors: Roots, twin-screw and centrifugal. (Retrieved from http://auto.howstuffworks.com/supercharger2.htm, http://auto.howstuffworks.com/supercharger3.htm, and http://auto.howstuffworks.com/supercharger4.htm on April 14, 2008)

Since the goal of this project is to produce a turbocharged system, superchargers will not be discussed again in this paper.

1.4 – Turbochargers

As stated in the previous section, a turbocharger is a device that uses engine exhaust gases to power a compressor that increases the pressure of the air entering the engine, which results in more power from the engine. Figure 1.9 is a simple illustration of how a turbocharger works with a single cylinder. Air enters the compressor from the left, is compressed and then directed to the intake valve of the cylinder. Exhaust exits the exhaust valve of the cylinder, spins the turbine and is expelled. The three major pieces of a turbocharger introduced in the previous section and shown in Fig. 1.6 are the compressor, bearings section and turbine. Each of these sections has an important function and deserves further attention. It is also important to recognize in any discussion of turbocharging that turbocharging an engine involves more than just slapping a turbocharger on to the engine. An entire system must be developed for the turbocharger, including a means of temperature and pressure control.

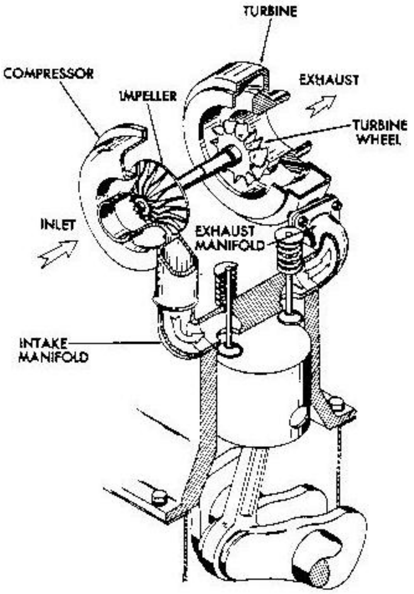

Figure 1.9 Simplified drawing of turbocharger and engine cylinder. (Retrieved from Hugh MacInnes' *Turbochargers,* courtesy of Schwitzer)

1.4.1 – Turbocharger as a Device

Before getting into the details of a turbocharged system, the turbocharger as a device will be described in more detail. Figure 1.10 shows a drawing of what a turbocharger would look like if it were cut right down the middle. It effectively illustrates the relationship between the three sections as well as the input and output of each section. The heart of the turbocharger is the assembly of compressor blades, shaft and turbine blades. It is this assembly that rotates at over a 100000 RPM when the turbocharger is operational. This assembly also serves as the common connection between the three major components of the turbocharger, which otherwise are independent of each other.

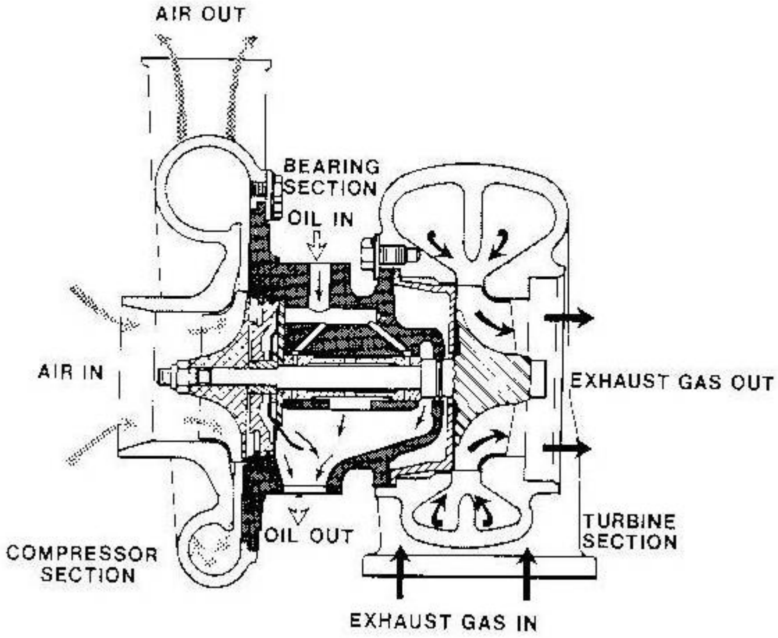

Figure 1.10 Sectioned drawing of turbocharger components. (Retrieved from Hugh MacInnes' *Turbochargers*, courtesy of Schwitzer)

The compressor blades, shaft and turbine blades that make up the rotating assembly are distinctly inside of the compressor housing, bearings housing and turbine housing, respectively. The role of the compressor housing is to direct ambient air axially into the spinning compressor blades. The blades and housing are designed together such that the ambient air is compressed and forced into the air channel wrapped around the center of the compressor housing, which expels the compressed air tangentially. The compressed air leaves the compressor with higher pressure and temperature. The pressure increase across the compressor is known as boost. For example, if ambient air enters the compressor at atmospheric pressure (14.7psi) and leaves at a pressure of 19.7psi, the turbocharger is said to be creating 5psi of boost. Generally speaking, the higher the boost pressure, the higher the power gains but more difficulty and cost are involved in developing the system. The compressor works best at a particular combination of airflow and boost pressure. The compressor should be chosen wisely to ensure most efficient operation.

The primary function of the bearings housing is to guide the rotating shaft connecting the compressor and turbine blades. This shaft is guided using either journal or ball bearings. The bearings housing has a secondary function of lubricating the shaft and bearings. This is accomplished by routing engine oil into the bearings housing, which distributes the oil around the shaft and bearings. The oil is then drained out of the bearings section at which point it can be returned to the engine. Heat from the exhaust gases can lead to oil coking, the charring of the oil on to the oil channels. This restricts oil flow and eventually destroys the bearings. Some bearings sections have water jackets that allow engine coolant to reduce the temperature of the oil.

The main function of the turbine housing is to direct exhaust gases to the turbine blades to accelerate them as quickly as possible. Exhaust gas enters the turbine housing tangentially and travels through the channels surrounding the center of the turbine. These channels lead the air into the turbine blades, forcing them to rotate. The exhaust is then expelled from the turbine housing axially. The size of the turbine housing has a significant impact on the behavior of the turbine. In particular, changing the size of the turbine effects turbocharger response, power gains and the engine speed at which the turbocharger is most effective.

The desirable features of each of these turbocharger components will be discussed in section *4 – Turbocharger Selection*.

1.4.2 – Turbocharger as a System

Now that the basics of a turbocharger have been covered, the major components of a turbocharged system can be discussed. Unfortunately, a turbocharged system is a lot more complicated than the illustration in Fig. 1.9 might suggest. Plumbing, engine modifications, intake air temperature and boost control are all major concerns that need to be addressed. Figure 1.11 illustrates more completely what a basic turbocharged system for an inline four cylinder engine would look like. It includes all the plumbing for the intake and exhaust, oil feed and drain lines, an intercooler for temperature control and a wastegate for boost control. These components will be discussed briefly here and then at length in subsequent sections of the paper.

The intake system consists of everything from the air filter to the intake ports on the engine. This includes the compressor, intercooler (see next paragraph), manifold and throttle bodies. The job of the intake system is to connect all of these components with hoses or pipes. The design of the manifold, which consists of the plenum and runners, and the throttle bodies are also considered part of the intake system. The intake system in general and specifically for this project will be discussed at length in section *6 – Intake System.*

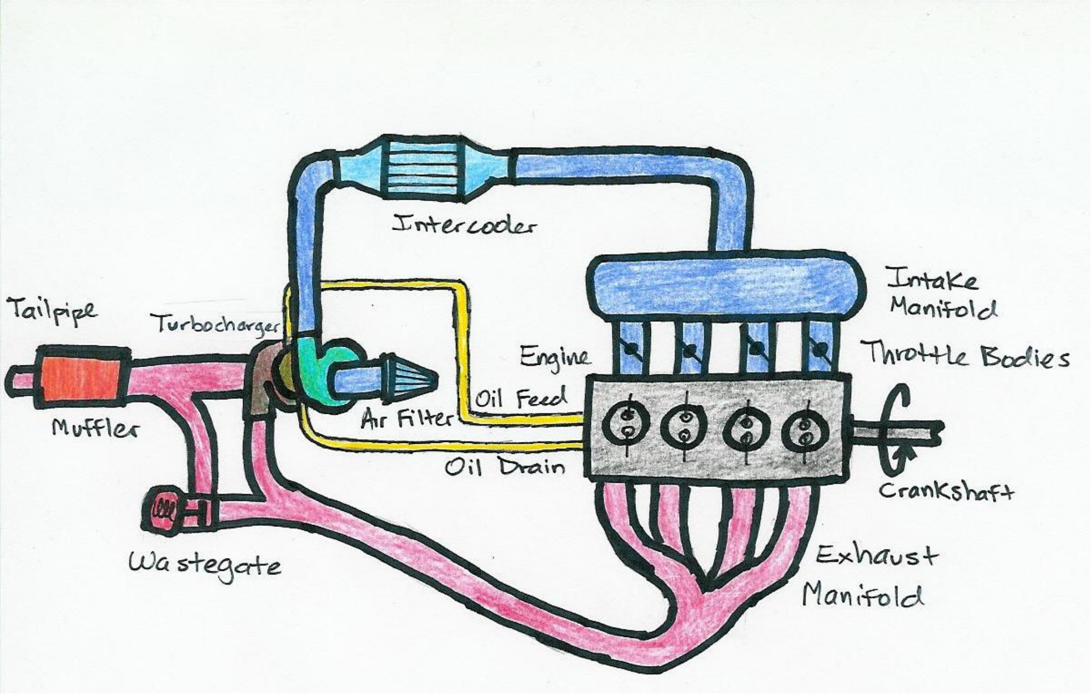

Figure 1.11 Drawing of basic turbocharged system. (Drawn by D. Curran)

The intercooler is a heat exchanger that is included to remove the unwanted heat added to the intake air by the compressor. It is impossible to prevent the compressor from adding heat to the air as it compresses it, though the amount of heat added can be limited by choosing a properly sized compressor. It is undesirable to just allow the hot intake air to go straight to the engine as it can reduce power gains and lead to engine knocking. An intercooler is thus included in the system to remove the heat added by the compressor. The heat is removed via cross flow of a cooling fluid, either air or water. The air is then free to flow to the engine with a lower temperature but still higher than atmospheric pressure. The intercooler in general and specifically for this project will be discussed at length in section *7 – Intercooler*.

The wastegate shown in Fig. 1.11 is the most common form of boost control on modern turbocharged systems. The concept of boost was introduced in the previous section as was the relationship between boost and the system. That is to say that while higher boost generally leads to higher power it also leads to increasingly complicated and expensive system requirements. Since a turbocharged system is rarely designed with unlimited budget and design freedom, there will always be a maximum boost that the system is designed to accommodate. This maximum boost is usually chosen based on performance goals, and the system is then designed specifically for that boost pressure. If this maximum boost pressure is exceeded, the system could very likely fail, resulting in damage to the turbocharger or engine. If left unchecked though, the turbocharger will continue to create boost well past the maximum boost pressure. A boost control system is thus added to limit the boost created by the turbocharger. A wastegate works by bleeding exhaust gas away from the turbine once the maximum boost pressure is reached. As less exhaust reaches the turbine, the turbocharger slows down and creates less boost pressure. Boost control and wastegates in general and specifically for this project will be discussed at length in section *8 – Boost Control*.

The exhaust system consists of everything from the engine exhaust ports to the tailpipe. This includes the manifold, turbine, wastegate and the muffler. The job of the exhaust system is to connect and support all of these components with pipes. The design of the exhaust manifold, including the primaries and merge collectors, is considered part of the exhaust system. The exhaust system in general and specifically for this project will be discussed at length in section *9 – Exhaust System*.

The final major system component shown in Fig. 1.11 is the lubrication system for the turbocharger bearings. In the drawing, this system consists of an oil feed line and an oil drain line between the turbocharger bearings and the engine. The oil feed line is connected to the engine at a location with positive oil pressure, and the oil drain line is connected to the engine's oil pan. More complicated systems including dedicated pumps and oil reserves are not uncommon. The coolant lines between the turbocharger bearings and the engine's radiator circuit are not included in the figure as water jackets are not available on all turbochargers and thus are not considered to be standard. Turbocharger lubrication in general and specifically for this project will be discussed at length in section *10 – Turbocharger Bearings*.

In addition to these components, any high boost turbocharged system is going to require modifications to various engine parts. Electrical and ignition systems may need to be upgraded to ensure proper ignition. The fuel injection systems may need to be upgraded to maintain the correct AFR. The throttle bodies and valve train may need to be changed to provide for proper flow conditions. A more detailed description of necessary engine modifications will be provided in section *12 – Engine Modifications*.

1.5 – FSAE Rules

Since one of the goals of the project is to design a system that meets all FSAE rules, it was important to review all rules for the 2008 FSAE competition. Appendix A contains a list of selected rules quoted directly from the actual document, which can be found at the FSAE website (http://students.sae.org/competitions/formulaseries/rules/). This section will provide a brief summary of the most notable rules affecting the design of the turbocharged system.

The rule that has the biggest impact is the requirement that a 20mm (0.7874 inch) restrictor be placed on the intake system between the throttle body and compressor. All air into the engine must pass through the restrictor. The 20mm diameter of the restrictor is intended to severely limit airflow into the engine and thus engine power. The order of intake components is mandated as throttle body, restrictor, compressor and engine. An intercooler is permitted between the compressor and engine but can only be cooled with ambient air or engine coolant, which must be either plain water or water with small amounts of rust or corrosion preventive additives.

The remaining significant rules follow. The engine must be a 4-stroke engine with displacement of 610cc or less. The fuel used must either be 93 octane gasoline or E85 and must not have its temperature altered or contain any fuel additives. A muffler must be included to limit noise. The exhaust outlet cannot extend more than 60cm (normal distance) beyond the rear axle and cannot be raised more than 60cm off the ground. There must be at least 25.4mm of ground clearance for all engine components. Finally, all engine components must fit in a roll envelope defined by the roll hoop and the rear tires as shown in Fig. 1.12.

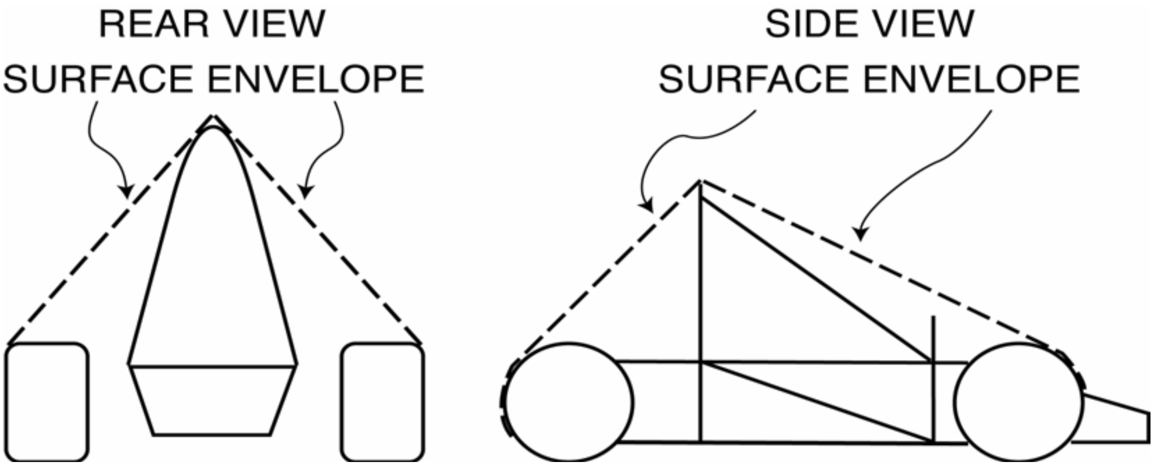

Figure 1.12 Roll envelope defined by FSAE rules. (Retrieved from http://students.sae.org/competitions/formulaseries/rules/rules.pdf on April 16, 2008)

2 – Naturally Aspirated Engine

The first step in the plan included in the *Introduction* section was to acquire an operational naturally aspirated engine. The team had to use an engine from a Honda CBR600 F4i motorcycle, the same engine that the 2008 WPI FSAE team was using. The same engine had to be used because one of the project goals was that the system be designed to be implemented on the 2008 WPI FSAE racecar. This section of the paper describes the team's efforts to acquire a Honda CBR600 F4i motorcycle engine and get it running.

2.1 – Original Engine

At the onset of the project, the team was anticipating that a complete and fully functional engine would be provided and available at WPI. Eventually the team was provided something very different. Several boxes of unlabeled and unorganized engine components were provided. The possibility of reassembling these components and essentially rebuilding the engine needed to be looked into. The team's lack of experience working with engines and lack of knowledge about the detailed inner workings of an engine proved an obstacle in this effort.

The collection of engine components was inventoried and analyzed to determine the condition of each component. There was a stripped down engine block that the team was told had a damaged journal bearing. The cylinder head was partially disassembled but did appear to have all camshafts and valves. A few other miscellaneous components that had been stripped off the block including a damaged clutch basket, an alternator cover, some gears and shafts were mixed in with dozens of unlabeled nuts and bolts.

There were no pistons in the block, but there were several removed pistons lying around, both with and without connecting rods. The spark plugs appeared to be missing. The entire fuel system was missing including the injectors, rail, pump and lines. There was neither radiator nor coolant lines. All the electrical components were missing from the harness, to the sensors, to the ECU. The throttle bodies and air box were also missing.

It took the team a significant amount of time just to sort through the mess and identify each part and assess its condition. After completing this process, it was concluded that rebuilding the engine from the components in hand would be next to impossible. Given the knowledge and experience levels in this area, it would have taken the full efforts of the entire team over months to retrieve all the missing parts and assemble the engine.

The possibility of rebuilding the engine was subsequently taken off the table. A new engine had to be obtained.

2.1 – New Engine

Once the decision to acquire a new engine was made, the team began the process of finding one. This process included extensive online searches, phone calls to motorcycle dealers and trips to junkyards. The objective was a recent model year Honda CBR600 F4i engine. Since money was not unlimited, a new bike could not be purchased just to be stripped down to its engine. The focus of the search had to be on finding a wrecked bike with an intact engine or an engine removed from the bike. The overall cost of the engine or wrecked bike had to be under $1000.

After completing this exhaustive search, a 2001 Honda CBR600 F4i was located on eBay (www.ebay.com). The engine was being sold by a vendor called Cycle Pros Salvage in Bridgewater, Massachusetts for $699.99. The engine was in excellent condition, so the team decided to purchase it. Table 2.1 below lists the engine specifications.

Make	Honda
Model	CBR600 F4i
Year	2001
Miles	8600
Displacement	599cc
Cylinders	4
Arrangement	Inline
Bore	67mm
Stroke	42.5mm
Compression Ratio	12.0:1
Redline	14500 RPM
Fuel	Electronic Fuel Injection
Cooling	Liquid Cooled
Valves	Dual Overhead Cams, 4 Ports/Cylinder
Block Material	Aluminum

Table 2.1 Engine specifications.

With the new engine in hand, the next task was to manufacture a stand on which the new engine could be placed.

2.3 – Engine Stand

The primary purpose of the engine stand was simply to support the engine so it would not just be lying on the ground. Since the team did not possess its own chassis, the engine also needed to actually be able to run on this stand. Ideally, the engine would be able to run on the stand both naturally aspirated and turbocharged. With this in mind, the team began the process of designing the engine stand.

At this point in the year, the 2007 WPI FSAE chassis was still in the shop and had its engine in it. Obviously, the naturally aspirated engine and its subsystems could fit and run in the chassis. Furthermore, the 2007 chassis was fairly similar to the anticipated 2008 chassis, in which the turbocharged system would need to fit. Since a new chassis was being designed for 2008, the team's first thought was to cut up the 2007 chassis to use as an engine stand once the FSAE team removed its engine. Unfortunately, this was also the FSAE team's plan, so this plan was scratched.

The next idea was to build a new engine stand based on the design of the 2007 FSAE chassis. Numerous measurements were taken of the chassis while it had the fully assembled engine in it. These measurements were then used to create a solid model of the upper half of the chassis in SolidWorks. Some modifications were made to the lower half of the model, so the engine would be a little higher off the ground, making it easier to work on. Finally, some rotating industrial wheels were acquired from the shop to be affixed to the bottom of the engine stand, making it mobile. The completed model of the engine stand is shown in Fig. 2.1.

Figure 2.1 Engine stand concept. (Modeled by D. Curran)

With the engine stand model complete, it was time to actually manufacture the stand. A technical drawing was made for the engine stand. The team took the drawing and engine to Merchants Sheet Metal in Auburn, Massachusetts. David Lamoureux, Sr. and Ronnie Withers were very helpful. After taking some measurements of the engine, Ronnie expressed some concerns over how well the dimensions on the drawings matched up with the actual engine. Figure 2.2 shows a model of the engine created by the FSAE team and inserted into the stand model well after the fact. It can clearly be seen that the mounting points on the engine stand do not match up with the bolt holes on the engine. An additional concern was that the fully assembled engine was too wide to be slid into the stand. Thus the engine would need to be disassembled before being bolted into the engine stand, and then the engine would need to be reassembled.

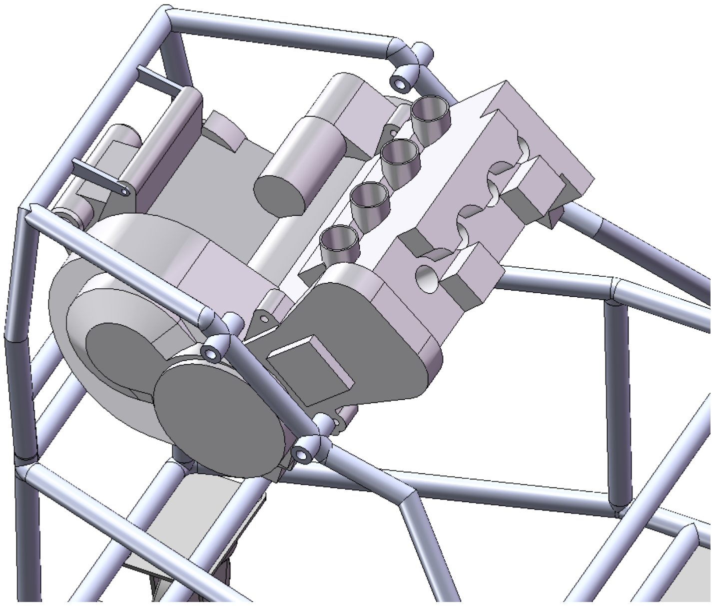

Figure 2.2 Engine inside of engine stand concept. (Stand modeled by D. Curran, engine modeled by members of the FSAE team)

After considering the potential problems brought up by Ronnie, the team decided to defer to the wisdom of David and Ronnie and let them modify the design as they saw fit. As a result, they were able to very quickly build an excellent engine stand made out of square steel beams and bolted the engine into the stand. The completed stand with the engine in it is shown in Fig. 2.3. (Several of the components shown in the picture were added well after the making of the engine stand.) The estimated cost of the parts and labor was $800, but they very generously let the team borrow the stand free of charge.

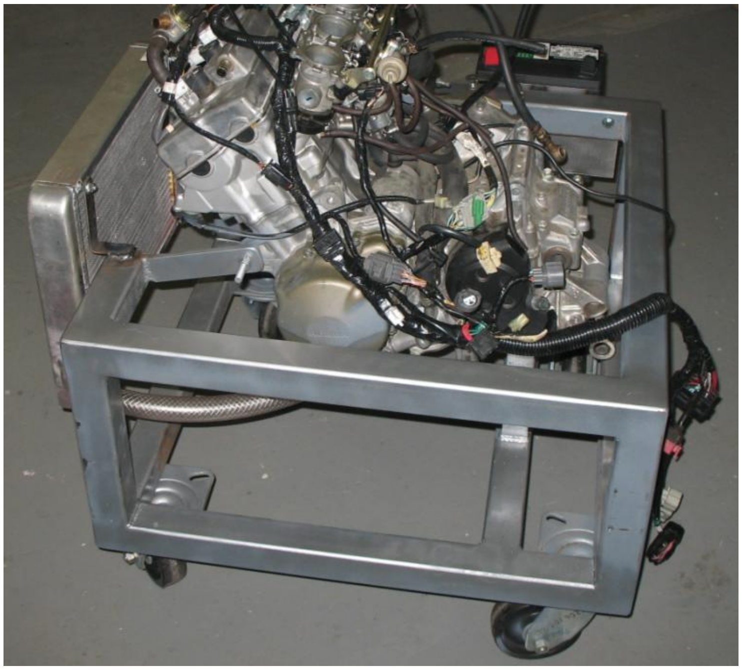

Figure 2.3 Engine bolted into engine stand.

The actual engine stand clearly came out very different than the design concept. The design of the physical stand is far simpler than the design concept for the stand. It also clearly addresses the problems of the assembled engine being too big to fit into the stand as well as the bolt holes not lining up properly.

With the engine securely bolted to the engine stand, the team turned its attention to getting the engine operational.

2.4 – Getting the Engine Running

Even though the team had an assembled engine on an engine stand, there was still a long way to go to get the engine running. The picture in Fig. 2.3 is somewhat deceptive because, as indicated in the previous section, many of the components clearly visible in the picture actually were not on the engine when it was first put on the engine stand. The radiator, battery, wiring harness, throttle bodies, fuel rail and fuel injectors were not on the engine, or even in the team's possession, when the engine was first put into the stand. They were acquired over a period of several months as part of the process of getting the engine running. In fact, when the engine was first put into the stand, there was not a whole lot more than the fully assembled block and cylinder head.

Given a lack of knowledge about engine components and a lack of clean and secure working space at WPI, the team took the engine and stand to Bruce and Keith Archambault's garage and workshop in Hollis, New Hampshire. They graciously offered a place to store and work on the engine as well as their years of experience of working with engines, all free of charge. They helped assess the state of the engine and determined what components were missing. Based on their recommendations, the team purchased stock components including a fuel pump, fuel rail with fuel injectors, throttle bodies, wiring harness, exhaust manifold and gaskets. A complete list of purchases can be found in Appendix B. An appropriately sized battery was purchased. A muffler was obtained from the WPI shop for free. The stock ECU and old FSAE intake manifold were temporarily borrowed from the FSAE team in order to get the engine running. The old FSAE intake manifold consisted of an air filter, restrictor, plenum, runners, and sensors. David Lamoureux, Sr. from Merchants Sheet Metal was again very helpful in manufacturing and lending a custom battery box of steel sheet metal and fuel tank of aluminum sheet metal, again free of charge.

Once all of these components were in hand, Keith assisted in assembling everything. The battery box was bolted to the stand, and the battery was inserted into it. The radiator was bolted to the engine stand, and the coolant hoses were attached to it. The stock throttle bodies were clamped to the intake ports, and the fuel rail and injectors were connected to the throttle bodies. The fuel pump was sealed in the fuel tank, which was then connected to the fuel lines of the engine. The stock exhaust manifold was bolted on to the exhaust ports with the new gaskets in between. At this point, the engine was ready to run mechanically.

Though the engine was in good shape mechanically, the electrical systems were still a problem. The stock wiring harness was connected to the sensors on the engine, and the stock ECU was then connected to the wiring harness. Keith wired a makeshift ignition switch to start the engine. With everything in place, the engine was theoretically ready to run. However, after multiple attempts, the engine still would not run. It would successfully turn over, but it would not run. The conclusion was that there was a problem with the wiring harness that was preventing a proper electrical signal from reaching the fuel pump, fuel injectors and spark plugs. Keith suggested that the problem was a result of using the stock ECU and wiring harness, which are supposed to be on a fully assembled motorcycle, on an engine stand that might be lacking some important sensors. The engine was then brought back to WPI to work on the electrical components.

In an attempt to solve this problem, the team bought the service manual for the 2001-2004 Honda CBR600F4i models. This manual contained the complete wiring diagram for the engine. After reviewing the diagram, some modifications were made. Several unnecessary connectors for things like headlights, brake lights, etc. were cut off of the wiring harness. Several other connectors needed to be bypassed. The ECU was expecting a signal from the kickstand and clutch sensors even though neither component was present on the engine stand. These sensors needed to be bypassed rather than simply cut, so the ECU would be receiving the signal it expected. However, after bypassing these sensors, the engine still would not run.

At this point, the team decided to enlist the help of the FSAE team, as they had experience wiring the engine to their chassis. All of the fuses were checked to make sure the problem was not there. A voltmeter was used to check all the connections and the battery, and everything had a strong voltage. The fuel pump was connected to the battery and ran properly. After further examination of the wiring diagram, the problem was found. The ignition switch needed to be altered. The bike actually had two ignition switches that needed to be activated for the engine to run. The first ignition switch was activated when the key was inserted into the ignition, and the second ignition switch was a push button connected to the starter relay, which actually started the engine. Since everything else checked out properly, the ignition switch problem was the only obstacle standing in the way. The FSAE team managed to construct a switch for their car, but they did not have time to assist in building a second one for this engine.

It was thus decided to completely redo the wiring. An aftermarket ECU was purchased for the turbocharged system at this point. (See section *11 – Electrical Systems* for details.) The aftermarket ECU came with a spool of wire for a new wiring harness. The plan was to build a new harness for the aftermarket ECU that would circumvent all the problems experienced with the stock harness. The FSAE team took the new spool of wire for their car, though. They offered the use of their completed wiring harness from the previous year. Theoretically this made the engine ready to go, both mechanically and electrically. Unfortunately, it was at this point that it was made clear that the engine required forty hours of tuning. Very few people at WPI are qualified to tune the engine, and it was so late in the year that none of them had the time to do it.

Getting the naturally aspirated engine running then became impossible. The goal shifted towards physically implementing the then completely designed turbocharged system. Assurances were made that the turbocharged system would be tuned if it were completely assembled.

2.5 – Results

The original engine was damaged and incomplete. It was given to Keith in exchange for his assistance. A new engine was found and purchased. David and Ronny manufactured an engine stand for the new engine. All necessary mechanical and electrical components were obtained and assembled. The engine successfully turned over but did not run. It would have taken an estimated forty hours to tune the engine, which was not feasible.

3 – Data Acquisition

Acquiring data from both the naturally aspirated engine and the turbocharged system was considered to be essential in order to properly design and optimize the turbocharged system as well as to compare the performance of each. In order to properly design and optimize the turbocharged system, several different types of data needed to be acquired such as airflow, horsepower, air temperature and pressure. Comparing the performance of the two separate systems, however, would only require horsepower data.

3.1 – Airflow Data

Airflow data refers to the rate at which the engine consumes air. This airflow rate depends on engine speed and displacement. It is also affected by whether the engine is naturally aspirated or under boost. Airflow is important for this project because it is a major factor in properly sizing the turbocharger compressor. The engine's airflow while naturally aspirated is used to predict airflow under boost which is plotted on the compressor's map to determine how efficient the compressor will be. This process of compressor mapping will be discussed more in section *4 – Turbocharger Selection*. Suffice it to say for now that the airflow data of the naturally aspirated engine across its rev range is necessary to properly choose a turbocharger for the engine.

Normally, the airflow rates of the naturally aspirated engine can be calculated based on known engine displacement and engine speed and an estimated volumetric efficiency. These calculated values are then used to predict boosted airflow to map the compressor. However, the FSAE restrictor makes it impossible to accurately do this. While the engine displacement and speed will still be known, the effects of the restrictor on the volumetric efficiency of the engine are unknown. It is reasonable to assume that the restrictor hurts the engine's volumetric efficiency, which is really just a measure of how much of the cylinder's volume is actually filled with air. Ideally, efficiency would be 100% and the cylinder would be completely filled with air. In reality, the efficiency typically varies between 50-90% depending on the engine speed and the design of the intake system. One can expect a drop in efficiency over the full rev range as a result of the restrictor. Figure 3.1 shows the difference between an unrestricted and restricted intake. Obviously, the unrestricted intake will allow much better airflow than the restricted intake.

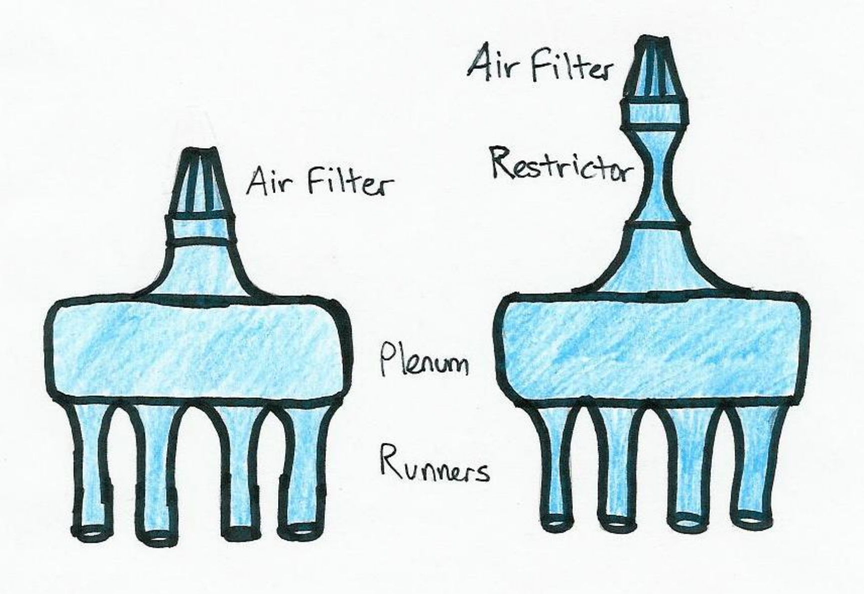

Figure 3.1 Unrestricted (left) and restricted (right) intakes. (Drawn by D. Curran)

In addition to hurting the volumetric efficiency of the engine throughout the rev range, the restrictor has a maximum airflow rate that it permits. The throat of the restrictor is 0.7874 inches in diameter. Since the throat has a circular cross section, the area of the throat is 0.4869 square inches or 0.003381 square feet. The maximum velocity that the air can reach through the throat is approximately equal to the speed of sound, about 1130 ft/sec. Since airflow is equal to velocity multiplied by area, the maximum airflow can be calculated and converted to 229 CFM (cubic feet per minute). This means that airflow into the engine can never exceed 229 CFM. This probably would not be an issue with a naturally aspirated engine, but it very well could be an issue with a turbocharged engine.

The complications posed by the restrictor made it next to impossible to accurately predict the airflow into the naturally aspirated engine. It thus became necessary to acquire this data experimentally. The plan was to use an air mass flow sensor that the FSAE team had to measure airflow into the running naturally aspirated engine. The sensor was going to be installed in the intake manifold after the restrictor. Once the engine was running, the airflow and engine speed data were going to be exported to a computer spreadsheet. The data could then be used to predict boost airflow and map the compressor. It could also be used to anticipate when the restrictor would choke, when the engine would consume 229 CFM.

Obviously, since the naturally aspirated engine never ran, this airflow data was never acquired. Unfortunately, the airflow data acquisition system was never completed either.

3.2 – Horsepower Data

Horsepower data basically consists of a graph of horsepower versus engine speed, which is simply a measure of how much power the engine is producing at a given speed. This data is useful for two important reasons. It provides a basis of comparison between the two systems, naturally aspirated and turbocharged. It also helps in the design of the turbocharged system.

Being able to quantitatively compare the turbocharged system to the naturally aspirated system makes it possible to definitively say which system provides superior performance. The naturally aspirated engine with restricted manifold would need to be run on a dynamometer over its full rev range to determine its base horsepower. This data could then be analyzed to determine how the restrictor reduces the engine power. The shape of the horsepower curve could then be used to figure out what part of the rev range experiences peak power and how broad that peak is. This information would be very useful in designing the turbocharged system. The turbocharger should be chosen such that it complements the part of the rev range that produces the most power. Once the turbocharged system was complete, it could be put on the dynamometer to find its horsepower curve. Comparing this horsepower curve to the one from the naturally aspirated engine would show how successful the turbocharging effort was in increasing power.

The need for horsepower data is pretty clear. Acquiring the data is no small task though. Both the naturally aspirated engine and the turbocharged system would be running on the engine stand since the team had no chassis to make use of. This ruled out the possibility of using a chassis dynamometer, which the most common and readily available type. Some kind of engine dynamometer would be required to get the necessary data. Since there is no engine dynamometer available at WPI, the team was left with three possibilities to explore: an engine dynamometer could be purchased, time could be rented on somebody else’s engine dynamometer, or a dynamometer could be built.

Extensive time was spent online trying to find new or used engine dynamometers for sale, but all of the dynamometers found were far too expensive to purchase. A few leads were contacted in an effort to find an engine dynamometer to rent time on, but they did not immediately result in anything. However, Al Smyth of Portatree Timing Systems in Uxbridge, Massachusetts did indicate that when the engine was operational, he might be able to find an engine dynamometer to rent time on. Finally, it was decided that there simply was not time to design and build a properly functioning engine dynamometer in addition to everything else that had to be done.

Ultimately, as it became clear that the engine was not going to run, the need for an engine dynamometer diminished. Efforts to get an engine dynamometer were eventually abandoned.

3.3 – Air Temperature and Pressure

Once the turbocharged system was finished, there was an intention to install several temperature and pressure sensors throughout the intake manifold to acquire data about the effects of the components. Referring to Fig. 3.2, sensors would have been placed (1) between the restrictor and compressor, (2) the compressor and intercooler, and (3) the intercooler and plenum. These sensors would have provided data about the temperature and pressure changes across the restrictor, compressor and intercooler. This data could have been used to improve the design of the restrictor, confirm the efficiency of the compressor and intercooler, and locate the best places to source the wastegates. However, since the turbocharged system was never operational, this plan was never taken out of the initial stages.

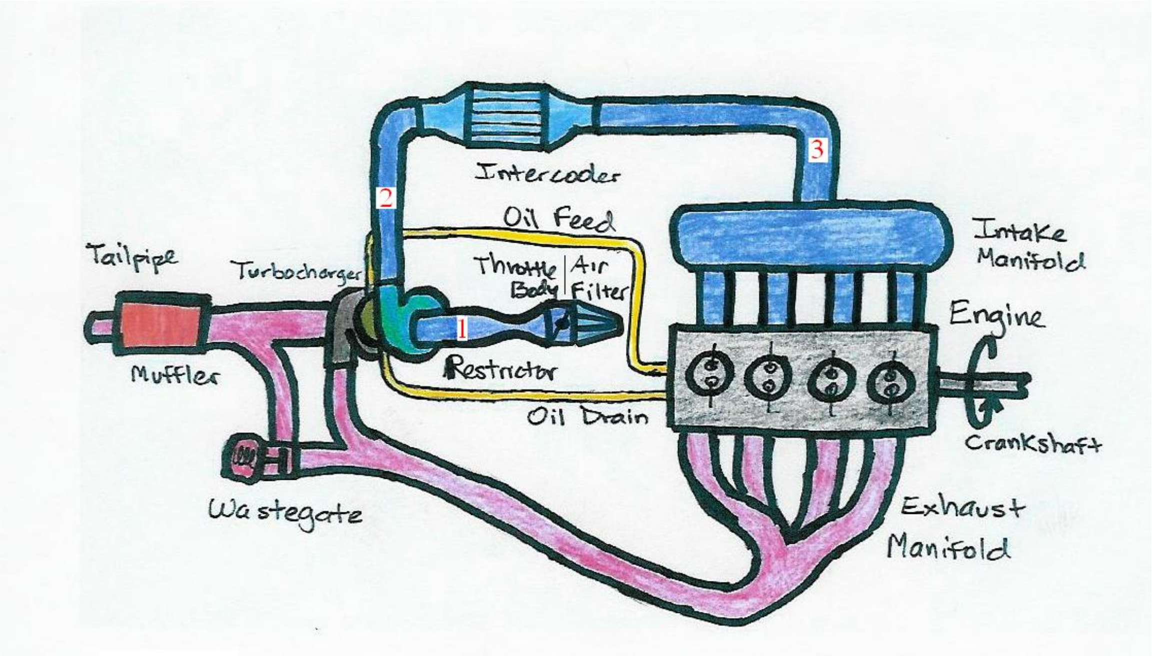

Figure 3.2 Turbocharged FSAE system with three sensor locations. (Drawn by D. Curran)

3.4 – Results

Since none of the data acquisition systems were ever realized, there benefits could not be enjoyed. However, not having an operational engine would have been a problem even should the data acquisition systems been completed. Without the airflow data, the compressor could not be properly sized. Sizing thus had to be approximated, poorly at that. Without horsepower data, a baseline could not be established for comparison, making it next to impossible to determine whether or not the designed turbocharged system actually would have improved power. Without the temperature and pressure sensors, the turbocharged system would not have been able to have been optimized had it been operational.

4 – Turbocharger Selection

The selection of the turbocharger for a system is possibly the most important decision that there in the entire process. A properly selected turbocharger with some nice features will wonderfully compliment the engine while simplifying the system design and plumbing. However, a poorly chosen turbocharger will do little to increase engine power while causing one headache after another over the system design and plumbing. A turbocharger is chosen primarily based on its size but there are several other desirable features to look for as well. The team went through a lengthy turbocharger selection process marked by repeated compromises leading to the final decision.

4.1 – Turbocharger Sizing

There are two separate parts of the turbocharger that need to be sized correctly, the compressor and the turbine. However, before either of these components can be sized, the objectives of the turbocharged system need to be determined. While engine displacement and peak horsepower are important pieces of information, a little planning needs to be added to this knowledge before proceeding with the selection of a turbocharger.

4.1.1 – Planning Ahead

The first step in choosing a turbocharger size is determining what the turbocharger would have to do. This means coming up with specific performance objectives as well as determining exactly how the engine is going to be used. The performance objectives are directly related to engine power, either horsepower or torque, and what kinds of power gain are desired. The operational objectives are generally related to issues of drivability and durability.

Specific performance objectives tend to be measurable quantities. For example, a particular objective could be to have 500 horsepower at 5000 RPM. This is not always the case, though. A performance objective could also be to simply raise the horsepower curve at engine speeds above 4000 RPM. These two examples illustrate the difference between peak and broad power gains. The first example would be a case of peak power gains. A naturally aspirated engine will experience a peak power output at a specific engine speed with power output less at both lower and higher engine speeds. One type of performance objective is to simply raise this peak power output while ignoring the rest of the horsepower curve. In this case, the turbocharger is chosen to operate most efficiently at the precise RPM at which the engine hits peaks power, while the turbocharger efficiency at other engine speeds is marginalized or ignored.

The second example is a case of broad power gains. If a broad power gain is the objective, then peak power is basically ignored as it occurs at a very specific RPM. In choosing a broad power gain performance objective, a range of engine speeds is selected over which the entire horsepower curve is to be elevated. This range of engine speeds can usually be characterized as either low-speed, mid-range or high-speed with the actually RPM values depending on the size of engine’s overall rev range. One of these speed ranges would be chosen to have its power improved, and the turbocharger would then be selected such that it operated most efficiently in that part of the rev range. Choosing the area of the rev range to improve upon has a lot to do with what kind of driving the automobile is going to be doing. Most street applications would probably look towards improving low and mid range power while racing applications tend more towards improving high speed power.

In addition to performance objectives, some operational objectives should be determined as well. These objectives generally involve the drivability and durability of the automobile and its engine. The drivability of the automobile is greatly impacted by boost threshold and turbo lag. The durability of the engine is directly related to heat.

Boost threshold is the engine speed at which the turbocharger starts to create enough boost for the engine to start generating more power. The amount of boost created by the turbocharger is linked to the compressor speed which is equal to the turbine speed which is driven by the amount of exhaust flowing through it. The amount of exhaust flowing through the turbine depends on the engine speed. Lower engine speeds produce less exhaust. This means that there is not much exhaust making it to the turbine at low engine speeds, which in turn leads to little boost pressure. There is actually an engine speed below which the turbocharger is spinning so slowly that no noticeable boost is being created, and this engine speed it known as the boost threshold. Prior to the boost threshold, the horsepower curve will pretty be the same as the naturally aspirated curve but it will shoot up once the boost threshold is reached. Accelerating through the boost threshold can lead to a brief power surge affecting drivability. The location of the boost threshold and its impact on drivability are dictated by the sizing of the turbocharger.

Turbo lag refers to the time delay between stepping on the gas pedal and increased power from the turbocharger. At low engine speeds, there can be over a second of waiting between flooring it and enjoying the benefits of the turbocharger. The presence of turbo lag cannot be eliminated as it is a result of the turbocharger needing the extra exhaust from the higher engine speed to rotate fast enough to create the extra boost. Turbo lag tends to decrease as engine speed increases past 4000 RPM and it can be reduced at lower engine speeds by choosing the correct sized turbocharger.

The turbocharger tends to create more heat in the engine system. The turbocharger raises the temperature of the intake air as it compresses it and increases the temperature of the exhaust as a result of the flow restriction posed by the turbine increasing its pressure. This added heat decreases the durability of the engine and can damage various parts of the system. While several other steps like adding an intercooler are important, choosing the proper turbocharger plays a big role in reducing the heat in the engine. The sizes of the compressor and turbine can be chosen as to minimize the heat added to the intake air and exhaust, respectively, at particular rev ranges.

With specific performance objectives chosen and the concepts of boost threshold, turbo lag and heat durability in mind, the actual sizing of the compressor and turbine can be discussed.

4.1.2 – Compressor Sizing

The size of the turbocharger compressor plays a major role in meeting performance objectives and reducing heat but is not so important in the areas of boost threshold and turbo lag. The compressor size is important in meeting performance objectives and reducing heat because its efficiency can vary greatly under different conditions and at different engine speeds. When operating efficiently, the compressor adds relatively little heat to intake air, increasing engine power and reducing overall heat in the system. When operating inefficiently, the compressor adds a lot of heat to the intake air, reducing power gains and increasing the overall heat in the engine. Hotter intake air hurts power gains because it is less dense than colder air, and denser air means more air in the cylinder, which means more fuel and more power. The compressor must be sized such that it operates most efficiently at the proper part of the rev range to meet performance objectives and limit heat added to the system. The boost threshold and turbo lag are not impacted by the compressor size but are products of the turbine size.

A compressor is sized based on two pieces of information, boost pressure and airflow. The desired boost pressure is chosen based on the performance objectives. Generally speaking, the minimum boost pressure needed to achieve the performance objectives is chosen. The airflow is directly related to the engine speed and is thus calculated based on what part of the rev range is desired to experience a power increase. Once the boost pressure and airflow are known, they are used to size the compressor. When the compressor was discussed in section 1.4.1, it was stated that the compressor works best at a particular combination of airflow and boost pressure. A comparison of this optimal combination of boost pressure and airflow for a given compressor to the anticipated combination of boost pressure and airflow determines the suitability of the compressor for the system.

A compressor’s optimum boost pressure and airflow is expressed on a compressor map. Figure 4.1 shows a sample compressor map that happens to be for a Garrett GT15 turbocharger. The x-axis is for the airflow, and the y-axis is for the boost pressure. Notice that the airflow is expressed as lb/min and not CFM, and the boost pressure is expressed as a pressure ratio, the ratio of absolute pressure under boost to atmospheric pressure. Equation 4.4 can be used to convert airflow in CFM to lb/min, and Eq. 4.2 can be used to convert boost pressure into a pressure ratio. Equation 4.1 is included to show how to calculate the basic airflow of the naturally aspirated engine, and Eq. 4.3 is used to calculate the predicted engine airflow under boost. The airflow on the compressor map is the predicted airflow under boost calculated with Eq. 4.3. Equations 4.1-4.4 can be used with a known engine displacement, an estimated or measured volumetric efficiencies and a desired boost pressure to properly map the compressor.

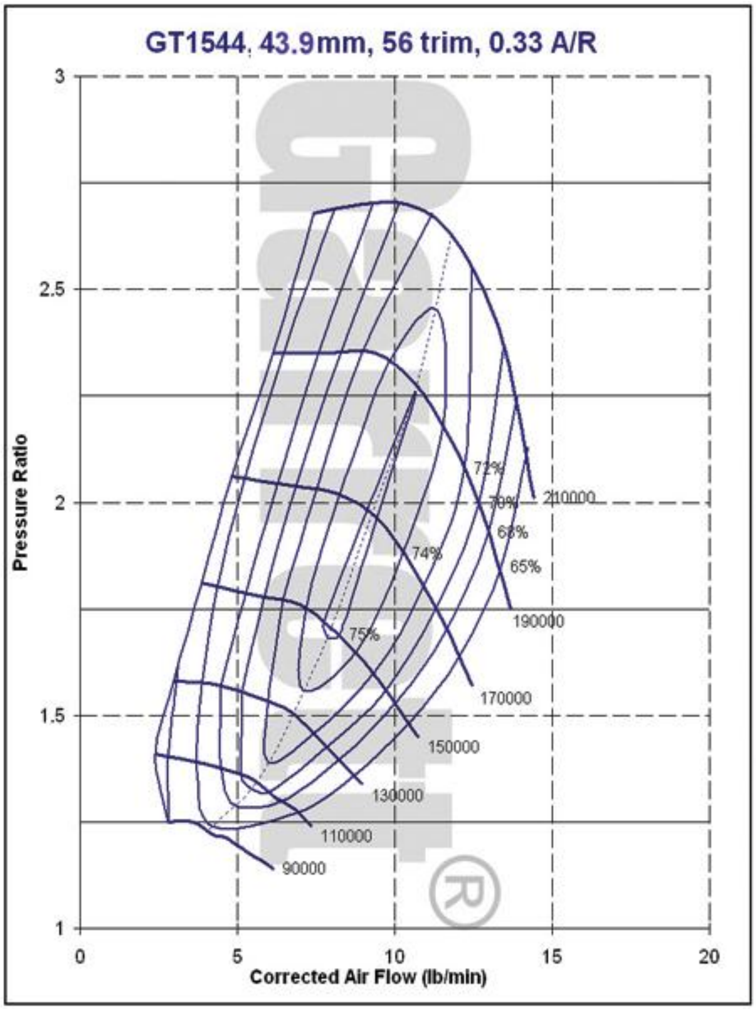

Figure 4.1 Sample compressor map for Garrett GT15. (Retrieved from http://www.turbobygarrett.com/turbobygarrett/catelog/Turbochargers/GT15/GT1544_454082_2.htm on April 18, 2008)

$$\text{Basic Airflow (CFM)} = (cid * rpm * 0.5 * E_v)/ 1728 \tag{4.1}$$

cid = engine displacement in cubic inches
rpm = engine speed in revolutions per minute
E_v = volumetric efficiency of the engine

$$\text{Pressure Ratio} = (14.7 + Boost) / 14.7 \tag{4.2}$$

$Boost$ = boost pressure in pounds per square inch

$$\text{Turbo Airflow (CFM)} = \text{Pressure Ratio} * \text{Basic Airflow} \tag{4.3}$$

$$\text{Airflow (lb/min)} = CFM / 14.27 \tag{4.4}$$

CFM = airflow rate in cubic feet per minute

The compressor is mapped by plotting three points on the compressor map and connecting them with straight lines. The first point is at the airflow calculated for the redline RPM and the pressure ratio for the desired boost pressure. The second point is at the airflow calculated for the engine speed one half of the redline RPM and the pressure ratio for the desired boost pressure. The third and final point is at the airflow equal to 20% of the maximum airflow and a pressure ratio of one. Figure 4.2 shows what the compressor map will look like with these points plotted and connected. The efficiency at point 1 is primarily what determines how well a fit the compressor is for the turbocharged system. In this example, the efficiency appears to be about 63%. Peak compressor efficiency is usually somewhere in the low to mid seventies, but anything above 60% should be considered acceptable. The line between points 2 and 3 must be completely to the right of the left most line of the map in order to prevent instability in the turbocharger known as compressor surge. This particular compressor would be a pretty good fit the system based on the sample points plotted.

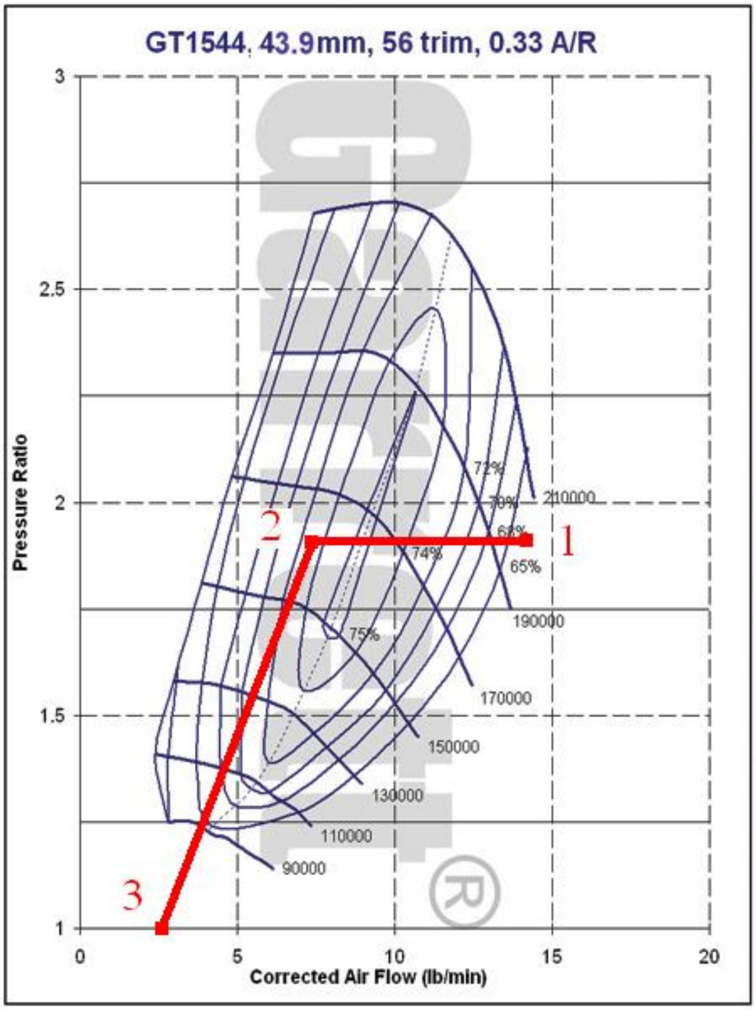

Figure 4.2 Plotted sample compressor map. (Retrieved from http://www.turbobygarrett.com/turbobygarrett/catelog/Turbochargers/GT15/GT1544_454082_2.htm and edited on April 18, 2008)

With proper airflow data and a desired boost pressure, the compressor can be mapped to determine how well it will work for the system being designed. The higher the efficiency at point 1, and the more of the plotted lines that line in the peak efficiency island of the map, the less heat the compressor will add to the intake air, resulting in higher engine power and better engine durability.

4.1.3 – Turbine Sizing

Ironically, the selection of the turbine size is a bit simpler than the sizing of the compressor despite the fact that the turbine size impacts the performance objectives, boost threshold, turbo lag and heat added. There are no complicated maps or processes used to select the turbine size. It is basically just a balancing act. Smaller turbines will provide lower boost thresholds and better turbo response but only be able to create limited airflow through the compressor and will create a lost of back pressure and heat in the exhaust manifold. Larger turbines, conversely, allow much more airflow through the compressor and reduce back pressure and heat in the exhaust manifold at the expense of higher boost thresholds and larger turbo lag. The airflow through the compressor is directly related to the boost pressure the compressor creates. Therefore larger airflow through the compressor means higher boost pressures. Back pressure in the exhaust manifold can lead to reversion, which is when the pressure is so high in the exhaust manifold that exhaust gases are forced back into the cylinder when the exhaust port opens during the exhaust stroke. This obviously not good and hurts engine power. These opposing concerns must thus be balanced against each other.

Once again the manner in which the automobile is to be used will come into play here. Performance objectives focused on high RPM ranges will likely require a large turbine, whereas low range objectives are better suited for a small turbine. The size of the turbine can be described either by its exducer bore or its A/R ratio. Using the exducer bore to size the turbine provides a ballpark size but it is pretty crude. Figure 4.3 provides a general guideline as to bore size based on airflow through the compressor, the value calculated using Eq. 4.3. Using the graph in Fig. 4.3, the approximate size of the turbine can be determined based on the compressor airflow. The rough size indicated is for a turbine that will adequately balance the design considerations described above, the A/R ratio needs to be used to fine tune the turbine size for maximum benefit.

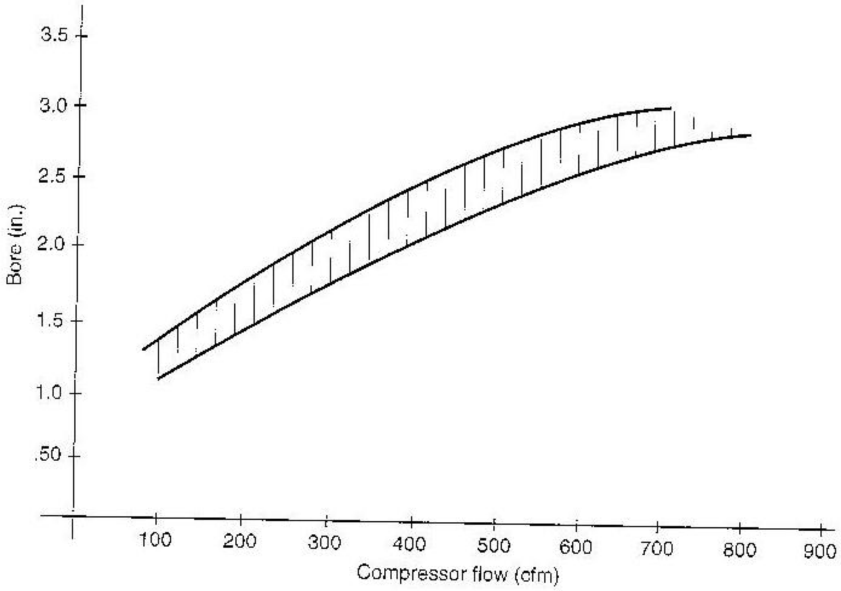

Figure 4.3 Turbine exducer bore for given compressor airflow. (Retrieved from Corky Bell's *Maximum Boost*)

The A/R ratio of the turbine is the ratio of the housing discharge area to the radius of the center of the discharge area to the center of the turbine blades. Figure 4.4 illustrates this relationship. Each circular area represents the discharge point of the turbine housing. The discharge point is the circular hole in the turbine housing through which the exhaust gases flow into the turbine blades. The smaller the area, the faster the gases flow into the turbine but the higher the exhaust back pressure. The reverse is true of a larger discharge area.

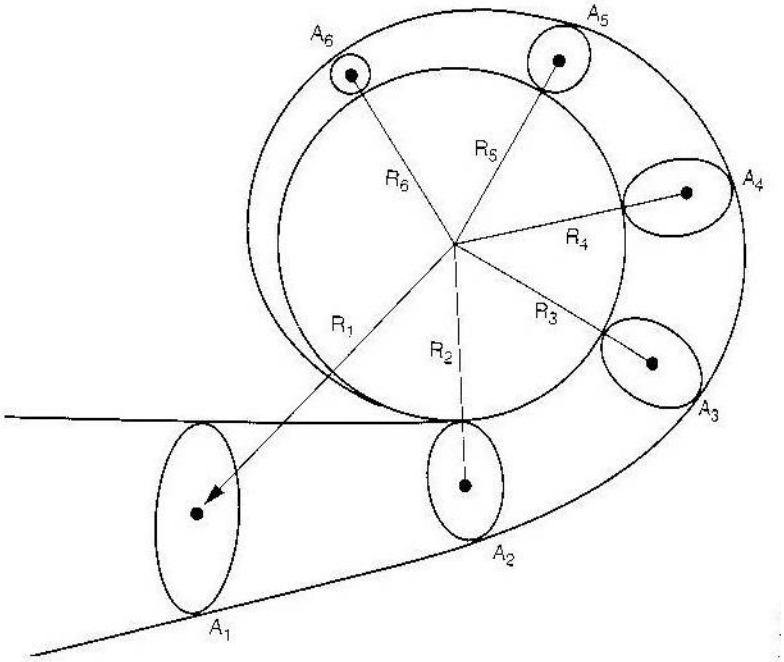

Figure 4.4 Illustration of turbine A/R ratio. (Retrieved from Corky Bell's *Maximum Boost*)

Selection of an appropriate A/R ratio can be a tricky task. It oftentimes requires some trial and error. If it is not possible to try different A/R ratios and test the results, then Fig. 4.5 can be used as a guide for selecting the A/R ratio. As can be seen from the figure, higher A/R ratios tend to produce more power from the engine. Lower A/R ratios, on the other hand, lead to better low-speed response, which means lower boost threshold and smaller turbo lag. As always, a balance needs to be struck depending on the desired system performance.

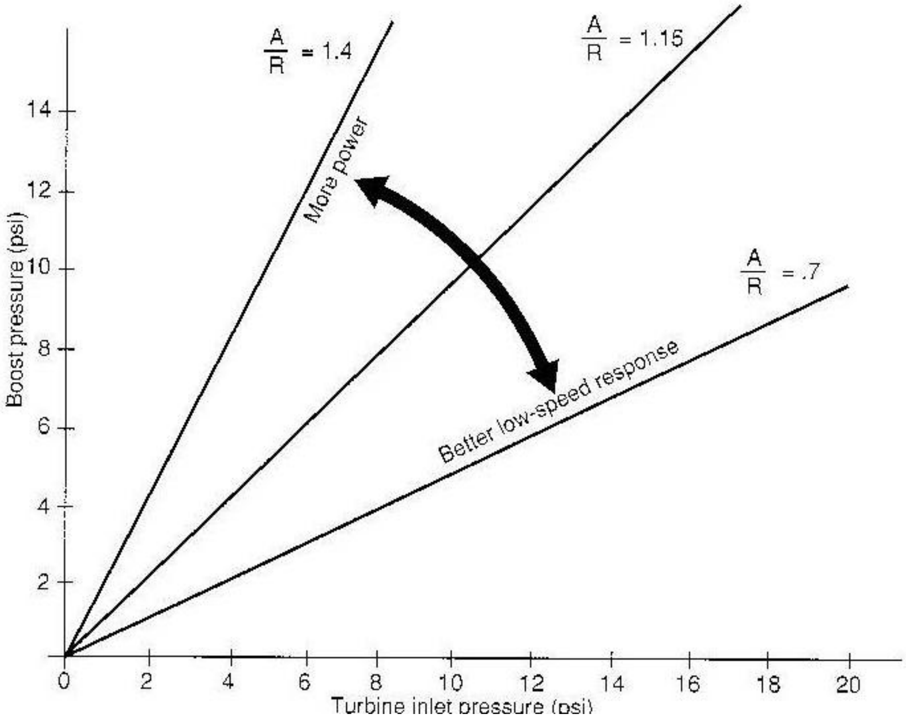

Figure 4.5 Guide for selecting a turbine A/R ratio. (Retrieved from Corky Bell's *Maximum Boost*)

4.2 Desirable Features of a Turbocharger

While the abovementioned sizing of the compressor and turbine is far and away the most import part of choosing a turbocharger, several other features contribute to making a turbocharger the right choice. It would difficult to provide an exhaustive list of everything that makes a turbocharger desirable. However, there are several pretty important features to look out for that will determine whether the turbocharger is a quality machine or a piece of junk.

The single most beneficial feature is a water jacket for the bearings section of the turbocharger. Section 1.4.1 already introduced the concept of a water jacket, and the details of this feature will be discussed in section *10 – Turbocharger Bearings*. The importance of having a water jacket will be discussed presently. The rotating shaft between the compressor and the turbine can rotate up to 200000 RPM while being exposed to some pretty high temperatures. The high speed of the rotating shaft makes maintaining properly lubricated bearings vital. Without a water jacket around the bearings, the oil lubricating the bearings can easily be broken down, resulting in charred oil inside the turbocharger preventing proper flow. In this case, the quality of oil being used and very frequent oil changes become very important in preventing oil coking. A water jacket will eliminate the problem of oil coking by keeping the oil cool enough that it does not break down. Using quality oil and getting regular oil changes are still important but not as vital. A water jacket is a purely beneficial feature to have on a turbocharger that can greatly extend the turbocharger's life and ease maintenance requirements.

Another purely beneficial feature of a turbocharger is the ability to rotate the compressor, bearings and turbine sections independent of each other with full 360° freedom, which is known as clocking. This is an important option to have due to oil and plumbing requirements. For reasons that will be explained in section *10 – Turbocharger Bearings*, the oil outlet of the bearings section must be facing down, as near vertical as possible. It is extremely helpful to be able to rotate the bearings into a position that satisfies this requirement while being able to orientate the compressor outlet and turbine inlet independently to facilitate plumbing. Most turbochargers without an integral wastegate allow the ideal type of clocking described here. However, most turbochargers with an integral wastegate, Fig. 4.6, do not allow this type of full clocking.

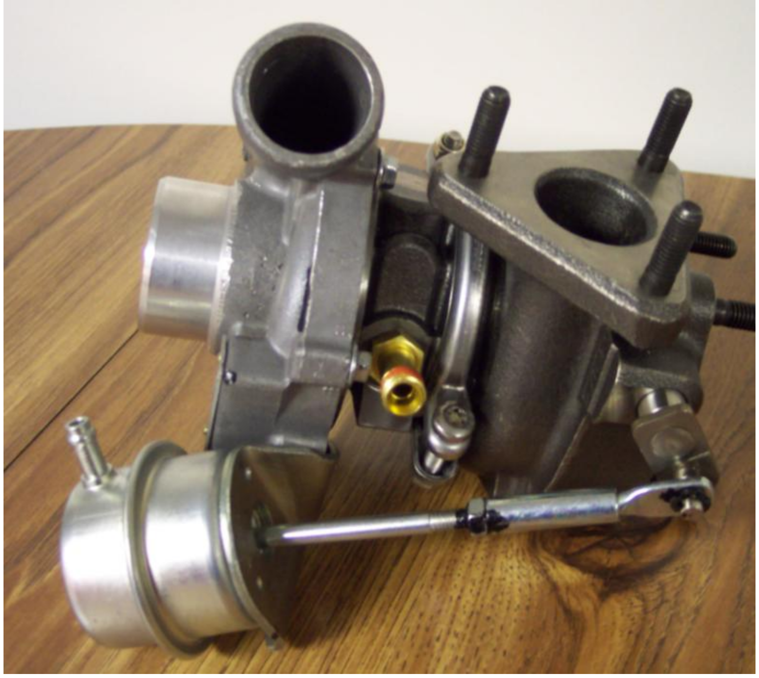

Figure 4.6 Turbocharger (Garret GT12) with integral wastegate.

The above picture of a Garrett GT12 shows a turbocharger consisting of the aforementioned (from left to right) compressor, bearings and turbine with an additional component in the foreground, an integral wastegate. The cylinder at the end of a shaft that is visible is actually the wastegate housing. The wastegate itself is inside the turbine housing. Wastegates will be described in more detail in section *8 – Boost Control*. Suffice it to say that integral wastegate will satisfy the system need for boost control, which means that no additional wastegate will be needed as shown in Fig. 1.11. This does tend to save money and space in the crowded engine compartment while simplifying the exhaust plumbing. However, an integral wastegate is not all good. Most integral wastegates create back pressure problems that hurt engine power and foster reversion while also causing the previously mentioned clocking problem. Put simply, integral wastegate are cheap and easy way of providing boost control, but any serious performance engine will benefit from avoiding an integral wastegate.

Finally, the connections for all inlets and outlets should be looked at. The compressor inlet and outlet will both likely be designed for a hose to be clamped on to them. If this is the case, they should have a lip to assist in clamping. Their diameters should also be standardized so standard sized hose can be purchased. The bearings section will have an oil inlet and outlet for sure but may also have a water inlet and outlet. These inlets and outlets should be examined to determine how easy it will be to acquire and connect the right hoses for the oil and, possibly, water. The oil inlet and outlet absolutely should not be on the same face. They should be on opposite faces, Fig. 4.7. The same should be true for the water inlet and outlets. If the inlet and outlet are on the same face, the small size of the turbocharger can create major problems in fitting all the necessary connections into that small area. The turbine inlet and outlet will most likely be bolt on connections requiring flanges. Preferably the turbocharger manufacturing will produce gaskets and flanges for the turbine, which saves the trouble of having to design and machine custom ones. Thicker flanges and flange connections seal better and are less likely to fail under the stress they experience.

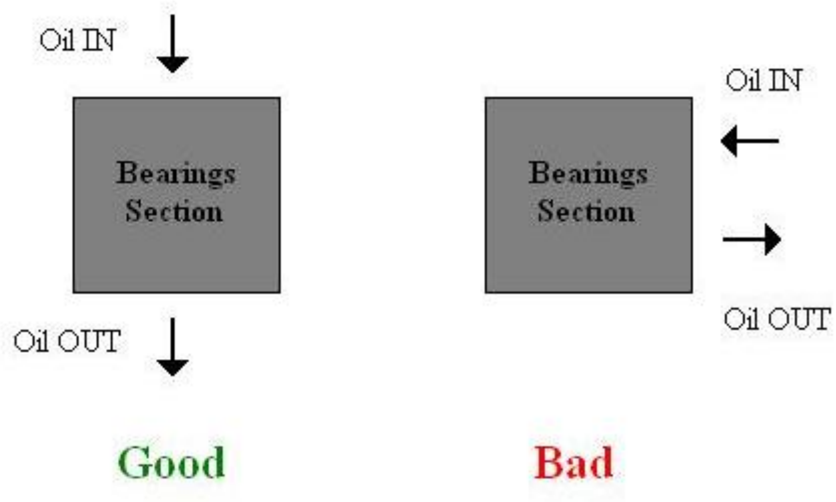

Figure 4.7 Good and bad oil inlet and outlet orientations.

4.3 – Finding the Right Turbocharger for the Project

At the start of the project, the team was given an unused Garrett GT12 turbocharger, pictured in Fig. 4.6. This turbocharger was selected and purchased by last year's team. After reviewing the rationale for their selection in their report, the decision was made to start the turbocharger selection process from scratch. This decision was made after examining the airflow data that was used to map the compressor. The data seemed to contradict itself and its source was not cited, making it suspect. The plan for this year was to get the naturally aspirated engine running with a restricted intake, and to then use an airflow sensor to experimentally acquire airflow data necessary to properly map a compressor. An ideal compressor map was to then be constructed and used to locate the most compatible turbocharger. Since the engine never successfully ran and the data acquisition system was never completed, this experimental data was never obtained and an ideal compressor map was never made. The turbocharger selection process thus had to proceed without enough data to make a proper decision.

In addition to the lack of data problem, a lack of choices problem also existed. Unfortunately, there is not much of a market for turbochargers for 600cc engines. Most of the commonly available turbochargers are for engines 1000cc and larger, with an emphasis on larger. Further complicating things was the restrictor. The restrictor was going to cause the engine to consume less air and produce less power than a normal 600cc engine. This means that a turbocharger designed for a 600cc engine could end up being too large for this system. The team decided to try to find the smallest turbocharger it could to try to compensate for the intake restrictor. Since no empirical data could be gathered to properly size the turbochargers, this seemed like as good a plan as any.

The search for small turbochargers thus began. One requirement, however, was that any turbochargers that were found had to have a compressor map. Even though the system could not be properly plotted on the map, being able to compare the compressor maps of two turbochargers is enough to determine which is better suited for smaller airflow rates characteristic of smaller engines. The team spent an extensive amount of time trying to find alternative turbochargers. The internet was searched from auction websites to turbocharger manufacturer websites to forums. Junkyards and motorcycle shops were contacted. An old turbocharger found lying around the shop was even investigated until it was found to have severely damaged bearings. However, after all that, there were no viable alternatives. The closest thing to a feasible alternative was the IHI RHF3. The IHI website indicates that this turbocharger is well suited for engines as small as 550cc and handles airflows up to 222.5 CFM. A complete compressor map could not be found for the RHF3, but Fig. 4.8 shows a simplistic compressor map comparison between the RHF3 and other members of the IHI RH series.

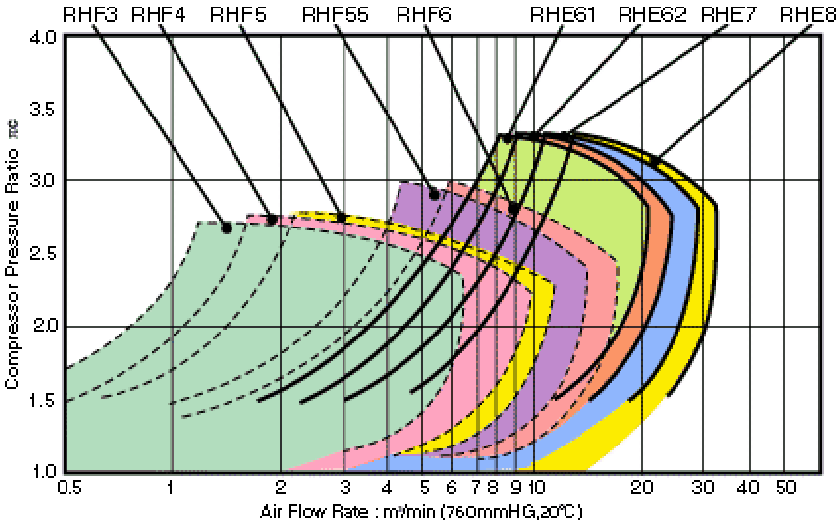

Figure 4.8 IHI RHF3 compressor map comparison chart. (Retrieved from http://www.ihi-turbo.com/turbo_RHE-RHF.htm on April 19, 2008)

Since the above compressor map contains no efficiency lines, it is not worth a whole lot. In contrast to the IHI RH3, Garrett claims its GT12 is well suited for engine displacements between 400-1200cc and horsepower between 50-130hp. Figure 4.9 shows the compressor map for the Garrett GT12, which happens to be a complete compressor map. Since these two compressor maps use different airflow units and scales, it is impossible to just eyeball them for comparison sake. However, there is a flawed method of comparison that can be used.

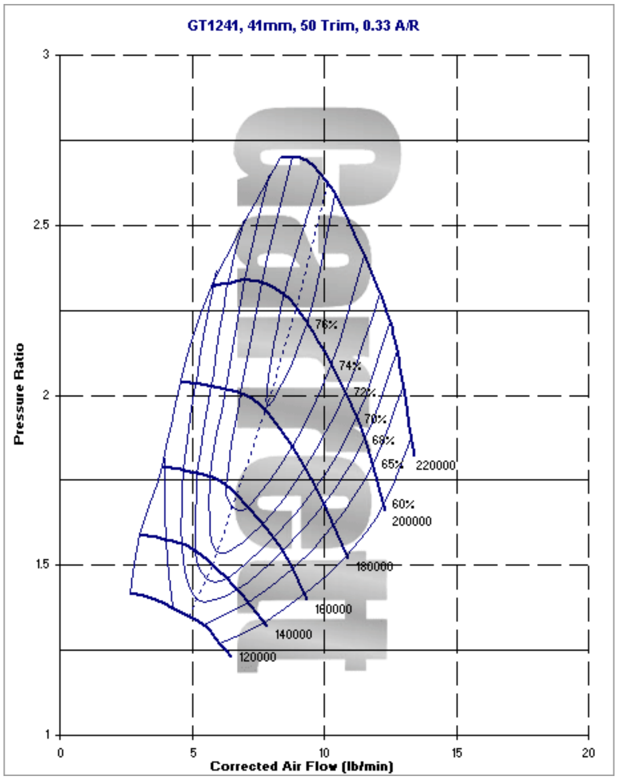

Figure 4.9 Compressor map for Garrett GT12. (Retrieved from http://www.turbobygarrett.com/turbobygarrett/catelog/Turbochargers/GT12/GT1241_756068__1.htm on April 19, 2008)

The reason that a compressor cannot be properly mapped is the lack of experimental airflow data, which is needed to account for the intake restrictor. There are equations that can be used to calculate expected airflow values that do not account for an intake restrictor, though. Thus one method of comparison between these two turbochargers is to map calculated points for a 600cc engine without an intake restrictor. While far from perfect, this would at least allow some level of comparison between the two compressors.

Equations 4.1-4.4 can be used to calculate anticipated airflow and pressure ratios necessary to plot a compressor map. The engine displacement is 600cc, which equals 36.6142 cubic inches. The redline engine speed is 14500 RPM, making half the redline 7250 RPM. For an unrestricted intake, the volumetric efficiency is assumed to be 85%. This is merely an assumption for calculations. The actual volumetric efficiency fluctuates greatly over the rev range and would really need to be obtained experimentally. However, 85% is an entirely reasonable assumption. With this information and Eq. 4.1, the basic airflow can be calculated. Two different boost pressures will be plotted. In order to maximize the horsepower of the engine, the team ideally wanted to produce about 18psi of boost. However, due to flow problems with the restrictor and the extensive requirements this high boost pressure would require, the team decided to initially shoot for a relatively low 7psi of boost. Thus both 7psi and 18psi will be plotted on the compressor maps in the form of pressure ratios, calculated using Eq. 4.2, of 1.476 and 2.224, respectively. The airflow data and pressure ratios can be used to calculate the corrected airflow rates from Eq. 4.3. Equation 4.4 can then be used to convert these values in CFM to pounds per minute. Additionally, multiplying the airflow in CFM by 0.028 will yield airflow in cubic meters per minutes. The results of these calculations can be viewed below in Table 4.1.

Point	**Boost in psi**	**Pressure Ratio**	**Airflow in CFM**	**Airflow in lb/min**	**Airflow in m^3/min**
1	7	1.476	192.74	13.51	5.40
2	7	1.476	96.37	6.75	2.70
3	0	1	38.55	2.70	1.08
1	18	2.224	290.41	20.35	8.13
2	18	2.224	145.20	10.18	4.07
3	0	1	58.08	4.07	1.63

Table 4.1 Pressure ratios and airflow for compressor mapping at $V_e = 0.85$.

The point designations refer to the previously description of compressor mapping in section 4.1.2 corresponding to Fig. 4.2. The airflow values are calculated in CFM but converted to cubic pounds per minute and cubic meters per minute so they can be plotted on the compressor maps for the GT12 and RHF3, respectively. These values are shown plotted in Fig. 4.10 and Fig. 4.11 on the compressor maps of the RHF3 and GT12, respectively.

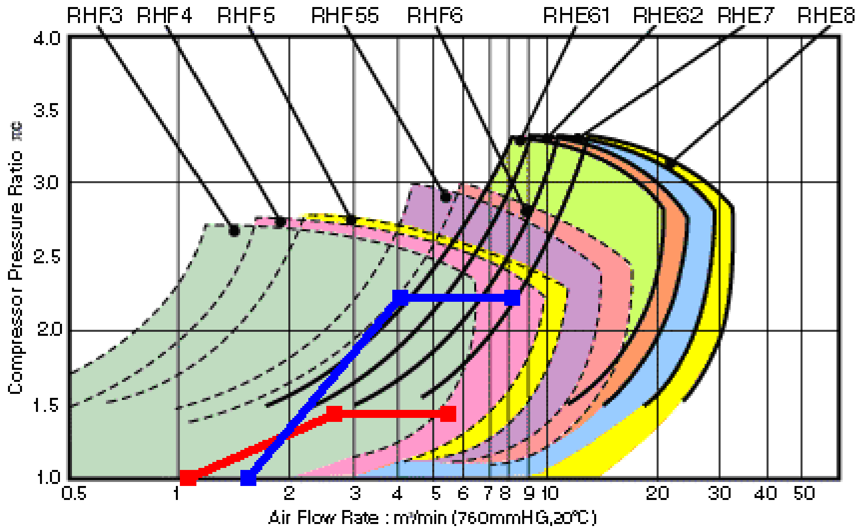

Figure 4.10 7psi (red) and 18psi (blue) plotted on RHF3 compressor map.

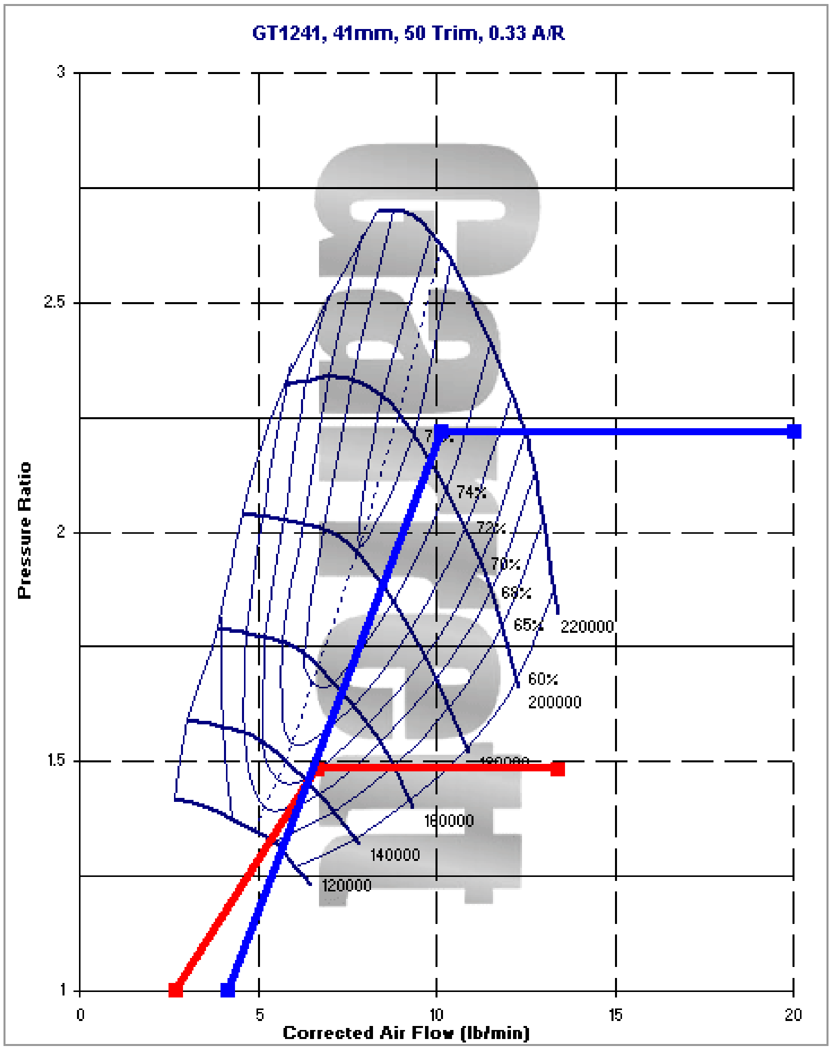

Figure 4.11 7psi (red) and 18psi (blue) plotted on GT12 compressor map.

These results indicate that neither of these turbochargers would be a great fit for an unrestricted 600cc engine. While everything is to the right of the surge line as it should be, the first point in all four cases appears to be at a point on the map well below 60% efficiency, which should be considered a minimum. However, since the engine has an intake restrictor, the entire plots will probably move to the left, resulting in a better for both turbochargers. The lack of proper data just makes it impossible to know for sure. Assume for an instance that the restrictor causes the volumetric efficiency to change from 85% to 60%. Following the same process as above just for the GT12, which has a real compressor map to illustrate this on, will result in the values in Table 4.2 and the plotted map in Fig. 4.12.

Point	Boost in psi	Pressure Ratio	Airflow in CFM	Airflow in lb/min
1	7	1.476	136.04	9.53
2	7	1.476	68.03	4.77
3	0	1	27.21	1.91
1	18	2.224	204.99	14.37
2	18	2.224	102.50	7.18
3	0	1	41.00	2.87

Table 4.2 Pressure ratios and airflow for compressor mapping at V_e = 0.60.

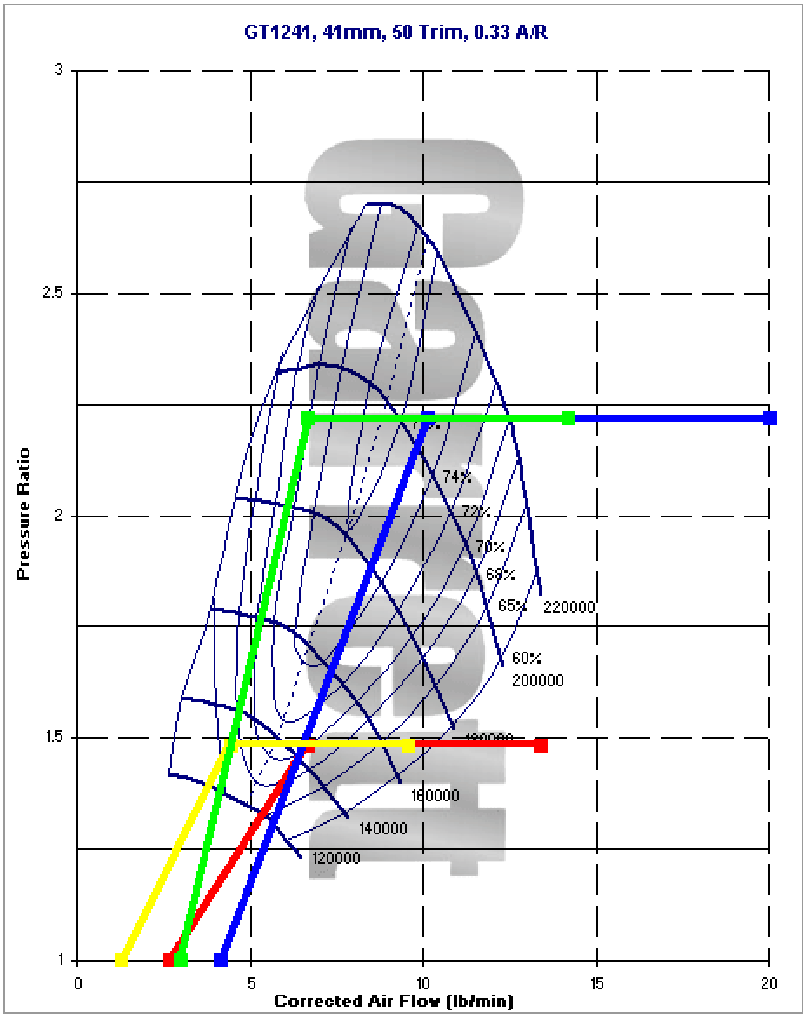

Figure 4.12 Plots of V_e = 0.85 and V_e = 0.60.

As can be seen in the above compressor map, changing the volumetric efficiency basically pushed the plotted lines to the left. Everything is still to right of the surge line as it should be, but the compressor does look like it would be more efficient, though still not perfect. The simple fact of the matter is that the restrictor could cause these lines to jump all over the place, so it is impossible to accurately map a compressor. Making matters worse, Garrett was the only turbocharger manufacturer that the team found that did in fact provide full compressor maps for turbochargers designed for small displacement engines. Unfortunately the data and choice problems mentioned at the outset of this section were never really resolved.

4.4 The Final Decision

The final decision was to go with the Garrett GT12. This was the default turbocharger, since it was in hand at the time. The team simply could not justify going out and buying any other turbocharger without experimentally obtained airflow data that could be used to properly map the compressor and show that the new turbocharger was superior to the GT12. Without this data and with few alternative options anyway, the Garrett GT12 became the best chance of getting the system operational.

The GT12 has some good features and some rather unfortunate ones. The best feature of the GT12 is that it does have a water jacket around its bearings, which is a big plus. Unfortunately it kind of goes downhill from there. The GT12 has an integral wastegate, which is better for economy systems than performance ones. Though, to its credit, it does have a separate outlet for the wastegate, which helps limit the back pressure created by the integral wastegate. This will be discussed in more detail in section

8 – Boost Control. The wastegate unfortunately limits the clocking ability of the turbocharger. The turbine and bearings sections cannot be rotated with respect to each other at all. The compressor can be rotated with respect to the turbine and bearings but only in 60° increments as a result of bolting requirements created by the integral wastegate mount. The compressor and turbine connections are not too bad, but they do have some downsides. The compressor has very unusual metric dimensions, making it next to impossible to find off-the-shelf hoses or pipes to attach to it. Garrett also does not appear to produce any gaskets or flanges for its turbine connections. Furthermore, the GT12 has the awful oil configuration listed as “BAD” in Fig. 4.7.

All in all, the GT12 is far from an ideal choice. It was in the team’s possession, though. Despite all the flaws described above, the sizing is still the primary factor in choosing a turbocharger. The GT12 is in the ballpark in terms of sizing. Without proper airflow data, it is impossible to narrow it down anymore than ballpark. Therefore, the team could not find or justify buying a turbocharger other than the GT12, making it the selected turbocharger by default. Several pictures and documentation for the GT12 are included in Appendix D.

5 – System Layout

System layout concerns the relative placement of the major components of the turbocharged system within the given vehicle. The components should be placed in a manner that is most favorable to each particular component while allowing the simplest plumbing possible. Once the working area, engine compartment or chassis, has been established, the first step is to place the turbocharger. The other major components then need to be placed. Finally, all the plumbing needs to be worked out.

5.1 – Laying the Foundation

A basic turbocharged system was introduced in section 1.4.2. Figure 5.1 shows a drawing of a basic turbocharged system that meets FSAE rules. There are clearly several major components including the air filter, throttle body, restrictor, intercooler, wastegate, and muffler connected by a series of pipes and hoses to each other and to the intake and exhaust manifolds on the engine. Finally, there is the turbocharger itself. The turbocharger is the cornerstone of a turbocharged engines system, and as such, it deserves to be placed first and foremost. The rest of these components will be placed in the system based on the location of the turbocharger. However, before the turbocharger can be placed in the engine system, the working area needs to be defined. Oftentimes it is defined as the engine compartment of an automobile. In this case, the working environment is the FSAE racecar.

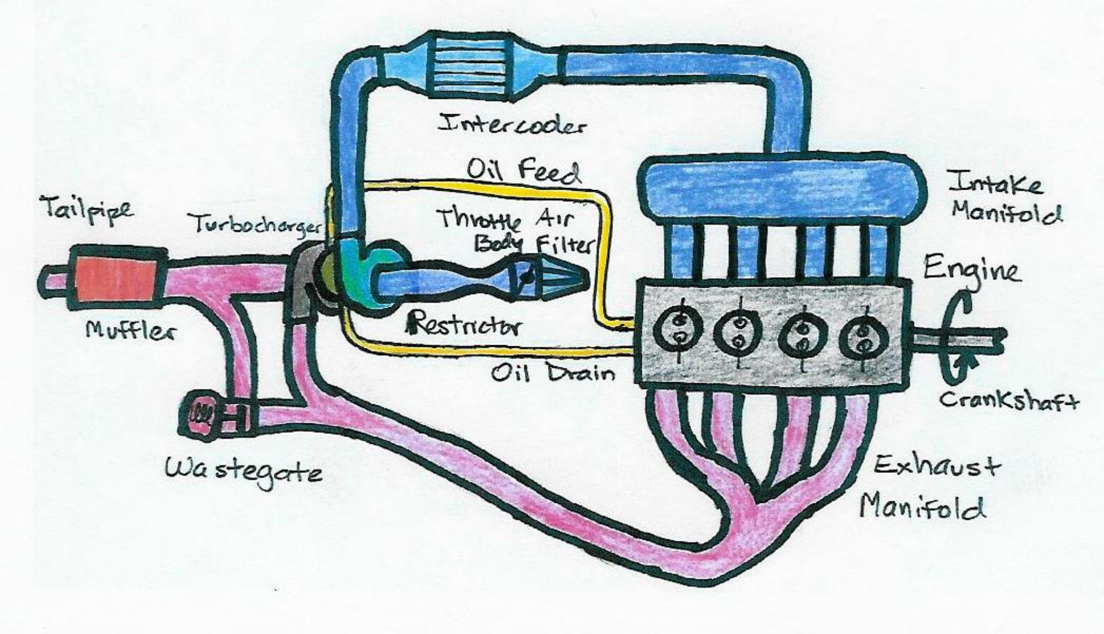

Figure 5.1 Drawing of basic FSAE turbocharged system. (Drawn by D. Curran)

The 2008 WPI FSAE team chose to use a Honda CBR600 F4i motorcycle engine in a custom designed and built chassis. The turbocharged system being designed had to fit within this chassis and its body. The actual physical FSAE chassis and body were not completed until the end of the year, but the FSAE team was able to provide a SolidWorks model of their chassis with the engine in it (Fig. 5.2) and some images of the body concept (Fig. 5.3-4). The model of the chassis and engine was used extensively in working on the system layout, particularly in working on the exact positions of the major components and the plumbing between them. The images of the body concept were used to assess the feasibility of the design concepts for system layout. While it is difficult to see from the model in Fig. 5.2, this chassis is actually pretty small and starved for space in the engine compartment where the turbocharged system needs to be housed. The body design tends to correspond to the FSAE roll envelope described in section 1.5 and shown in Fig. 1.12. This envelope placed shrunk the limited space available even more.

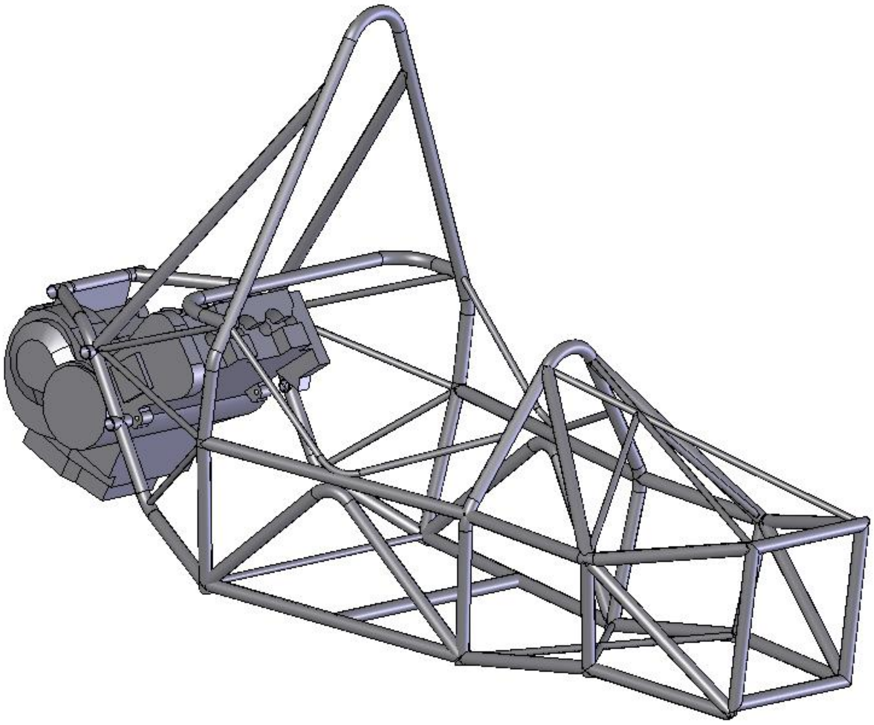

Figure 5.2 2008 WPI FSAE chassis and engine model. (Modeled by FSAE team)

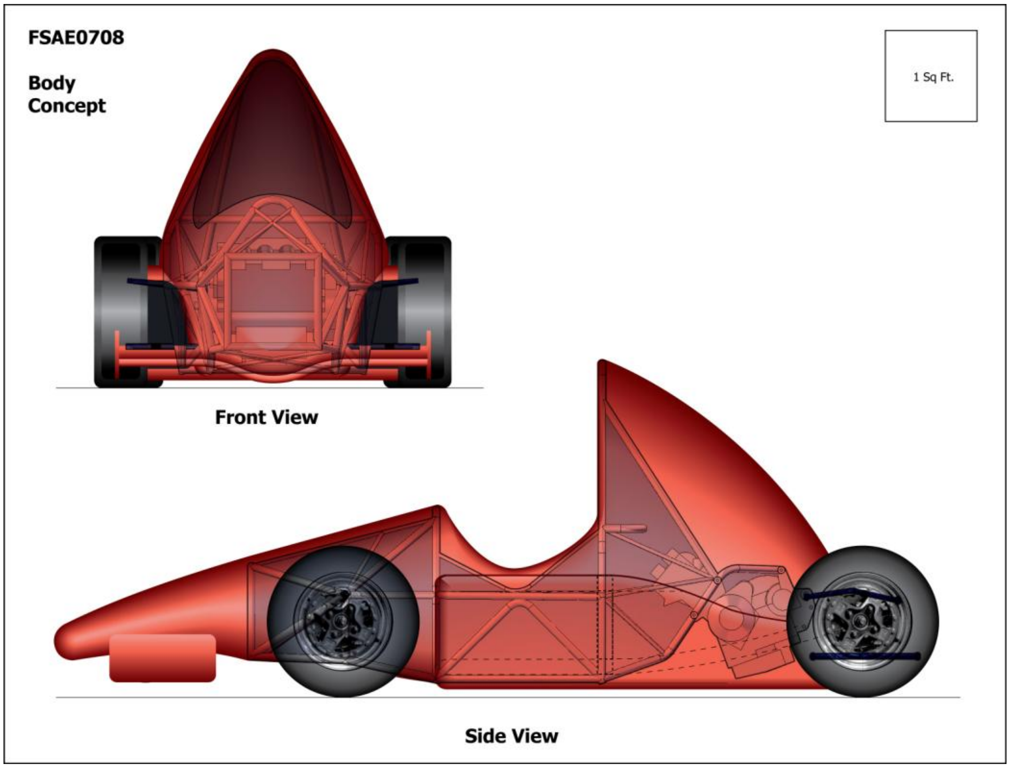

Figure 5.3 2008 WPI FSAE body concept, front and side view. (Created by FSAE team)

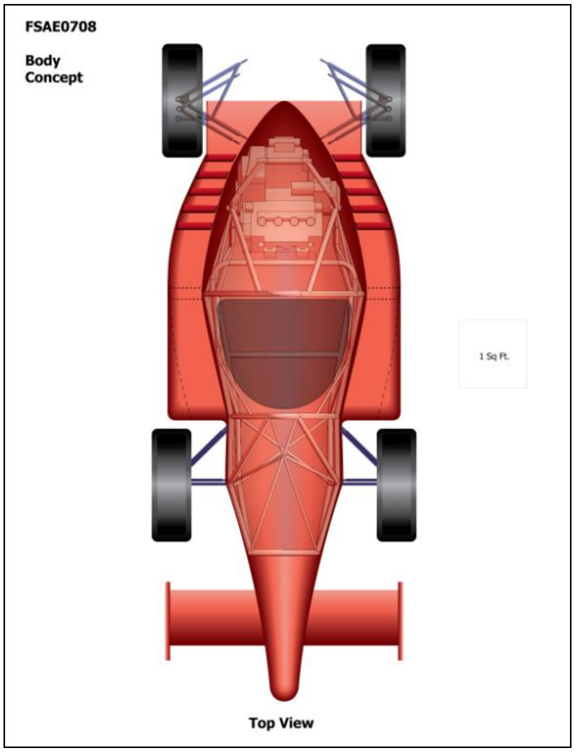

Figure 5.4 2008 WPI FSAE body concept, top view. (Created by FSAE team)

Now that the chassis and body in which the turbocharged system must fit have been defined, the actual placement of the turbocharger can be decided. Several considerations should factor into decided where to put the turbocharger. It should be placed high enough that the oil can be smoothly drained from the bearings to the oil pan using only gravity. It should provide easy access to both the intake and exhaust systems. The compressor should be positioned to allow direct plumbing from the air filter and direct plumbing to the intercooler with as few twists and turns as possible. The turbine should be positioned to allow a smooth, large radius exhaust manifold to be routed directly to it while providing room for the wastegate plumbing. Finally, the turbocharger should not be placed in a location where it is likely to come into contact with any hoses, pieces of metal, wires or anything else. Bearing these things in mind, the team studied the chassis model, as well as last year's actual chassis. After speaking with the FSAE team, the location of the turbocharger was narrowed down the two choices shown in Fig. 5.5.

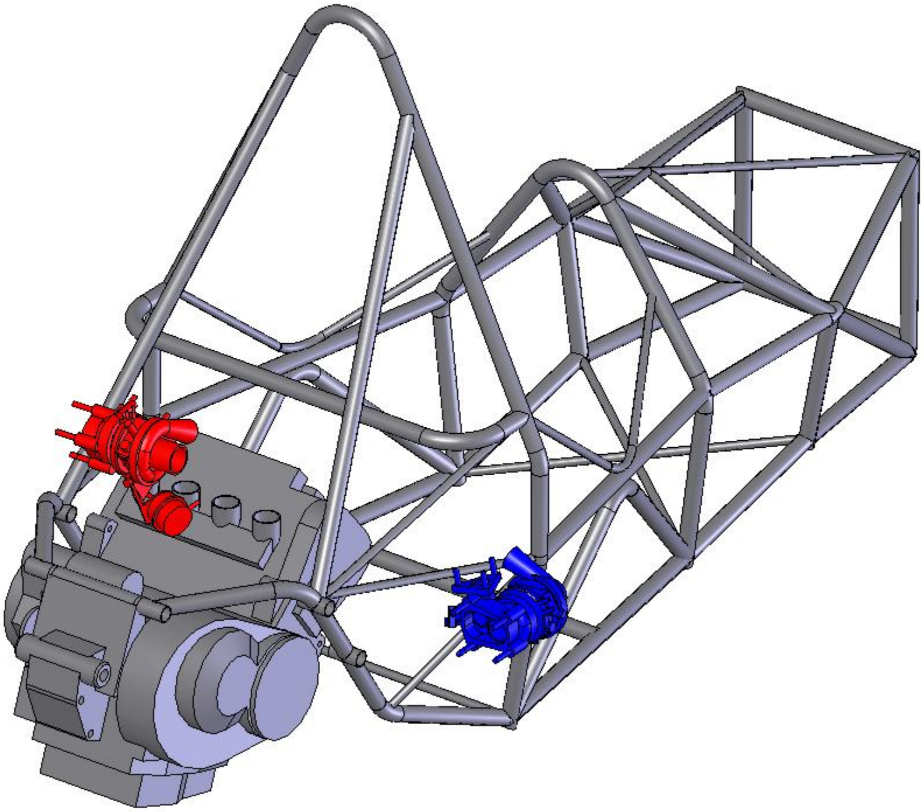

Figure 5.5 Possible locations of the turbocharger in the chassis. (Modeled by D. Curran, engine/chassis by FSAE team)

According to the FSAE team, these were really the only two places that the turbocharger would be able to fit once they finished adding all of the racecar's components. The position above the engine and behind the head, shown in red, was favored by last year's turbo MQP team. However, members of this year's FSAE team strongly recommended using the location to side of the engine, shown in blue. They said that they were going to be adding equipment to the area above engine, making it a very tight fit for the turbocharger. The side, however, was going to remain relatively open, leaving much more room to place to the turbocharger. They did ask that if the side was to be used, that it be the right side of the chassis that the turbocharger is placed on. Apparently there was an issue with the differentials that caused the car to pull a bit to one side, and placing the turbocharger on the right side of the chassis would help counteract that effect.

Based largely on the recommendations of the FSAE team, the right side of the chassis was chosen as the location for the turbocharger. Also contributing to this decision was the belief that the side of the chassis would allow for simpler plumbing and better packaging than above the engine. With the location of the turbocharger decided, it was time to start considering the other components.

5.2 – Design Concepts for the Basic Layout

The next step in the system layout process is to determine the location of the remaining major components. The precise location and exact plumbing will be left until a final general layout has been determined. Since the intake and exhaust manifolds are in fixed locations coming out of the intake and exhaust ports of the engine, respectively, they will not be considered presently. Furthermore, since the throttle body and restrictor will be attached to the air filter and do not have any special placement requirements, they will simply be considered part of the air filter as it is discussed in this section. The four components to be placed are thus the air filter, intercooler, remote wastegate and muffler. Each component has its own placement considerations that will be discussed here.

The air filter must have easy access to ambient air. It should be high enough off the ground to protect it from road debris and splashing water or mud. Ideally, it should be placed in a location that receives the coldest air possible. Finally, it should be placed as close to the compressor inlet as possible to minimize the bends and limit the overall length of the hoses or pipes connecting the two. Long pipes or hoses with many bends lead to flow losses, and longer pipes or hoses have a way of finding themselves exposed to the heat of the exhaust manifold.

The intercooler should be placed outside of the engine compartment, away from the heat of the exhaust manifold, if at all possible. It must have easy access to cool ambient air if it has an air/air core. It must be located somewhere that has enough space for a sufficiently sized core. It should be placed between the compressor and plenum to facilitate plumbing if possible. Like the air filter, it should be kept high enough to avoid any road debris or splashing. Ideally, it should be placed in a location that would permit use of either an air/air or air/water core, each of which has it own plumbing requirements that will be discussed in section *7 – Intercooler*. Finally, any air/air core needs to be placed in an area that will receive a high flow of ambient air or can be ducted to such an area.

The remote wastegate really only has one major requirement related to its plumbing. It needs to be located in place where an exhaust pipe can be routed to it from a spot after the primaries have merged. Another exhaust pipe needs to be able to be routed from the wastegate to the pipe between the turbine and muffler. One more thing to consider is the heat the wastegate is exposed to. The diaphragm of the wastegate is somewhat sensitive to heat. It is thus advisable to try to keep the diaphragm away from the exhaust manifold.

The muffler needs to be placed somewhere that has enough room to accommodate its relatively large size. It can only extend so far past the rear axle, as mandated by FSAE rules. However, it should be pushed as far back as the rules will permit to allow for the longest pipe between the turbine and the muffler as possible. This is desirable because the muffler and the merging of the wastegate pipe both create turbulence and can increase pressure in the pipes. This disturbance should be kept as far away from the turbine as possible.

With these individual component needs in mind, the overall system layout can be considered. It is highly unlikely that it will be possible to have a layout that fully accommodates each component’s needs. Determining the layout thus becomes a matter of meeting as many needs as possible while making as few compromises as are necessary. Based on this information, three different system layout concepts were developed for the turbocharged system.

The first system layout concept, shown in Fig. 5.6, routes the air filter upwards, above the driver’s head. Here it has easy access to ambient air and is high enough to be protected from the road. However, its plumbing could receive some heat from the exhaust manifold below, and the airflow must undergo a long trip and a sharp 90° bend into the compressor, which will result in flow losses. The intercooler is above the turbocharger on the right side of the chassis, just inside the body. This location is mainly intended for an air/air core but could accommodate an air/water core. It does provided easy access to ambient air with a duct, allow for simple plumbing and keep the intercooler high enough to protect it from the road. However, it is inside of the engine compartment and exposed to the heat of the exhaust manifold. It would be difficult to fit the core and ducting inside of the roll envelope, and at best, the core would be severely limited in size due to the tight fit. It would also create a weight imbalance in the car by adding even more weight to the right side. The remote wastegate is next to the engine, slightly behind and below the turbocharger. Exhaust can easily be properly routed to and from the wastegate in this position. The diaphragm is also far enough from the exhaust manifold that it should be alright. Finally, the muffler is lined up the turbine outlet and extended as far back in the car as possible. There is enough space to fit it in there without it extending to far back or shortening the pipe between it and the turbocharger.

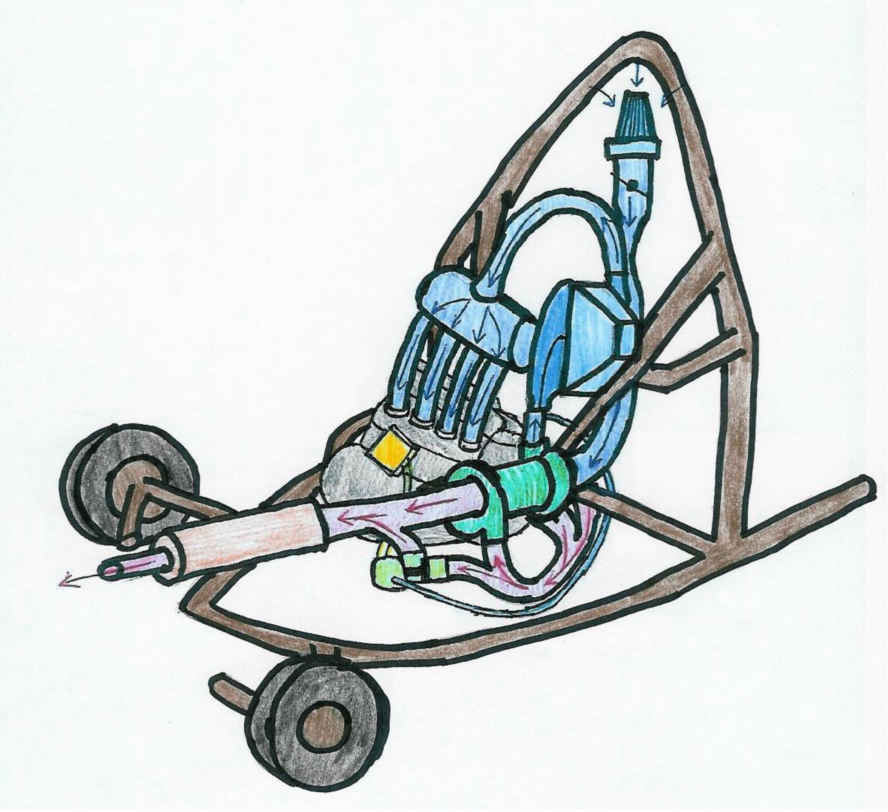

Figure 5.6 First system layout concept. (Drawn by D. Curran)

The second system layout concept, shown in Fig. 5.7, really only involves a change to the intercooler placement. Instead of placing the intercooler on the right side, above the turbocharger, the intercooler is now placed above engine head and ducted above the driver's head. In this location, it will receive more than adequate ambient airflow while having much more space for a good size core. The plumbing will also be pretty simple, and it is high enough to be protected from the road. Unlike, the first design concept though, this location will balance the weight of the intercooler evenly and will allow the entire core and duct to be fit within the chassis and body. Like the first design concept, this location is intended for an air/air core but could work for an air/water core. It also still is exposed to heat in the engine compartment. Other than the change to the intercooler, the layout is the same. The air filter, wastegate and muffler are all the same as they are in the first design concept with the same advantages and disadvantages.

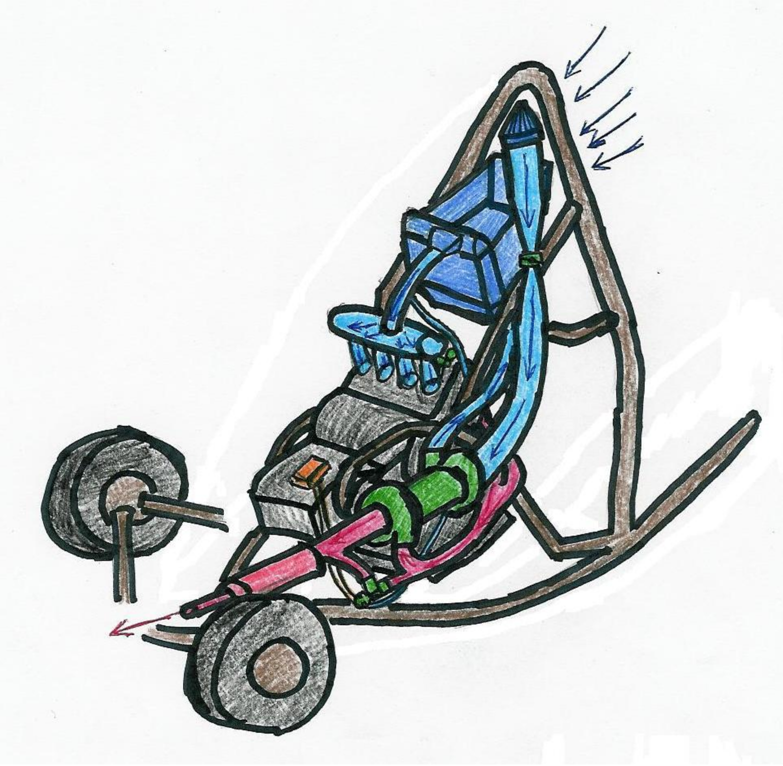

Figure 5.7 Second system layout concept. (Drawn by D. Curran)

The third and final system layout concept, shown in Fig. 5.8, is similar to the second design concept but redirects the air filter through the side pod. This allows for much more direct plumbing with less bending and traveling distance for the intake air. The air filter still receives plenty of access to ambient air without being exposed to as much heat from the exhaust manifold. The only downside is that the air filter is now much closer to the road, but it still should be high enough to remain protected. The intercooler, remote wastegate and muffler all remain as they are in the second design concept with the same advantages and disadvantages.

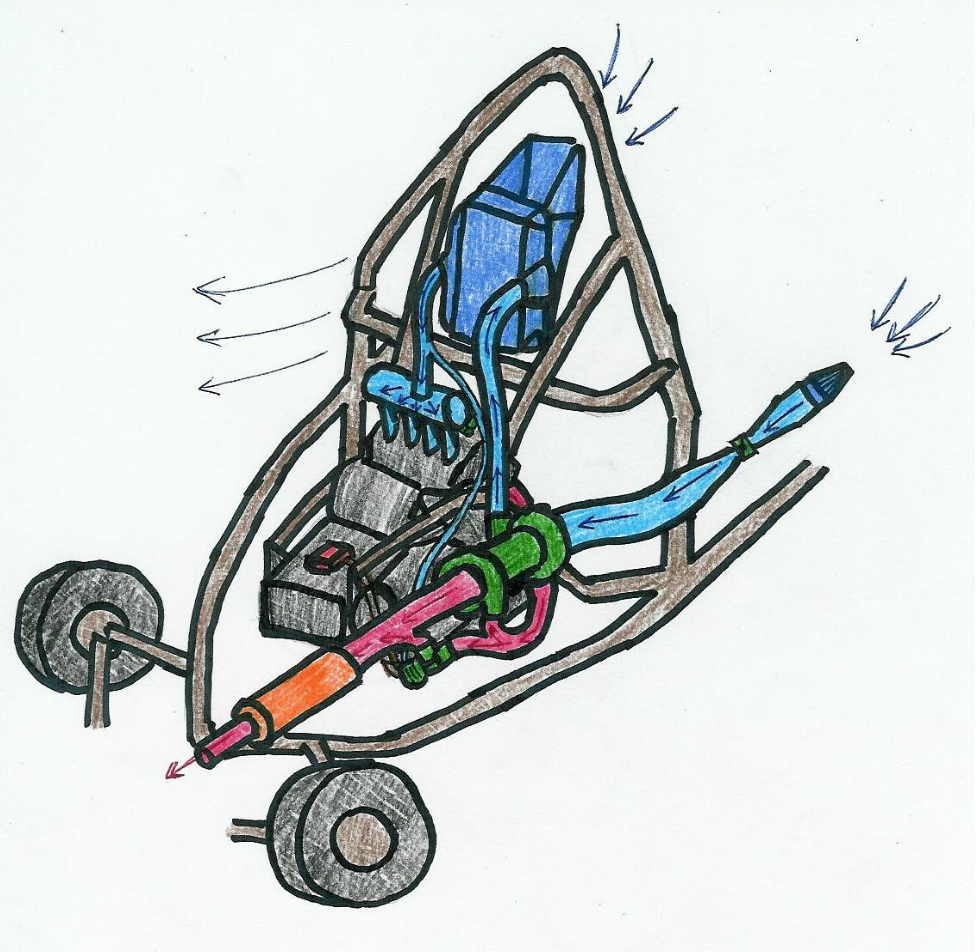

Figure 5.8 Third system layout concept. (Drawn by D. Curran)

5.3 – Selection of the Final Layout

Each of the three concepts for the system layout was analyzed in an effort to determine which one was the best. The first design concept had a terrible problem with its intercooler. It would be extremely difficult to get the intercooler into that position without violating the roll envelope. Even if the intercooler could be squeezed in there, the core size would be extremely limited, and there would be a weight imbalance created. The second design concept resolved these intercooler problems, but the first and second design concepts both had problems with the air filter. Particularly, the air filter had poor plumbing that required the intake air to make a sharp 90° bend while exposing it to heat from the exhaust. The second design concept experienced this problem to greater extent because the air filter and intercooler were both trying to access ambient air from the same spot, meaning their separate plumbing had to wrap around each other. The third design concept resolves the intercooler problems of the first concept while also resolving the air filter problems faced by both other concepts. The third concept does still have a problem with the intercooler being in the engine compartment and exposed to heat, but due to the design of the racecar and FSAE rules, this is unavoidable. Table 5.1 below shows a numerical comparison of the three concepts. Each concept was given a score for each of the design considerations of the major components. A score of “1” means that the concern is satisfactorily addressed. A score of “0.5” means that concern has been partially addressed. A score of “0” means that the concern has not been addressed. The total score for each concept is a reflection of how well it addresses all of the concerns. It can be seen from the table that Concept 3 received the highest score, meaning it satisfactorily addressed the most concerns.

	Concept 1	**Concept 2**	**Concept 3**
Air Filter	-	-	-
Ambient Air Access	1	1	1
Height	1	1	0.5
Heat Exposure	0.5	0.5	1
Plumbing	0	0	1
Intercooler	-	-	-
Heat Exposure	0	0	0
Ambient Air Access	1	1	1
Available Space	0	1	1
Plumbing	1	1	1
Height	1	1	1
Core Type Options	0.5	0.5	0.5
Weight Balance	0	1	1
Packaging	0	1	1
Wastegate	-	-	-
Plumbing	1	1	1
Heat Exposure	1	1	1
Muffler	-	-	-
Available Space	1	1	1
Packaging	1	1	1
Plumbing	1	1	1
TOTAL	**11**	**14**	**15**

Table 5.1 Numerical comparison of system layout concepts.

The team decided to go with the third system layout concept. Not only did it receive the highest score in the numerical comparison, but it also appeared to allow the simplest and most direct plumbing. All of the major system components were modeled in SolidWorks and put into an assembly with the chassis, shown in Fig. 5.9. With the major components modeled and put in their designated locations, the detailed system layout could be completed.

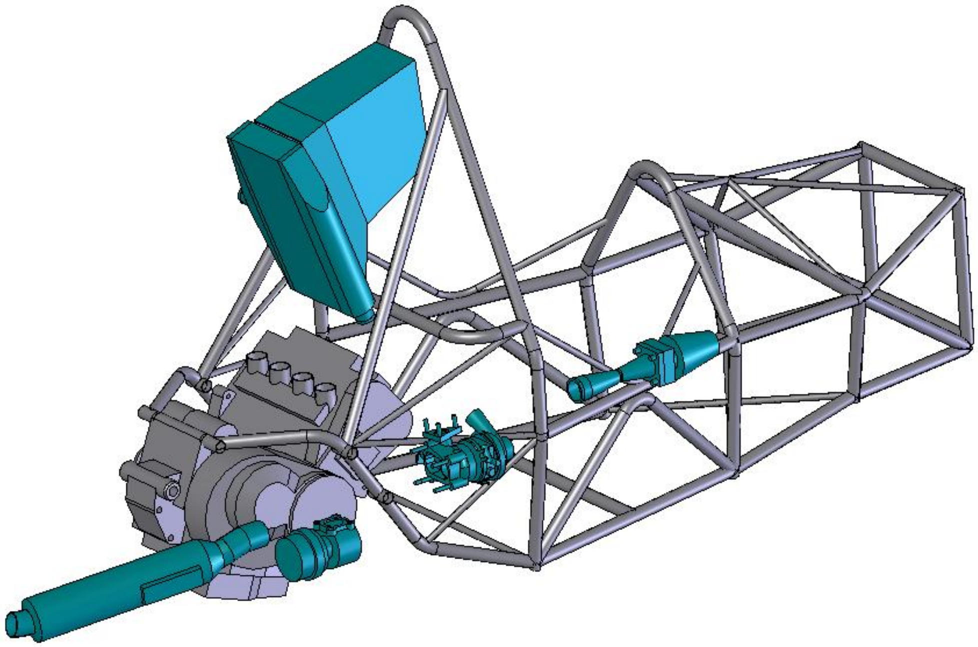

Figure 5.9 Major components in locations for final layout. (Modeled by D. Curran, engine/chassis by FSAE)

5.4 – Results of Selected Layout

The details of the selected layout are thoroughly covered in the subsequent sections, so they will not be discussed at length here. Only the results with respect to the layout itself will be discussed. Figure 5.10 shows the final SolidWorks model of the system with everything included. Appendix E contains multiple, full page views of this final model. The component placement turned out to be pretty good, but needed some slight tweaking, nothing major. The plumbing into the compressor and out of the turbine worked out really well. The intermediate plumbing, particularly the manifolds, was not as good. The intake manifold was not that bad, but the exhaust manifold was a major project due to the tight spaces available on the chassis. It might be possible to improve the exhaust manifold by moving the turbocharger towards the back of the car a bit, but this could also create additional plumbing problems and would really need to be looked into further before anything can be determined.

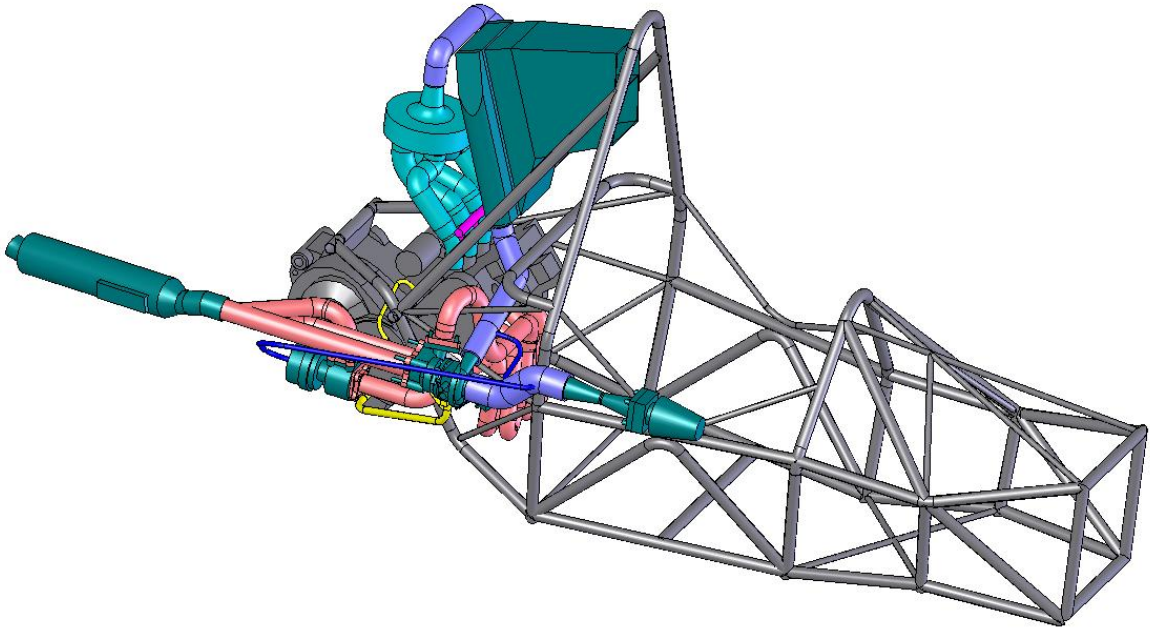

Figure 5.10 Final system model. (Modeled by D. Curran, engine/chassis by FSAE)

All in all, the chosen system layout worked out pretty well. The small size of the chassis was really the problem. This was the best layout the team could come up with for a chassis that was too small for such a system.

6 – Intake System

The intake system was introduced in section 1.4.2 and consists of an air filter, throttle body, restrictor, compressor, intercooler, plenum, runners and all intermediate plumbing. The previous section discussed the system layout. Figure 5.8 shows the system layout concept with the intake plumbing in blue. Referring back to this figure will provide a basis for what the intake system will look like. However, in order to go from the concept shown in Fig. 5.8 to the final design, which is included in Fig. 5.10, the components of the intake system must be considered both individually and collectively.

6.1 – Air Filter

The air filter is probably the simplest and most straightforward component of the intake system. The singular purpose of the air filter is to prevent anything other than air from getting into the air filter. Leaves, bugs, dirt, pebbles and dust are just a few of the things that the air filter keeps out of the engine. It would obviously not be good for the engine to have a bunch of sand get drawn into the cylinders, so the importance of the air filter is pretty clear. Many vehicles, including the Honda CBR600 F4i motorcycle, come with something called an air box. An air box is simply a compartment that contains the air filter and some engine sensors that directs intake air through the filter and into the throttle bodies. The filters used in an air box are usually a circular or rectangular shape and must be placed inside the air box to be used. Figure 6.1 shows the stock air filter used in the air box of the Honda CBR600 F4i motorcycle.

Figure 6.1 Stock air filter from air box of Honda CBR600 F4i. (Retrieved from http://kandn.com/images/l/HA-6001.jpg on April 21, 2008)

While an air box is all well and good for naturally aspirated street applications, it does not tend to work with a turbocharger or with FSAE rules. Since the air needs to be filtered before it enters the compressor, to protect the compressor as well as the engine, the air filter needs to be placed before the turbocharger, meaning it needs to be outside of the air box. Furthermore, FSAE rules require bringing the throttle body outside of the area usually occupied by the air box. The air box should thus just be scrapped in favor of a pod style air filter like the ones shown in Fig. 6.2. These types of air filters are readily available in a variety of sizes. The filter can be sized to be clamped on to a pipe coming out of the throttle body.

Figure 6.2 Pod style air filters. (Retrieved from http://kandn.com/images/l/RC-1082.jpg on April 21, 2008)

The team decided to try to find a high-flow pod style air filter for the intake system. The air filter represents a flow obstruction. A high flow filter basically minimizes this flow loss across the air filter. While such a device is certainly desirable, the flow loss across the filter is typically small enough that is it not vital. The team never found or purchased the right air filter because of an issue with the throttle body. Since air filters are so readily available, the team decided to wait until the throttle body was finished to buy one that was sized perfectly for the throttle body. The throttle body was never finished, as described as the next section, and thus an air filter was never purchased.

6.2 – Throttle Body

An engine’s throttle body is the device that regulates the amount of air entering the engine. A series of cables and linkages connect the gas pedal to a valve that opens and closes to let air into the engine. The valve generally comes in the form of a butterfly valve (Fig. 6.3) but can also be a simple sliding plate. Stepping on the gas pedal will cause the valve to open and let more air into the engine, allowing it to run faster and produce more power. The throttle body is usually placed directly above the intake ports with their being a valve for each port. Thus a four cylinder engine will have four butterfly valves on its throttle body. The FSAE rules, however, require moving the throttle body upstream of the restrictor, meaning it is going to be placed at the beginning of the intake system rather than at the end. This can lead to a few problems. The one throttle plate per cylinder approach seems to produce better flow and throttle response than the single throttle plate. Furthermore, placing the throttle body upstream of the turbocharger can cause some problems. At high engine speeds, if the throttle goes from wide open to completely closed, the turbocharger will still be spinning very fast and will create a vacuum in the intake manifold by trying to draw in air through the closed throttle body. These problems are unavoidable unfortunately due to the FSAE rules. The vacuum problem is addressed via the remote wastegate, as will be discussed in section *8 – Boost Control*.

Figure 6.3 Butterfly valve throttle body. (Retrieved from http://www.headtune.co.uk/images/Vauxhall%20Vectra%20Ecotec%202.0%20TB%20-%20White.jpg on April 21, 2008)

Rather than trying to completely design an efficient and functioning throttle body, the team tried to find one to purchase. Based on information in Corky Bell's *Maximum Boost* and recommendations from the FSAE team, the team decided to try to find a butterfly valve throttle body with a 50mm or 2 inch diameter throttle plate. Unfortunately, the budget did not permit the purchase of a new throttle body that could run several hundred dollars. Instead a much cheaper used throttle body was being sought, but no such device was found. The next plan was to just use the same throttle body design as the FSAE team. The FSAE team was custom designing a 40mm diameter barrel style throttle body. Though the diameter was a little smaller than what was desirable, it was deemed close enough that it could be used in the interest of getting the system running. A barrel throttle body, shown in Fig. 6.4, involves a rotating cylinder within another cylinder. Holes in each cylinder line up to allow airflow.

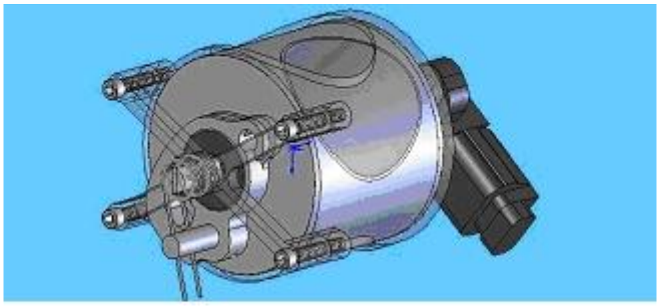

Figure 6.4 Sample barrel throttle body. (Retrieved from http://www.mne.psu.edu/me415/fall05/SAE/throttle.html on April 21, 2008)

The team thus decided to just use the FSAE design and manufacture its own throttle body. Unfortunately, the team never received the designs for the throttle body and was never able to begin machining it. Therefore, the throttle body situation was never resolved, leading to the air filter not being purchased as previously stated.

6.3 – Restrictor

FSAE rules require that a 20mm diameter restrictor be place on the intake system between the throttle body and compressor and that all air into the engine must pass through this restrictor. The restrictor severely reduces the airflow into the engine, and since the engine cannot breathe properly, its power production greatly decreases. In order to maximize the flow through the restrictor as much as possible it was designed as a converging-diverging duct with a 20mm inner diameter (ID) throat, Fig. 6.5. The inlet side would be welded to a flange that connected to the throttle body. Thus the inlet side would have a 40mm ID (1.575 inch) to match that of the intended throttle body. The outlet side was designed with a 1.65 inch ID and 1.90 inch OD. The end of the cone was extended like a regular pipe and a 2 inch outer lip was added to facilitate hose clamping. Based on the recommendations of the FSAE team, the inlet and outlet cones of the restrictor have an 8° taper leading into and out of the throat, respectively. This angle apparently yields the best flow results. Both cones would have wall thickness of 0.125 inches. More detailed drawings of the restrictor can be found in Appendix F.

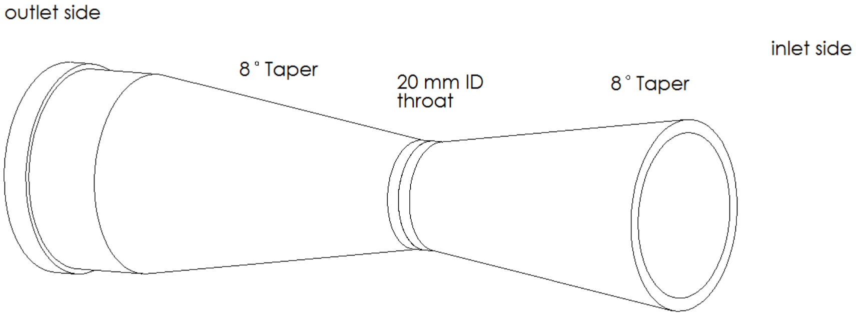

Figure 6.5 Drawing of restrictor.

In order to manufacture the restrictor, the team decided to split the restrictor into two separate cones, the inlet cone and the outlet cone. Each cone was modeled in SolidWorks and then imported into GibbsCAM to write the tool paths. Each cone would be turned down on a CNC lathe separately from 2 inch round stock. The team decided to use Aluminum 6061 for the restrictor because aluminum is a fairly lightweight and easy-to-machine metal. Neil Whitehouse showed the team how to use GibbsCAM and the CNC lathes. The team took the stock to Washburn Shops and machined the restrictor cones with the help of Neil Whitehouse. The process used to machine these and other parts is included in Appendix G. The two machined restrictor cones are shown next to each other in Fig. 6.6. The few inches of raw stock on each cone were used to clamp the part into the chuck of the lathe. The extra stock was going to be cut off when the parts were completely finished, and the two cones were going to be welded together at the throat to form the restrictor shown in Fig. 6.5.

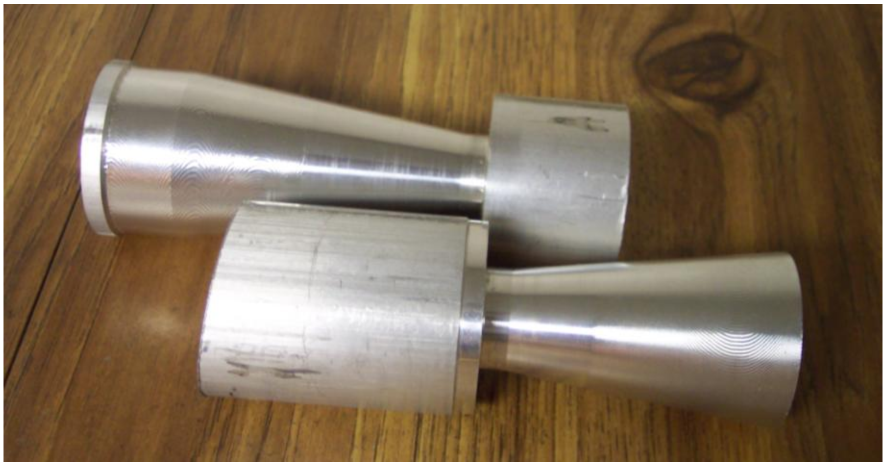

Figure 6.6 Restrictor inlet (bottom) and outlet (top) cones.

The extra raw stock was never cut off and the cones were never welded together because the restrictor outlet was never completed. Figure 6.7 shows the interior of both inlet and outlet cones. The inlet of the left has a smooth tapered interior down to its throat. The outlet on the right has an abrupt surface extending outward near the throat. The boring bar could not be extended far enough into the restrictor outlet for it to be finished.

Figure 6.7 Restrictor inlet (left) and outlet (right) cones.

The design of the restrictor was completed as was most of the machining. Only a few steps need to be taken to finish the restrictor. A smaller, longer boring bar needs to be found with appropriate tool holder to finish boring out the restrictor outlet. The raw stock needs to then be cut off of each cone. Finally, the two cones need to be welded together at their throats.

It should be noted that in addition to reducing the power output of the engine, the restrictor can create a vacuum in the intake manifold of a turbocharged system. The restrictor will become choked at 229 CFM, meaning that it is impossible to draw more than 229 CFM of air into the engine. As can be seen from the sample compressor mapping in section 4.3, the engine will be trying to consume 290 CFM of air at high engine speeds with 18psi of boost. The engine will literally be trying to draw in more air than is possible, and since the engine will be operating at a high RPM, the turbocharger will be spinning very fast. This means the turbocharger will be trying to create even more boost and draw in even more air. This ultimately leads to the actual boost pressure decreasing and a vacuum forming in the intake system, which can suck the oil through the compressor seals of the turbocharger. The team’s solution to this problem is discussed in section *8 – Boost Control*.

6.4 – Intake Manifold

The intake manifold is an assembly consisting of the plenum and the runners. After the intake air passes through the restrictor, a series of hoses or pipes guides the air through the compressor and intercooler, respectively, before delivering it to the intake manifold. Each engine cylinder will have its own individual runner which extends from the intake port to the plenum, from where it draws in air. The design of the plenum and runners is very important in optimizing airflow into the engine. The design of the plenum and runners is based largely on recommendations made in Corky Bell's *Maximum Boost* and A. Graham Bell's *Forced Induction Performance Tuning*. The overall design is also very similar to the old FSAE intake manifold.

6.4.1 – Plenum

The plenum's job is to gather air and deliver it to the runners so the cylinders can be properly filled during their intake strokes. The three most important design considerations of the plenum are thus its volume, how equally it distributes air to the runners and how smoothly it intersects with the runners. As long as these three areas receive their due attention, the shape of the plenum is not that important.

The volume of the plenum should be directly related to the engine displacement volume. However, that is where the agreement over volume ends. Corky Bell suggests a volume between 50-70% of engine displacement. A. Graham Bell recommends anything between 80-150% of engine displacement while also stating that many others believe 200-300% works very well. The team narrowed this range of 50-300% of engine displacement down to a target range of 100-150% of engine displacement. For an engine with 36.61 CID, this means a plenum with a volume of 36.61-54.92 cubic inches.

The air in the plenum must be distributed between the runners as equally as possible. A poorly designed plenum will result in one or more cylinders receiving more air than they require while one or more other cylinders receive less air than they require. Unless the engine is tuned cylinder by cylinder, this will result is some power loss and could damage the engine. If the ECU is expecting each cylinder to receive the same amount of air and is injecting the fuel according, then the cylinders receiving more air will be running lean and the cylinders receiving less air will be running rich. As always, upsetting the AFR by running lean or rich is not a good idea. Careful attention therefore must be paid to evenly distributed airflow between the runners. This can best be accomplished by having the air enter the plenum in the middle of the chamber rather than on one of its sides, as illustrated in Figure 6.8. The plenum on the left evenly distributes the air between all four cylinders much more effectively than the plenum on the right. The plenum on the right would result in the leftmost cylinder receiving less air than it should while the rightmost cylinder would receive more air than it should. Nevertheless this side entrance style plenum can be made to work if it is designed precisely enough and includes a tapered top, unlike the flat top shown in the figure. However, since the team did not have time to spend six months in a flow analysis program, it was decided to use a plenum design that more naturally distributed the air equally between the cylinders, like the center inlet plenum shown on the left in Fig. 6.8.

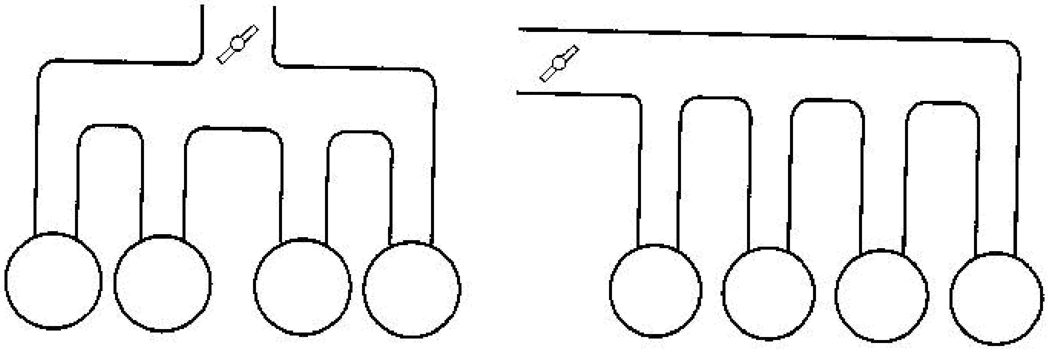

Figure 6.8 Even and uneven airflow distribution in plenums. (Retrieved from Corky Bell's *Maximum Boost*)

The final part of the plenum that deserves attention is the interface at the inlet and, particularly, the outlets. This interface should consist of a very smooth arc of a radius equal to that of inner diameter of the runner or inlet pipe. Figure 6.9 shows this relationship schematically. This smooth arc creates a bell mouth shape when seen in three dimensions. This bell mouth shape is essential in maximizing flow from the plenum into the runners and it is also beneficial to have the same bell mouth coming into the plenum.

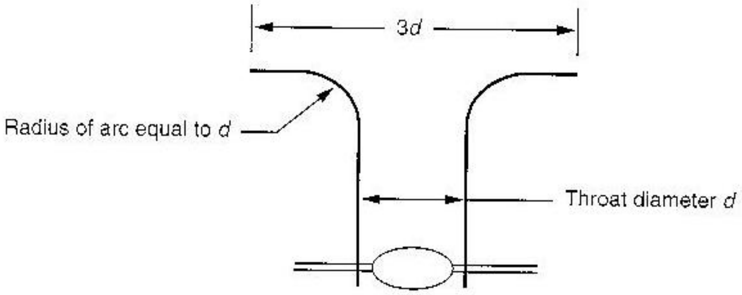

Figure 6.9 Ideal bell mouth intersection between plenum and runner. (Retrieved from Corky Bell's *Maximum Boost*)

With these three major design considerations in mind, the team began designing the plenum. Following the example of the old FSAE intake manifold, the team decided to use a right cylinder as the shape of its plenum. A few other irregular shapes were considered but none could match the simplicity and effective flow distributions of a cylinder. With the shape of the plenum fixed as a cylinder, the two variable dimensions became the diameter and height of the cylinder. These dimensions had to be chosen to produce a plenum volume between 36.61-54.92 cubic inches while allowing bell mouth entries into the runners. The runner throat diameter was 1.625 inches, the determination of which will be discussed in the next section. This means that each runner would need an arc of 1.625 inch radius, which would mean that each runner would require a circular area with a diameter of 4.875 inches. The middle two intake ports on the engine are separated 2.945 inches. The runners need to line up with these ports, so the centers of the plenum outlets could only be separated by 2.945 inches. After some trial and error, it was determined that an arc radius of only 0.5 inches could actually be implemented. Unfortunately this also meant making the inner diameter 7.25 inches. The inner height of the plenum was reduced to 1.75 inches, which seemed pretty small. These dimensions resulted in a volume of 72.24 cubic inches, which is 197% of the engine displacement volume. This is obviously much larger than the target volume, but after countless hours of working on paper and in SolidWorks, this seemed the best compromised. Even though volume was little large and the bell mouths were not quite as smooth as the ideal, the flow was distributed between the cylinders perfectly equally. The relatively small compromises in the other areas were deemed worth it to ensure this equal flow. Figure 6.10 shows the completed plenum design.

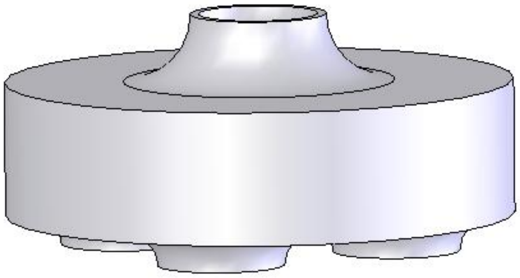

Figure 6.10 Completed plenum design. (Modeled by D. Curran)

The complete drawing for the plenum with all dimensions is included in Appendix F. Figure 6.11 below shows the side and top views of the plenum with hidden lines. The side view shows the bell mouth entries into the top and bottom of the plenum.

The top view shows how the inlet is positioned directly above the center of the outlets, providing equal flow to all runners.

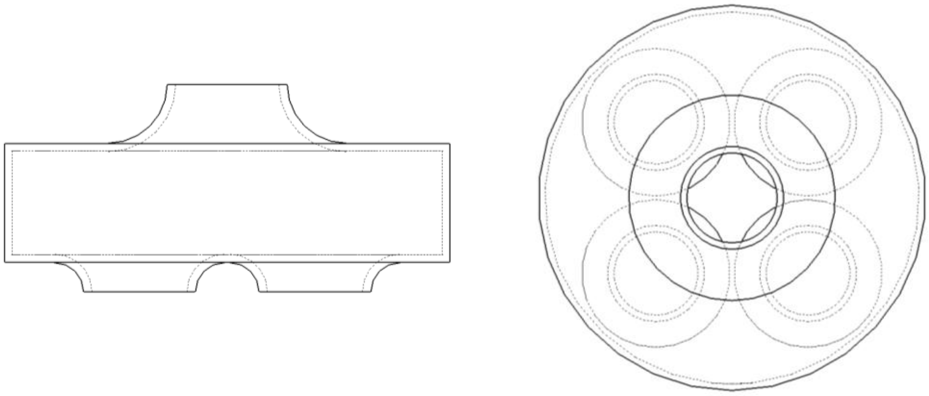

Figure 6.11 Side and top views of plenum with hidden lines. (Modeled by D. Curran)

With the plenum designed, it was time to consider manufacturing it. The plenum was to be made of 0.125 inch thick aluminum. Upon the recommendation of Neil Whitehouse, the team decided to manufacture each of the bell mouths from round stock on the lathe and then weld them to sheet metal cut to the appropriate sizes and shapes to form the top and bottom plates of the plenum. A strip of sheet metal could then be welded between the plates to form the plenum wall. The team purchased Aluminum 6061 round stock to make the inlet and the four outlets. GibbsCAM programs were written for the parts. However, there was some trouble locating the necessary tools, and eventually time ran out. Thus the plenum was not finished, but the stock for the bell mouths is available along with the programs needed to machine them.

One final thing to mention regarding the plenum is the plenum cone. At this point in the project, the team knew it was going to have to connect the top of the plenum to the intercooler somehow. The intercooler had an outlet with a 1.5 inch outer diameter hose connection. It was decided to design and build a part that could be welded to the top of the plenum bell mouth inlet and would provide a 1.5 inch outer diameter hose connection. This part was called the plenum cone, shown highlighted on the plenum in Fig. 6.12.

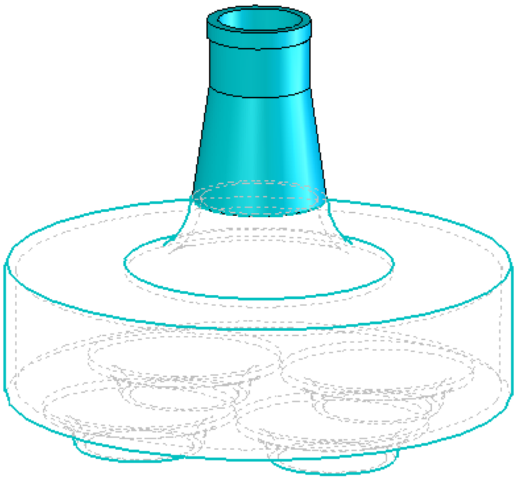

Figure 6.12 Plenum cone on the plenum. (Modeled by D. Curran)

The complete drawing for the plenum cone is in Appendix F. The plenum cone was successfully machined from a piece of Aluminum 6061 round stock using a CNC lathe. For some reason though, the wall thickness was only about a sixteenth of an inch when it was supposed to be an eighth. It was concluded that there was an error either somewhere in the GibbsCAM program or in the tool offsets. However, this wall thickness discrepancy fortunately did not represent a major problem. The completed plenum cone is shown in Fig. 6.13.

Figure 6.13 Completed plenum cone.

6.4.2 – Runners

The intake runners are the pipes that connect the plenum to the intake ports. The runners basically serve as an extension of the intake ports, so each intake port will have its own runner. The path of each runner should be as smooth and direct from the plenum to intake port as possible. The major variables for the runners are their length and inner diameter. All of the runners should be the same length and diameter, and changing these dimensions will have a strong impact on the airflow into the cylinders and thus the engine's power curve.

The length and diameter of the runners should really be looked at together. Understanding their effects on the power curve will help decide what their dimensions should be. Figure 6.14 shows a graph from A. Graham Bell's *Forced Induction Performance Tuning* that illustrates this relationship. The choice of runner diameter will basically set the RPM at which the engine achieves peak horsepower. Runners with larger diameters tend to result in peaks at a higher RPM. The length of the runner will essentially rotate the power curve around the peak point fixed by the diameter. Longer runners tend to increase low speed power at the expense of high end power. Shorter runners will do the opposite, drive up high end power while hurting low speed power. For circuit and drag racing, A. Graham Bell recommends a runner length of 10.24-11.81 inches and a runner diameter of about 1.51-1.59 inches, based on the diameter of the intake ports. In *Maximum Boost*¸ Corky Bell recommends using longer runners to provide a broad power curve at lower speeds while relying on the turbocharger to boost high end power.

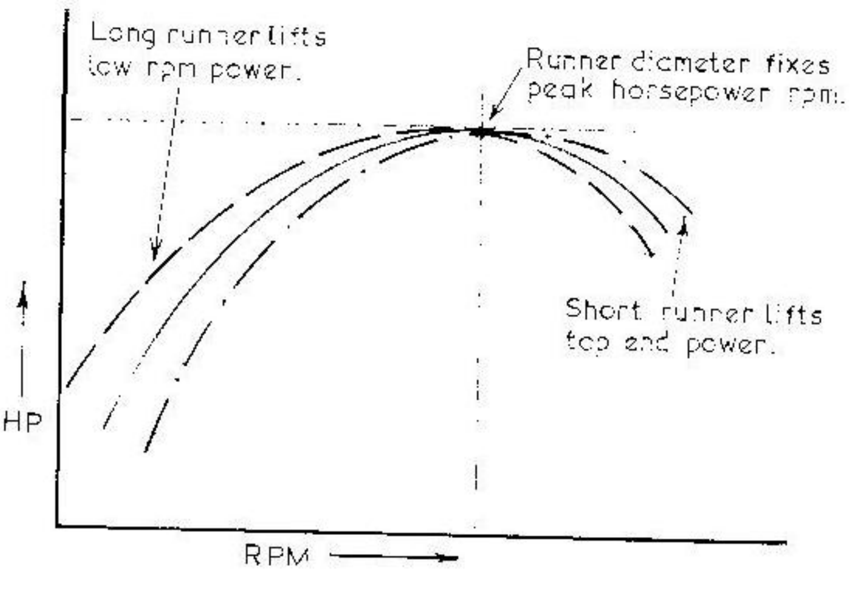

Figure 6.14 Effects of runner diameter and length on power curve. (Retrieved from A. Graham Bell's *Forced Induction Performance Tuning*)

Since the restrictor is going to absolutely kill top end power regardless of runner design, it seems like a good idea to focus on lifted the power curve broadly over low to mid range engine speeds. It would seem the best way to accomplish this would be with slightly large diameter runners and long runners. This should fix the peak horsepower at some rid range RPM before the restrictor chokes. The longer runners will then raise the power curve leading up to this point, and the turbocharger should be able to maintain good power from this point until the restrictor chokes, after which all bets are off. The diameter of the runners was chosen as 1.625 inches, slightly higher than recommended. The length of runners was nominally set to about 12 inches, also slightly higher than recommended. The diameter of the intake ports is 1.346 inches, so the runner diameter would have to be reduced before reaching the intake ports in order to allow proper clamping. Including conical reducers near the bottom of the runner has the added benefit of accelerating the air heading into the cylinders, just prior to fuel injection. Accelerating the air at this point allows the fuel to be injected into the air at its maximum velocity, promoting better mixing. It also helps completely fill the cylinder and create some turbulence in there, which helps lead to more power and better combustion. The conical reducers are one inch long segments with an 8° taper that reduce the inner diameter of the runners from 1.625 inches to 1.375 inches. These four conical reducers were successfully machined on the CNC lathe. The resulting runners are highlighted in Fig. 6.15.

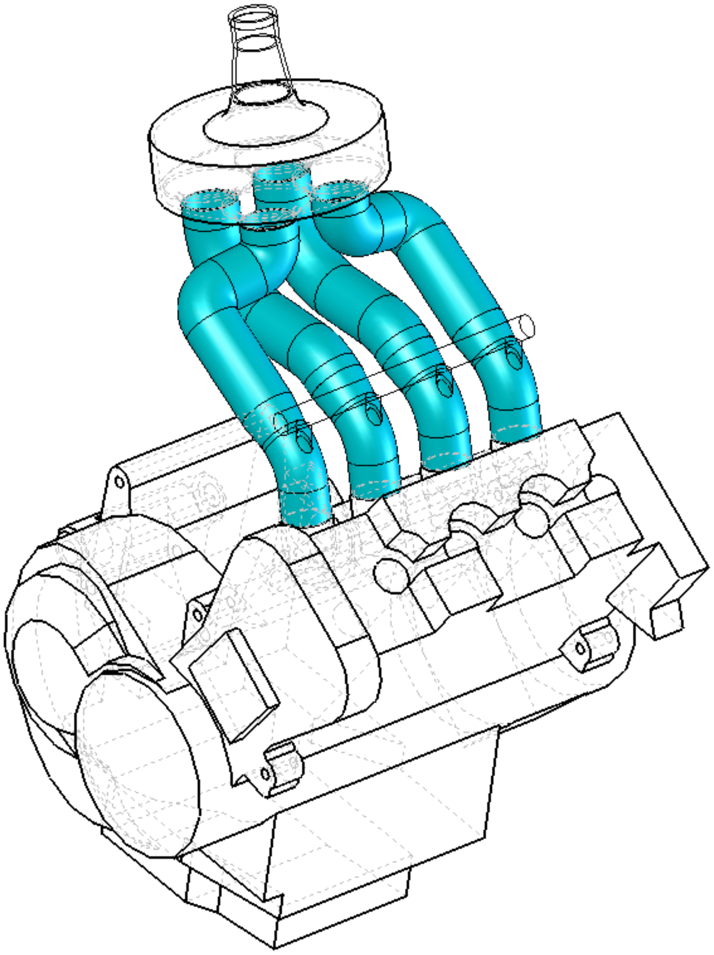

Figure 6.15 Intake runners highlighted between plenum and engine. (Modeled by D. Curran, engine by FSAE team)

The general layout for the intake manifold is based on the one used on the old FSAE manifold, but the dimensions and angles are quite different. Detailed drawings of each runner are included in Appendix F. All the runner inner diameters are as stated above. The runners are aluminum pipes with 0.125 inch wall thicknesses. The precise lengths of the inner runners are 12.21 inches, and the precise lengths of the outer runners are 11.95 inches. While the lengths of the inner and outer runners are not exactly the same, they are pretty close to each and pretty close to the nominal 12 inches.

Reviewing the drawings for the runners will reveal that there are some pretty tight and precise bends required. Therefore acquiring the necessary pipes for this manifold became a major problem. None of the locations that the team contacted in Massachusetts could do it. The team did receive a tip that there may be a mandrel in New Hampshire that could create these pipes, but the general estimate for the intake pipes was about $800. There was not enough money left to purchase these expensive pipes. The designed runners were thus not built, but a summary of the pipes that would be needed to complete the intake runners is available in Appendix H.

6.4.3 – Intake Manifold Results

The design of the intake manifold was completed and is shown in Fig. 6.16. However, implementing the design was a problem. The WPI machine shop did not have the proper tools to immediately machine the bell mouths for the plenum, and time eventually just ran out on those parts. Without those, the plenum could not be finished. The runners simply proved too expensive to purchase from the team's depleted budget, and they were thus not finished.

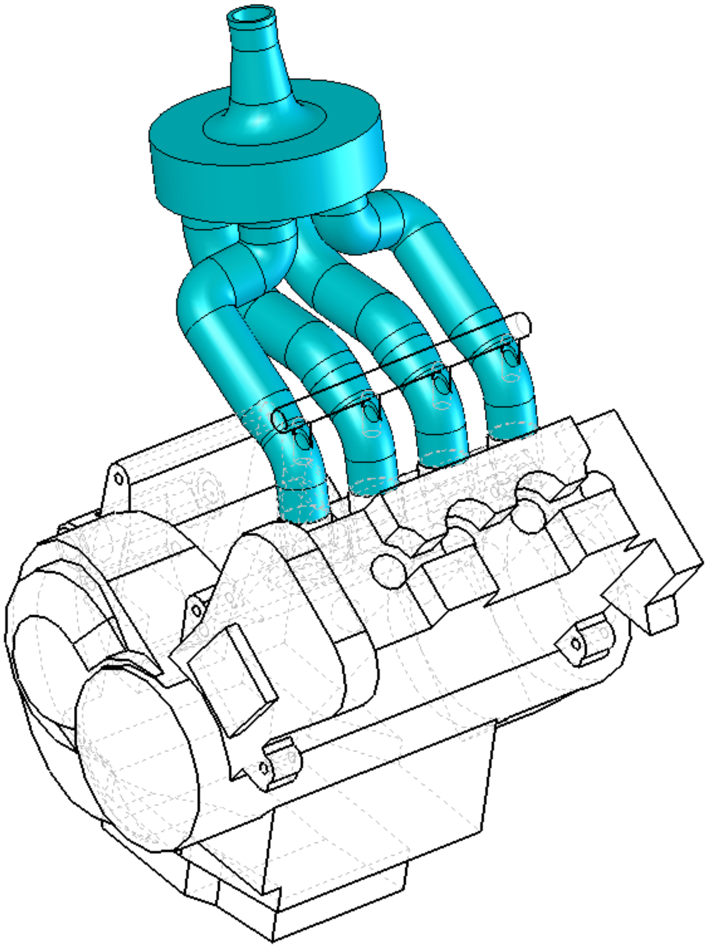

Figure 6.16 Intake manifold highlighted on engine. (Modeled by D. Curran, engine by FSAE team)

There was a late effort to redesign the intake manifold, particularly the runners, to allow the use of cheaper pre-bent pipes. However, it proved to be too little too late, and the redesign was never completed.

6.5 Exact Layout and Plumbing

The overall system layout has already been determined, but now that the intake components have all been fully defined, their exact locations can be determined. Figure 6.17 shows the exact position of the turbocharger. It has been raised high enough to allow a gravity drain for the oil. The compressor outlet and turbine inlet connections have been located to allow plumbing through the chassis beams into the interior of the engine compartment. Figure 6.18 shows the exact position of the air filter, throttle body and restrictor. This assembly has been moved forward into the side pod. It had to be raised higher than the compressor inlet because the FSAE radiator is actually right where the assembly would be if it were directly in line with the compressor. Figure 6.19 shows the exact location of the intercooler. The left image shows that the intercooler is at an angle to the roll hoop. This was done to leave enough room for the duct to fit around and over the driver's seat, which would be pretty much flush with the exterior of the duct. The right image shows how the intercooler is placed right in the middle of the chassis, evenly distributing the weight to both sides of the car. Figure 6.20 shows the placement of the intake manifold in the chassis relative to the other intake components.

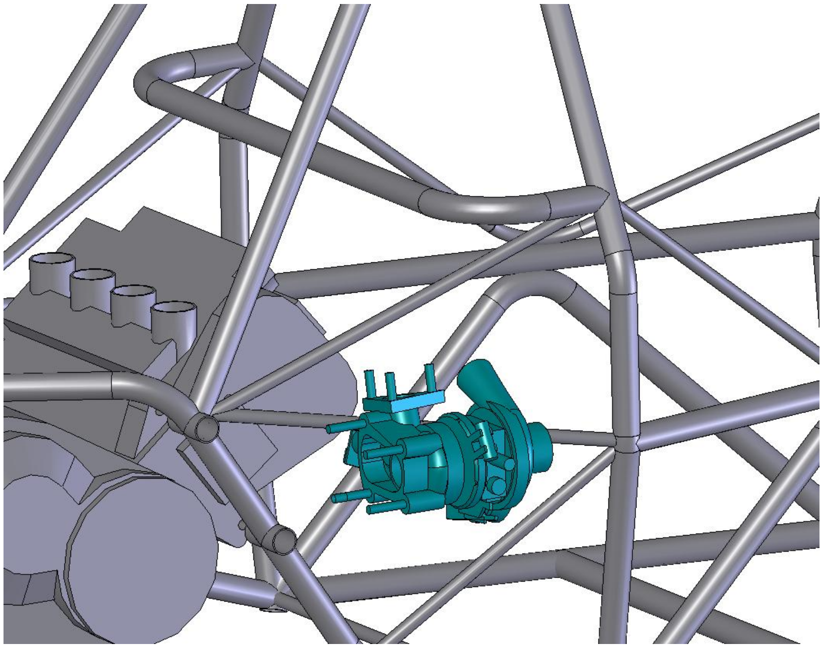

Figure 6.17 Exact location of turbocharger. (Modeled by D. Curran, engine/chassis by FSAE team)

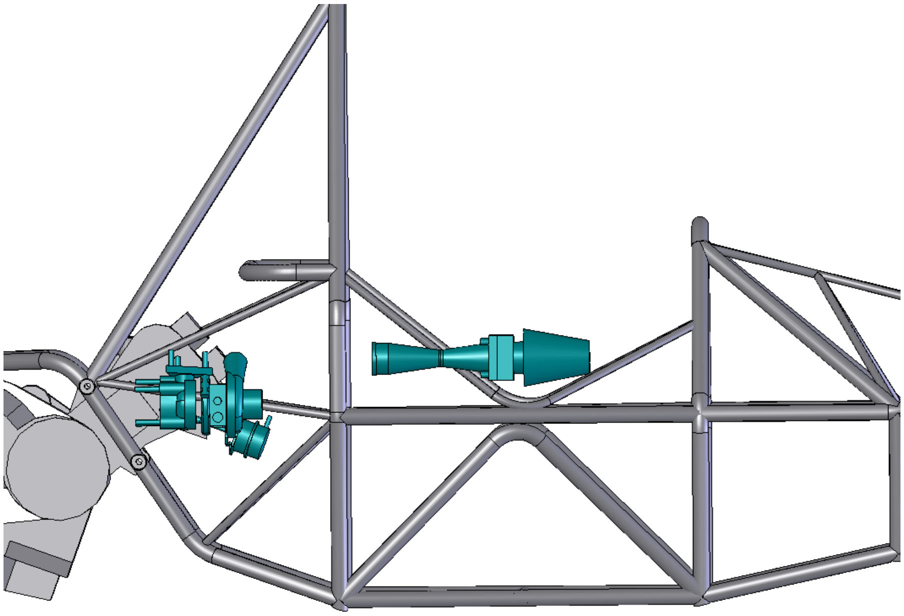

Figure 6.18 Exact locations of air filter, throttle body and restrictor. (Modeled by D. Curran, engine/chassis by FSAE team)

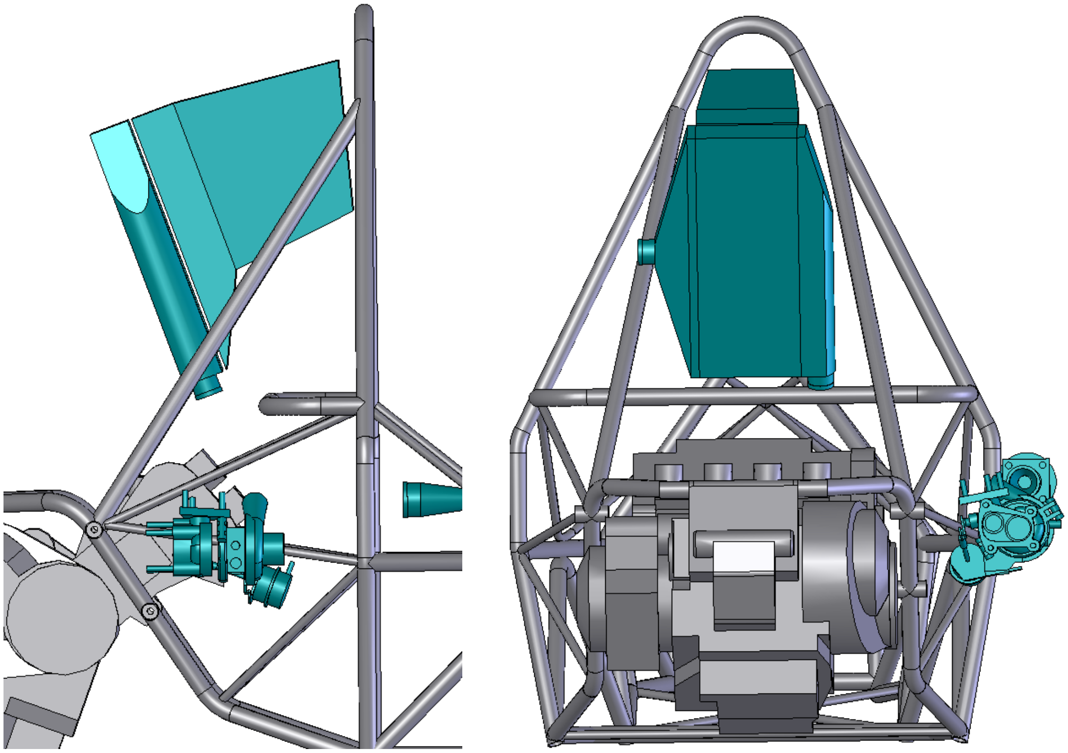

Figure 6.19 Exact location of intercooler from right and rear. (Modeled by D. Curran, engine/chassis by FSAE team)

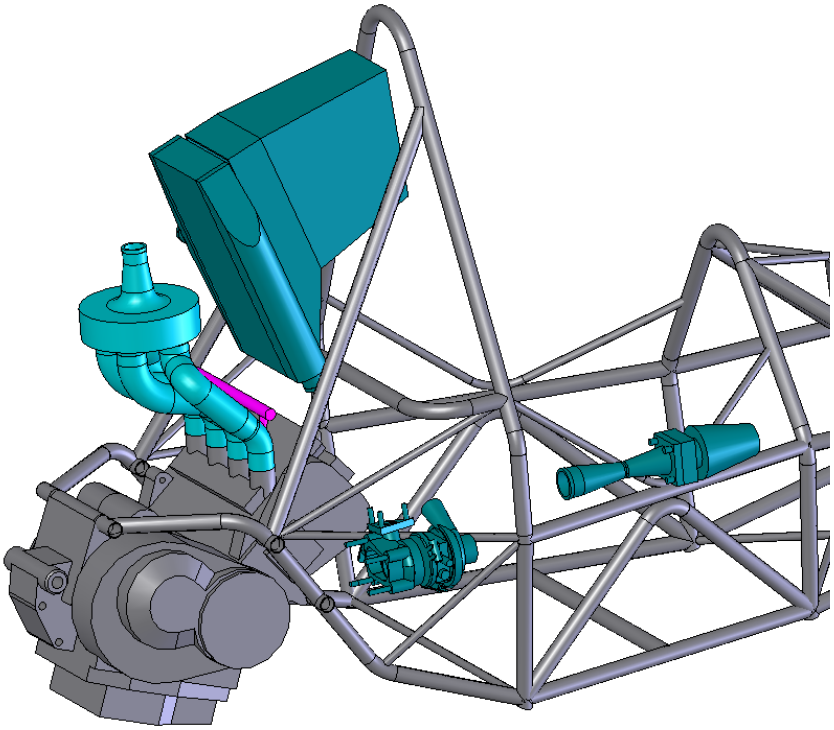

Figure 6.20 Exact location of all intake components. (Modeled by D. Curran, engine/chassis by FSAE team)

With the exact location of the major intake components determined, the intermediate plumbing was then addressed. There were two options for this plumbing, aluminum pipes or silicone hoses. The team decided to go with the silicone hoses for a few reasons. First of all, the intercooler and turbocharger were designed with hose style connections, making the use of pipes a bit more difficult. Since the plumbing required a lot of bends, it seemed like a good idea to opt for the small amount of flexibility of the silicone over the absolutely rigidity of the aluminum. Finally, the silicone hoses would add less weight to the system than the aluminum.

There are three separate connections that required silicone hoses: restrictor/compressor, compressor/intercooler and intercooler/plenum. The first set of hose connections between the compressor and restrictor required 2.0 inch ID silicone hoses. The compressor inlet actually has an outer diameter of only 1.815 inches, which is too big for 1.75 inch ID hose but too small for 2.0 inch ID hose. An aluminum sleeve, the drawing of which is in Appendix F, thus needs to be machined from 2 inch round stock. The sleeve will have an outer diameter of 2 inches and can just be slid over the compressor inlet to facilitate clamping of the hose. Two 45° bent silicone hoses will be clamped together in the middle and to the compressor and the restrictor on their ends as shown in Fig. 6.21. This arrangement allows the air filter assembly to be placed above the FSAE radiator while still providing pretty direct plumbing.

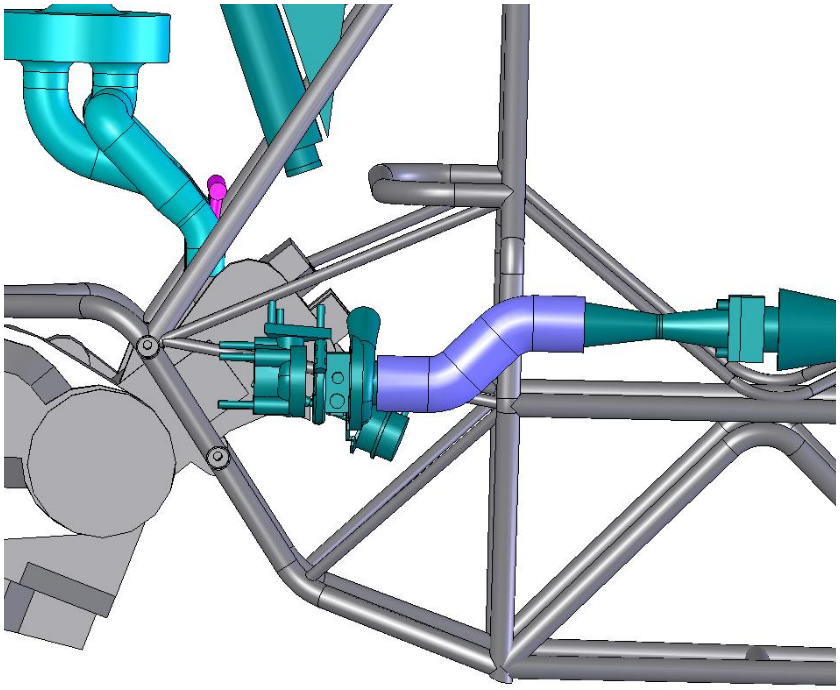

Figure 6.21 First set of silicone hoses. (Modeled by D. Curran, engine/chassis by FSAE team)

The second set of connections between the compressor and the intercooler required 1.5 inch ID silicone hoses. The compressor out actually had a diameter of 1.415 inches, which should be larger enough to permit clamping of the 1.5 inch ID hose. The intercooler inlet was designed to fit a 1.5 inch ID hose. Two 45° bent silicone hoses will be clamped together with several inches of straight hose coming off of the compressor outlet as shown in Fig. 6.22. This somewhat tight plumbing is necessary to fit connect the turbocharger and intercooler in such a small space using only 45° and/or 90° bends.

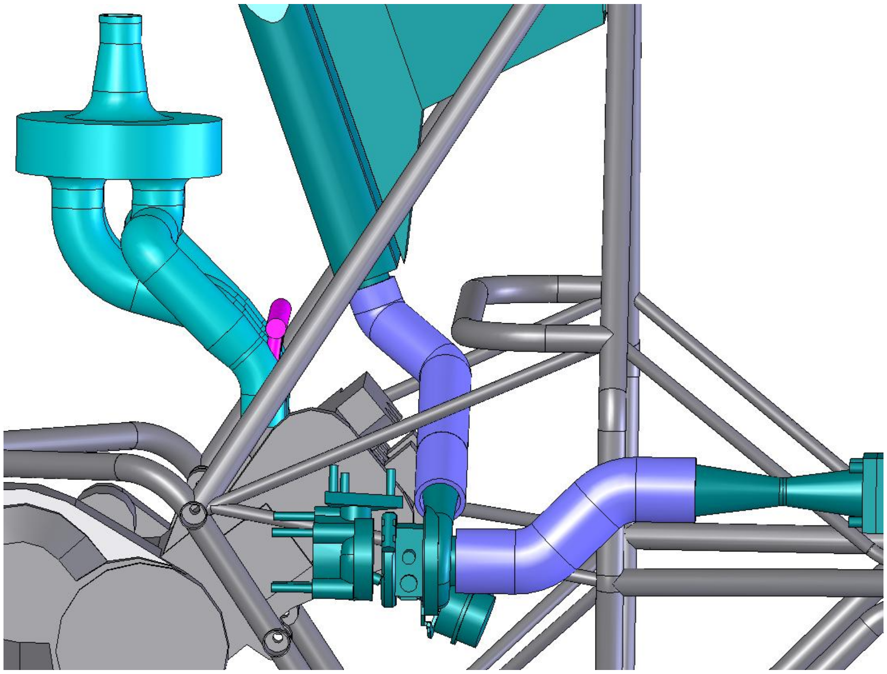

Figure 6.22 Second set of silicone hoses. (Modeled by D. Curran, engine/chassis by FSAE team)

The third and final set of connections between the intercooler and plenum cone also require 1.5 inch ID silicone hoses. Three 90° bent silicone hoses will be clamped together with some straight hose as shown in Fig. 6.23. Unfortunately, only being able to use small radius 45° and/or 90° bends made it impossible to connect these two components with a smooth large radius bent hose.

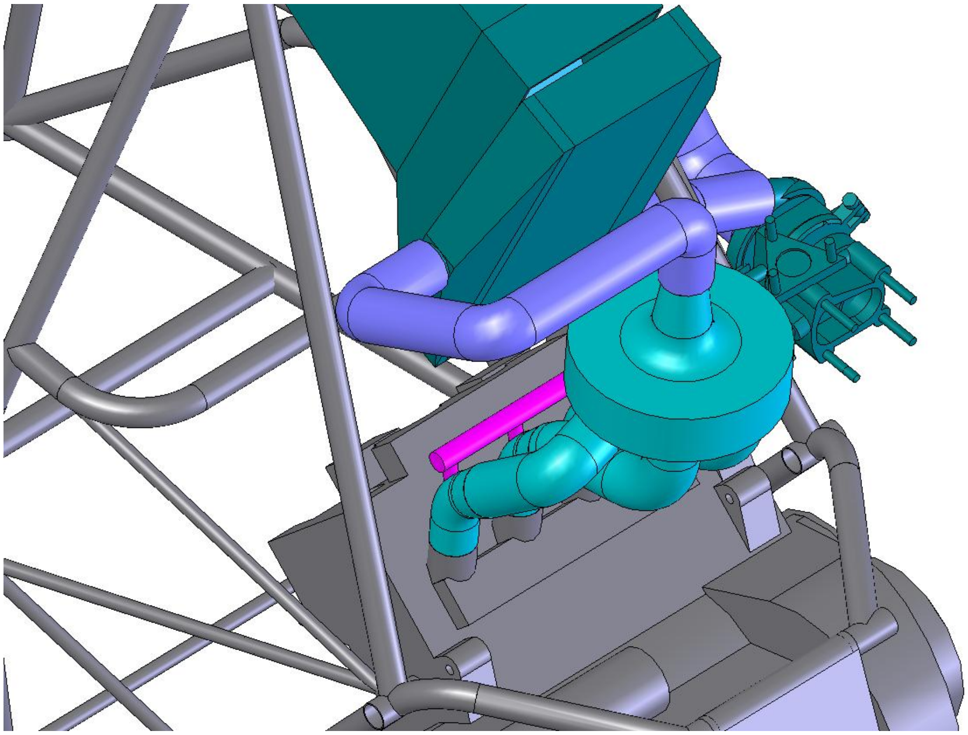

Figure 6.23 Third set of silicone hoses. (Modeled by D. Curran, engine/chassis by FSAE team)

All of the necessary silicone hoses and clamps were purchased to complete this plumbing design as detailed in Appendix B. The vendor only had a joiner for the 2.0 inch ID hoses, so the team bought a 1.5 inch OD (outer diameter) stainless steel pipe to cut into 3 inch segments to be used as joiners. Pictures of all of these components are provided in Appendix I.

6.6 – Results

The overall design of the intake system was completed. A model of the designed system, isolated from the chassis, is shown in Fig. 6.24. The turbocharger and intercooler are both in the team's possession and ready to use. The air filter was not purchased. The throttle body was not designed or manufactured, though it is possible to use the same design as the FSAE team, as was the team's intention. The restrictor inlet cone was machined and is ready to have its extra stock cut off. The restrictor outlet cone needs to be finished on the CNC lathe with the correct tools. A sleeve needs to be machined for the compressor using some of the extra 2 inch aluminum round. All of the hoses, clamps and joiners have been purchased and are ready to go. The plenum cone was successfully machine and is ready to have its extra stock cut off, as it the case with the runner cones. The plenum bell mouth inlet and outlets have been designed, and their stock has been purchased. They just need to be machined, and the plenum welded together. The runners have been designed. A pipe bender capable of making them just needs to be found, and a large amount of money will be needed to purchase them,

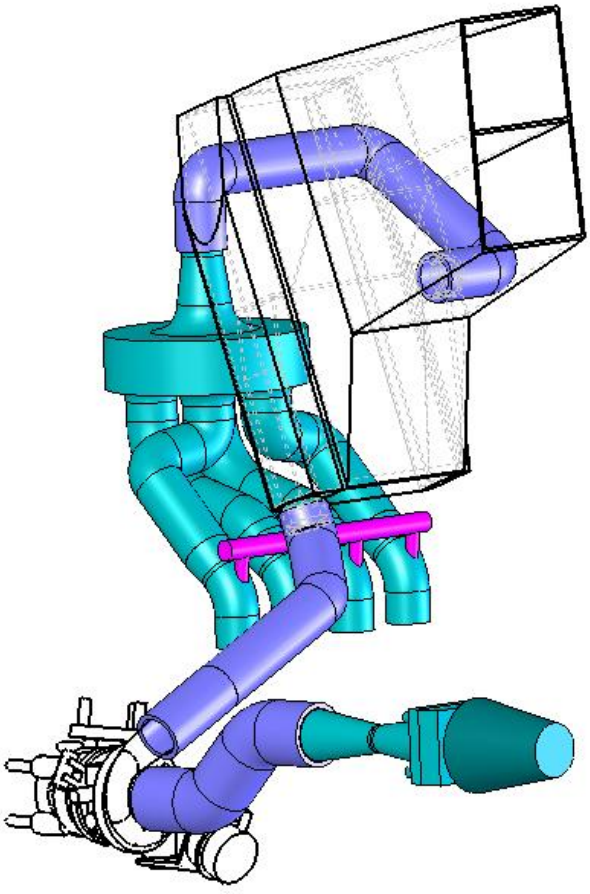

Figure 6.24 Complete intake system, isolated. (Modeled by D. Curran)

7 – Intercooler

The intercooler was introduced in section 1.4.2 as a necessary component of any turbocharged system. An intercooler is a heat exchanger that removes the unwanted heat added to the intake air by the compressor. While this is highly desirable, it is possible to have a turbocharged system without an intercooler. The team decided to investigate the benefits of an intercooler and the available options for such a device.

7.1 – The Basics of Intercooling

It has already been said but bears repeating that the intercooler is just a heat exchanger placed between the compressor and the plenum. As such, the intake air will have heat removed from it by a cooling fluid as it passes through the intercooler, but it will also experience a pressure lose across the intercooler. Choosing an intercooler is then a matter of balancing heat reduction against boost pressure losses. Sometimes, this means choosing no intercooler if the system is designed to handle the anticipated heat but needs to produce as much boost into the engine as possible. That being said, the teams decided very early on that an intercooler was a necessary component. It really takes firsthand knowledge and extensive planning to design a system that will function well without an intercooler. Since the team did not have any firsthand experience, it decided to go the safer route of using an intercooler. Generally speaking, a well chosen intercooler will do a lot more good than harm, which can be seen by looking at the potential risks and rewards.

The compressor is going to be sending very hot, high pressure air into towards the engine. The high temperature of the intake care had two big drawbacks. The first is that it has a tendency to lead to detonation, commonly known as knocking, in the engine. Section 1.1 talked about how the spark plug will ignite the air/fuel mixture at the top of the cylinder during the power stroke, causing a flame to move smoothly through the mixture igniting the rest of mixture and forcing the piston downwards. Detonation occurs when the air/fuel mixture ignites ahead of the flame. This uncontrolled combustion can cause serious damage to the engine, which is a precisely controlled machine that is supposed to work in harmony. Detonation destroys this harmony. Higher intake air temperatures always increase the likelihood of detonation. The second big drawback of hot intake air is a decrease in power output. Because of the constant AFR of the engine, more air molecules entering the engine means more fuel gets injected into the engine, resulting in more power output. It is thus desirable to get as many air molecules into the engine as possible, hence the compressor increasing the pressure of the intake air. Unfortunately, the heat added by the compressor causes the density of the air to decrease, meaning fewer air molecules per cubic inch. This density loss turns into a power loss. Basically, higher temperature means lower density which means less power.

An intercooler would lower the temperature of the intake air, eliminating these two problems. The lower temperatures will make detonation much less likely, though excessive boost and high compression ratios can still cause detonation without hot intake air. Lower temperatures also mean that the intake air density will be higher, meaning more air molecules per cubic inch. This means that there will be more air molecules in the cylinder, leading to more fuel in the cylinder and higher engine power outputs. The benefits of an intercooler are clear, no detonation and more power.

There is a downside to adding an intercooler, though. The intake air will experience a pressure loss as it goes through the intercooler. In order to foster heat transfer with the cooling fluid, the air must basically rub up against pieces of metal called turbulators. The turbulators disrupt the airflow through the intercooler core and put additional drag forces on the air, both of which contribute to the pressure loss across the intercooler. The denser the turbulators, the more heat transfer there is but the greater the flow loss exists. The reverse is also true that less dense turbulators result in less heat exchange but do not cause as much flow loss. The pressure loss basically equates to a decrease in boost pressure, meaning less engine power. The cost of an intercooler is thus that the boost pressure decreases before the air reaches the engine.

If the core of the intercooler is large enough for the anticipated airflow into the engine, the pressure losses can be kept to a minimum. According to Corky Bell in *Maximum Boost,* a good intercooler can increase air density 10-15% while staving off detonation for another 4-5psi of boost and causing pressure loss of less than 2psi. This suggests that a well chosen intercooler will result in a net power increase while also reducing the risk of detonation. The team thus decided that the benefits outweighed the costs and that an intercooler should be included. However, one additional downside that team had to consider was the potential weight of intercooler.

7.2 – Core Type

The core of the intercooler is the actual heat exchanger. Two fluids enter the core perpendicular to each other (cross flow) and travel through a series of channels with turbulators in them. Heat then travels through the core material as the temperatures of the two fluids get closer and closer to each other. One of fluids entering the core will be the hot intake air from the compressor. The other fluid entering the core will be a colder fluid, either ambient air or water from a radiator system. While the detailed design and internal workings of intercooler cores do vary and constitute different “types of cores,” this phrase when used here refers to kind of cooling fluid used by the core. The two types of cores are thus those that use ambient air to cool the intake air and those that use water in a radiator system to cool the intake air. The internal workings of the core are left to the experts and not discussed here.

An intercooler that uses ambient air to cool the intake air is known as an air/air intercooler. An air/air intercooling system only requires the addition of a single component, the actual intercooler, between the turbocharger and engine, shown in Fig. 7.1. Charge air (hot intake air) enters through the inlet side end tank and travels through the core to the outlet side end tank. Ambient air from the surrounding environment flows in the perpendicular direction to cool the charge air. This relatively simple system is very reliable and requires little maintenance. The single component is not too expensive and adds little weight to the car. It will have good efficiency at high speeds but can suffer a bit at low speeds. Another downside to air/air intercoolers is that the core has to be much bigger than that required for cores that use water, which does a better job removing heat than air. Since the air does not transfer heat as well as the water, the heat transfer area needs to be larger for equal heat transfer rates, meaning larger core sizes.

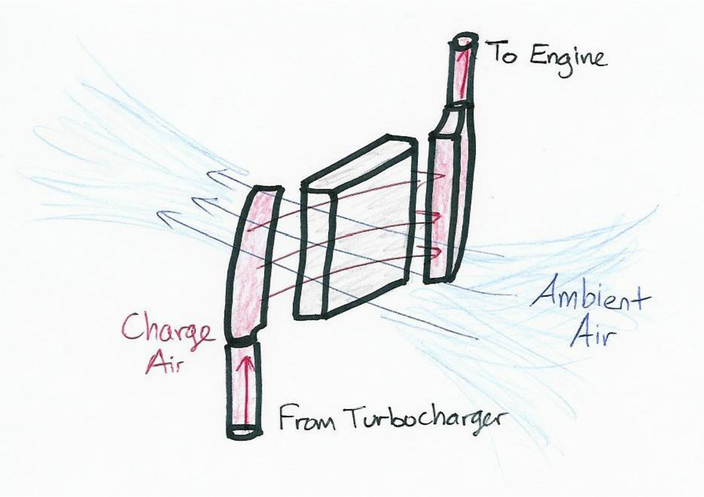

Figure 7.1 Simple schematic of an air/air intercooler. (Drawn by D. Curran)

The other types of cores that use water to cool the charge air are known as air/water intercoolers. Air/water intercoolers have better low speed efficiency and require smaller cores than their air/air counterparts. They also tend to result in smaller pressure losses. All this makes an air/water intercooler sound far superior to an air/air, but all these benefits come at a cost. An air/water intercooler requires the addition of an entire system to the car, Fig. 7.2. A reservoir, dedicated radiator and water pump must be installed along with several gallons of water. This complicated system ends up being fairly expensive while adding a lot of weight to the car. While the size of the core is smaller than that of an air/air intercooler, the extra components and plumbing ends up taking up a decent amount of space. Furthermore, the air/water intercooler loses its efficiency edge at higher speeds. Thus the environment and application of the turbocharged system must be considered before selecting a core type.

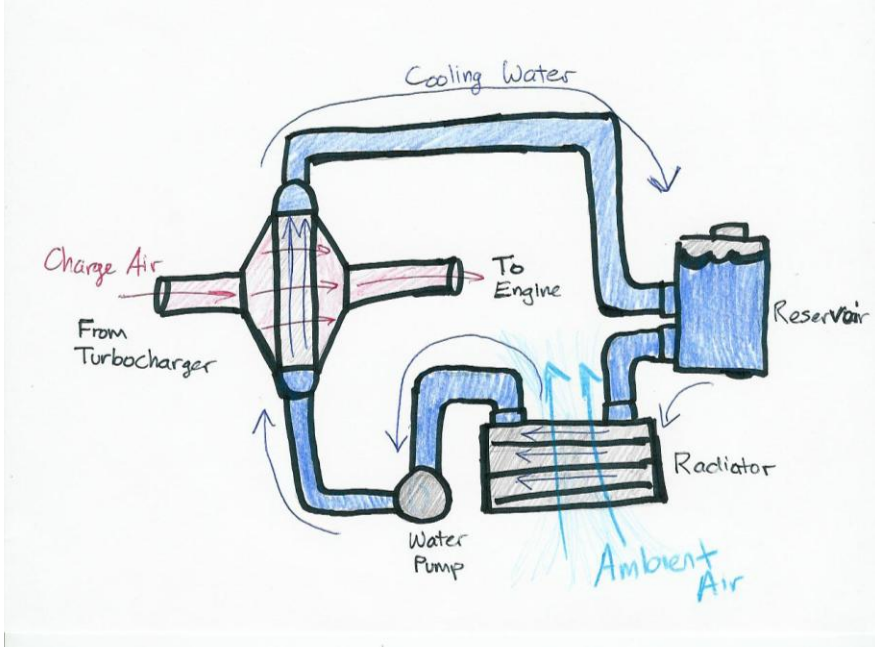

Figure 7.2 Schematic of an air/water intercooler system. (Drawn by D. Curran)

In the case of the system being designed for the FSAE car, there is large enough space for a good size intercooler core in the location selected for the intercooler. This space has the added benefit of having easy access to ambient air. The racecar will be spending much of its time operating at high speeds. The weight of the system needs to be limited as much as possible. There is a limited budget to purchase components with. Finally, simplicity is always appealing. All of these factors tend to favor an air/air intercooler, and the team thus decided to use an air/air intercooler.

7.3 – Core Sizing

With the core type selected, the proper size of the core could be determined. The process used to size the core was taken directly out of Corky Bell's *Maximum Boost*. Figure 7.3 shows the graph used to begin the core sizing process. Using the maximum flow rate of 229 CFM, the necessary internal flow area was determined to be 13 square inches. The area of the charge face, the rectangular face that the intake air enters, was determined by dividing the internal flow area of 13 square inches by 0.45. This is done because only approximately 45% of the charge face is open to the airflow. This yields a charge face with a required area of about 29 square inches. The width of the charge face is actually the thickness of the core. Smaller core thicknesses tend to result in better efficiency, so the team chose to use a core thickness of 2 inches. This meant that in order to have a charge face with an area of at least 29 square inches, the height of the charge face would have to be at least 14.5 inches. The resulting charge air face has dimensions of 2x14.5 inches. The only remaining unknown dimension of the core is on the cooling air face. The height of the cooling air face is the same as that of the charge air face, 14.5 inches. The width of the cooling air face was what needed to be determined. Generally speaking larger areas for the cooling face provided better heat transfer (since there is more heat transfer area), but the width should not be increased too much as it tends to result in higher flow losses. The team decided to go with a width of 8 inches to provide a large cooling surface area while keeping the flow losses within reason. The resulting core, shown in Fig. 7.4, had dimensions of 14.5x8x2 inches.

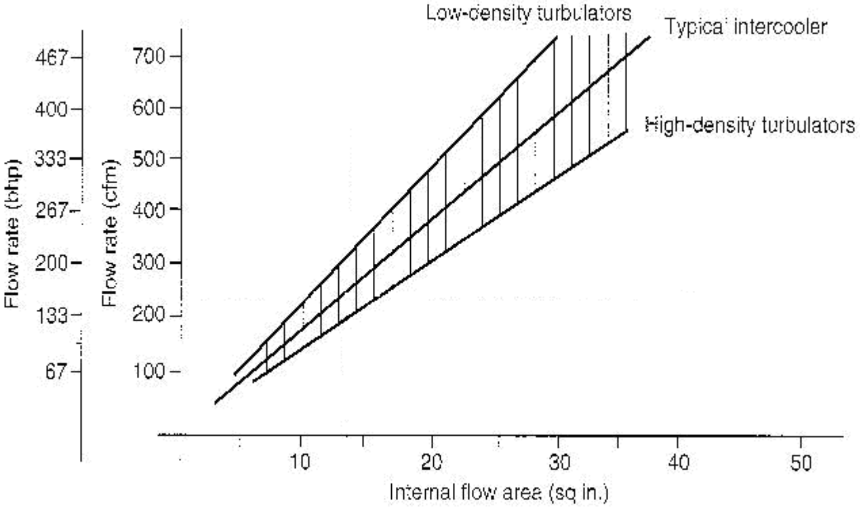

Figure 7.3 Intercooler core sizing graph. (Retrieved from Corky Bell's *Maximum Boost*)

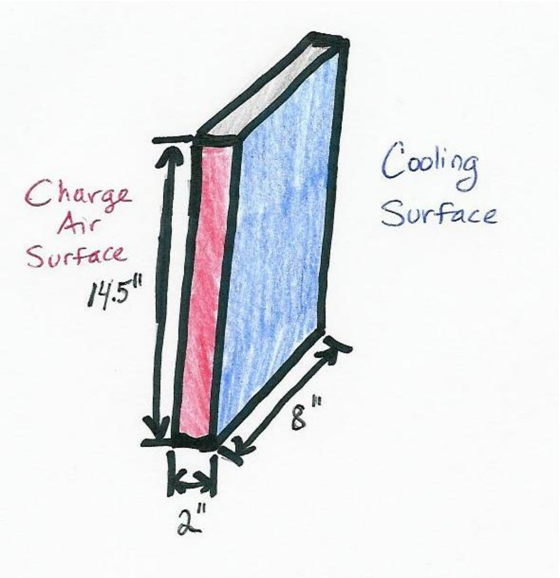

Figure 7.4 Necessary dimensions are core. (Drawn by D. Curran)

With the necessary dimensions of the core known, a basic core modeled in SolidWorks and put into the FSAE chassis to confirm that sufficient space was available. Fortunately, there was enough space for the core with plenty of extra room for end tanks. It was thus decided to try to obtain an intercooler with these core dimensions. After designing the end tanks, discussed in the next section, the team began looking for available intercoolers. Bell Intercoolers was located online. They offered a variety of core sizes and would custom build end tanks. The closest available size to the calculated 14.5x8x2 inches core was actually 15.1x6x2.25 inches. The new core size resulted in a larger charge face area but a smaller cooling face area, but the values were relatively close to the originals.

7.4 – End Tanks

The end tanks on either side of the intercooler are responsible for guiding the charge air into and out of the intercooler core. Well designed end tanks can lead to better efficiency and lower flow losses whereas poorly designed end tanks can lead to the opposite. The inlet side end tank should smoothly and evenly distribute the incoming air throughout the charge air face of the core. Keeping the flow smooth reduces the flow losses while evenly distributing the air will utilize the entire core rather than just concentrating the flow through one small area, which will help improved efficiency. The outlet side end tank should then smoothly gather all of the charge air coming out of the core into a single pipe. If the outlet end tank is designed correctly, there will minimal flow losses, but a poorly designed outlet side end tank will lead to large flow losses, reducing the boost pressure of the air leaving the intercooler. Several good and bad end tank designs are shown in Fig. 7.5.

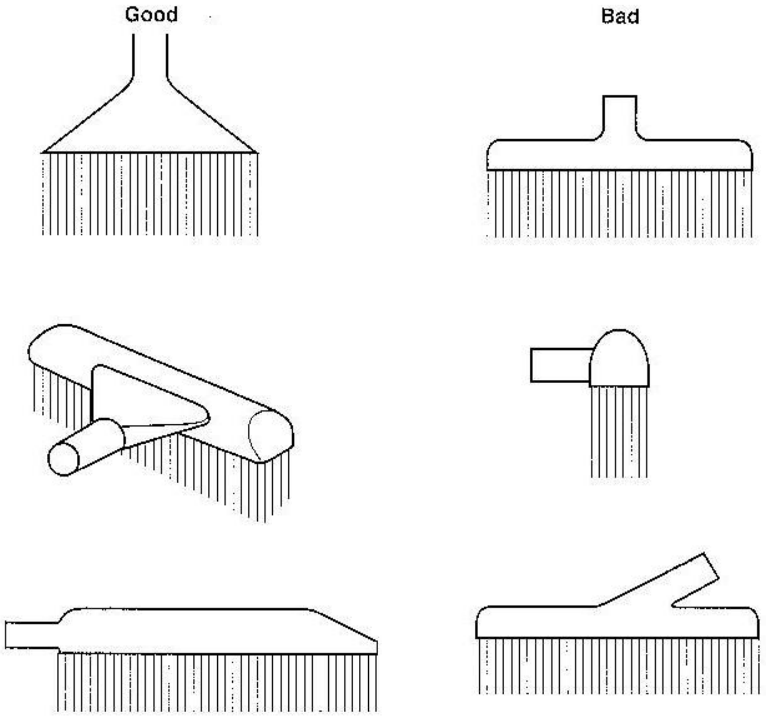

Figure 7.5 Good and bad intercooler end tanks. (Retrieved from Corky Bell's *Maximum Boost*)

The team decided to design their own end tanks based on the chosen core dimensions and the above guidelines to end tank design. In addition to the aforementioned design considerations of the end tanks, the plumbing requirements created by the end tank inlet and outlet were of serious concern. Since available space for plumbing was very limited and the only hoses available to work with were straight, 45° bends and 90° bends, the end tanks had to be designed specifically to facilitate plumbing within this system. With the location of the intercooler already fixed, this was not too difficult. The inlet side end tank was chosen such that its inlet was at the bottom of the intercooler, the goal being to allow a relatively simple and smooth 90° hose to connect it to the compressor outlet. The general shape of this end tank is similar to the bottom left end tank in Fig. 7.5. An internal baffle was added to this end tank to ensure a more even distribution of airflow throughout the core, see Fig. 7.6. The outlet side end tank was chosen such that the outlet would be a bit higher than the plenum cone and oriented to allow a U-shaped hose connection between the two. The general shape of the outside side end tank is similar to top left end tank in Fig. 7.5.

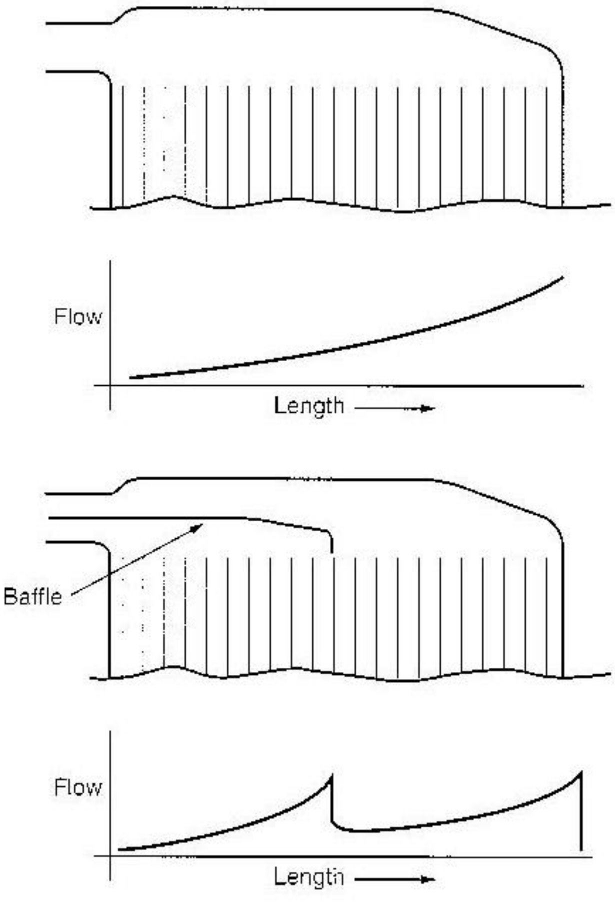

Figure 7.6 Effects on an internal baffle on intercooler flow. (Retrieved from Corky Bell's *Maximum Boost*)

Detailed technical drawings of the intercooler end tanks are included in Appendix J. Figure 7.7 shows a model of the designed intercooler. The end tanks are shown at 50%

transparency so the internal baffle is visible. As was stated in the previous section, Bell Intercoolers was contacted to manufacture the intercooler. The end tank drawings were faxed to them, and they agreed to custom manufacture the intercooler assembly.

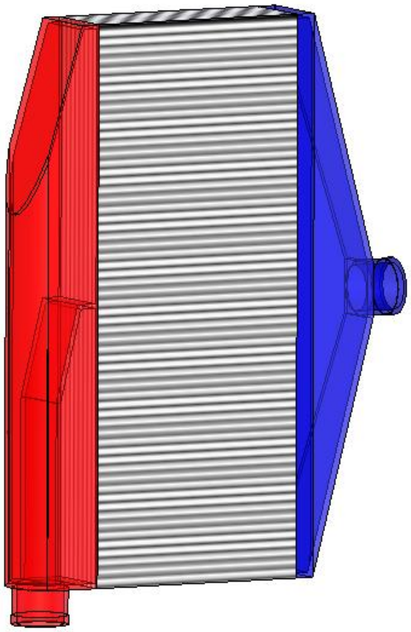

Figure 7.7 Selected core with custom end tanks. (Modeled by D. Curran)

7.5 – Ducting

Ducting is a method of increasing the ambient airflow through the intercooler core, thus improving heat transfer. In cases where the core will receive a significant amount of ambient airflow as a result of its location, a duct really is not necessary but can still lead to an improvement in efficiency. Therefore, it is generally a good idea to include a duct on the intercooler. Figure 7.8 shows how to properly duct an intercooler. As can be seen in the figure, no duct or a widening duct will allow a lot of air to simply go around the intercooler core. The air is going to go wherever the path of least resistance leads, and the core represents a pretty significant resistance. The air really needs to be forced to go through the core to maximize flow and heat transfer, as is the case with the narrowing duct. The duct inlet area should be a product of the cooling surface area of the core, at least 25%.

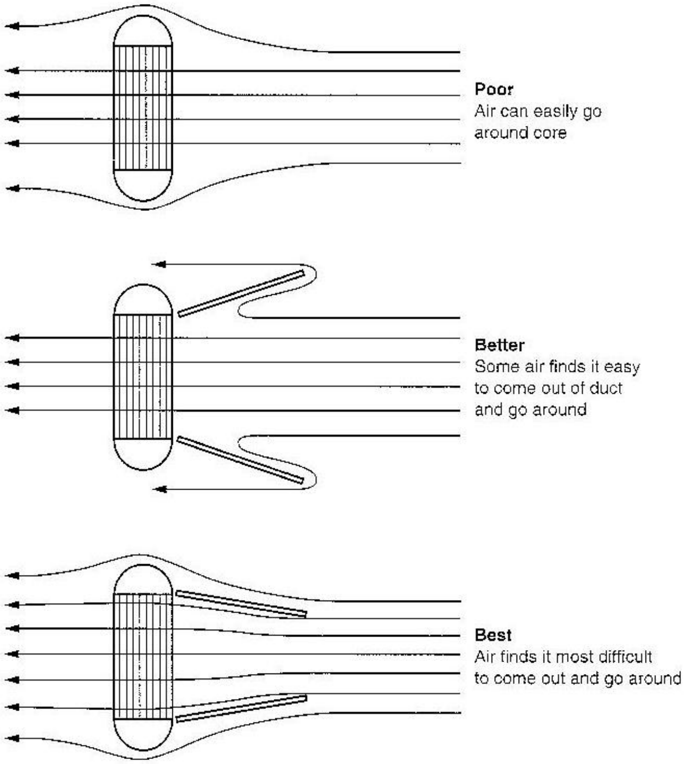

Figure 7.8 Correct way to effectively duct an intercooler. (Retrieved from Corky Bell's *Maximum Boost*)

The team decided to include a duct on the intercooler to ensure the best possible heat transfer between the ambient air and intake air. Furthermore, there was a concern that the top of the driver's seat could obstruct airflow to the intercooler core. Figure 7.9 shows how the duct is designed to deliver air to bottom of the core, blocked by the driver's seat. Figure 7.10 shows that the duct itself has an internal baffle to ensure that airflow does actually make to the lower portions of the intercooler. The duct would be constructed of aluminum sheet metal and attached to the intercooler via a rubber strip. The actual detailed design of the duct was never completed because the concept was never matched up the actual chassis and body. The duct was deemed less important than efforts to get the system operational. The finalized duct would likely have a smaller inlet area since the actual area above the driver's seat is less than the estimate in the model. Furthermore, the shape of the duct inlet would likely be changed to fit snugly against the inside of the roll hoop.

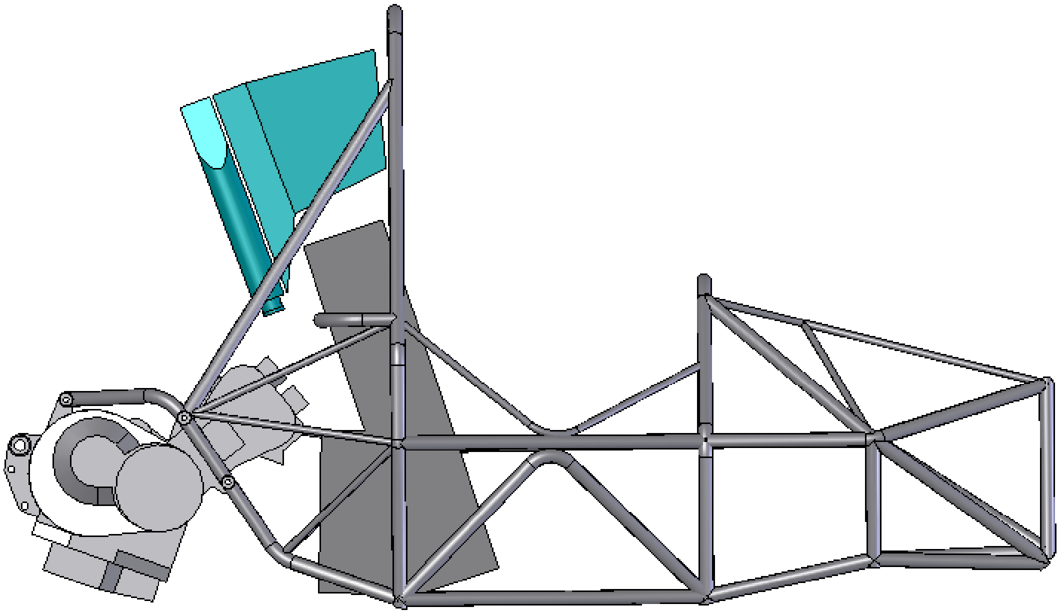

Figure 7.9 Ducted intercooler in chassis. (Modeled by D. Curran, engine/chassis by FSAE team)

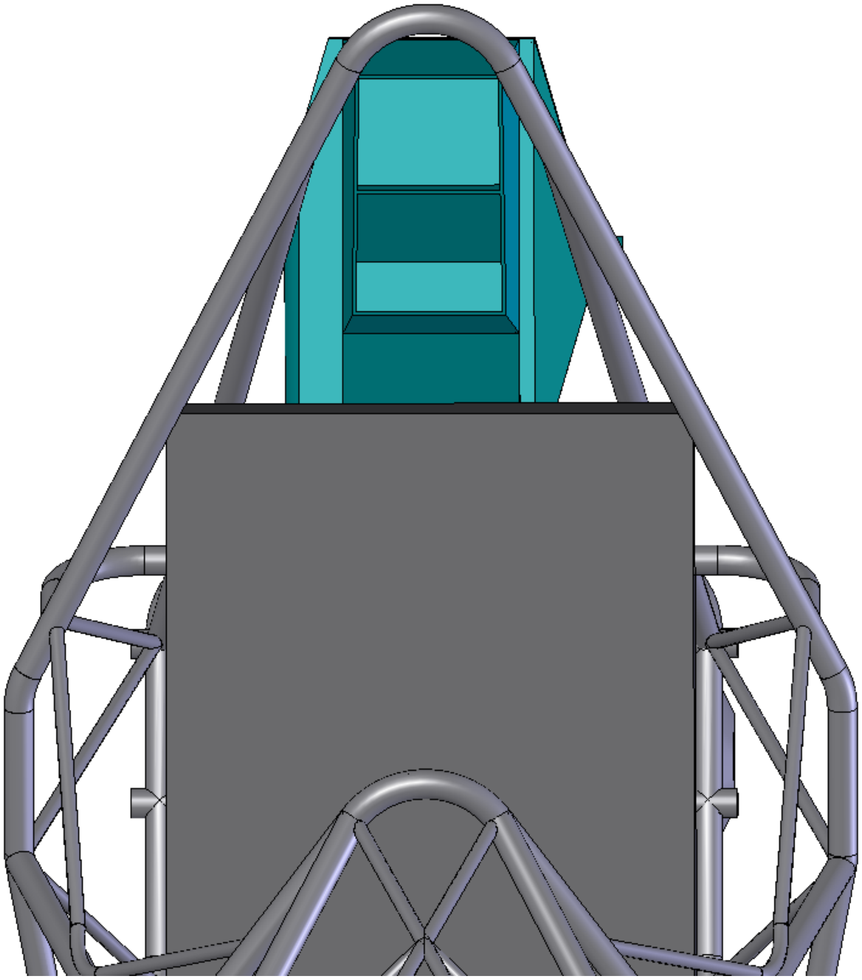

Figure 7.10 Internal baffling of the intercooler duct. (Modeled by D. Curran, engine/chassis by FSAE team)

7.6 – Results

The intercooler assembly was completely designed. An air/air core was designed to efficiently remove the heat added by the compressor while limiting the pressure loss across the intercooler. The two end tanks were designed to allow the smoothest possible flow while evenly distributed the air throughout the intercooler. Drawings of the core and end tanks were sent to Bell Intercoolers who proceeded to manufacture the designed intercooler. The completed intercooler is shown in Fig. 7.11. Appendix J contains detailed drawings of the intercooler components and several pictures of the finished product. Preliminary ducting design was completed but needs to be finalized for the chassis and body. The duct should then be constructed and attached to the intercooler assembly.

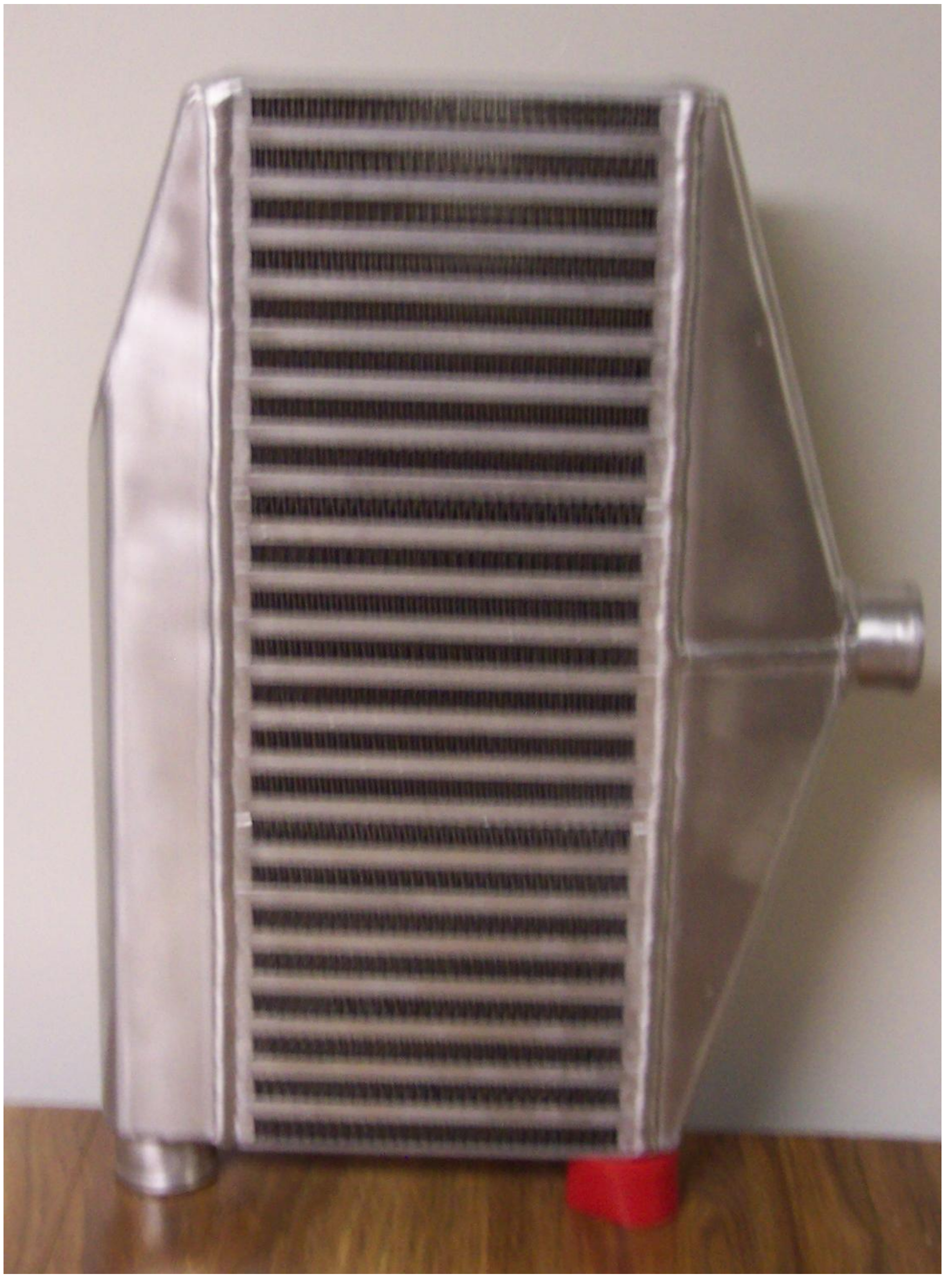

Figure 7.11 Completed intercooler.

8 – Boost Control

Boost control is really nothing more than a means of controlling the turbocharger. The turbocharger creates boost, which increases engine power. A turbocharged system and engine is only designed to handle a certain amount of boost, so there must be a means to stop the turbocharger from creating more than the system can handle. Without any such boost control, the turbocharger will create enough boost pressure to damage the engine. Detonation would probably be the first thing to occur, and it could get worse from there. There are a few different types of boost control. The intake restrictor is considered a boost control since it prevents the turbocharger from drawing in more than 229 CFM, but the vacuum problem makes it a poor choice. Blow-off valves are a better option, but they are not very accurate. Far and away the most popular choice of boost control is the wastegate, and that is what the team focused on.

8.1 – Wastegates

A wastegate is basically a poppet valve that is placed somewhere between the exhaust manifold and turbine. When the wastegate is closed, all the exhaust goes to the turbine, spinning the turbocharger as fast as possible. When the wastegate is open, a significant portion of exhaust gas will flow through the wastegate, bypassing the turbine and slowing the turbocharger. The wastegate will usually be opened or closed depending on the boost pressure, but other more complicated electronic arrangements are also possible. Wastegates can either be located inside the turbocharger or as a completely separate device.

Most wastegates use a spring and diaphragm assembly to control the poppet valve, Fig. 8.1. A signal line will be connected somewhere after the compressor to the boost pressure port. The spring will be precisely calibrated such that it will begin to compress at a predetermined boost pressure. As the spring compresses, the poppet valve rises, allowing exhaust to flow through the wastegate. This is known as simple mechanical actuation. The variables here are the spring and the signal source. The spring can be adjusted or replaced to change the boost pressure at which the wastegate will open. The location of the pressure source can be changed, as well. The wastegate will control the pressure at whatever point the signal line is sourced. It thus makes sense and most common to just source the signal directly after the compressor.

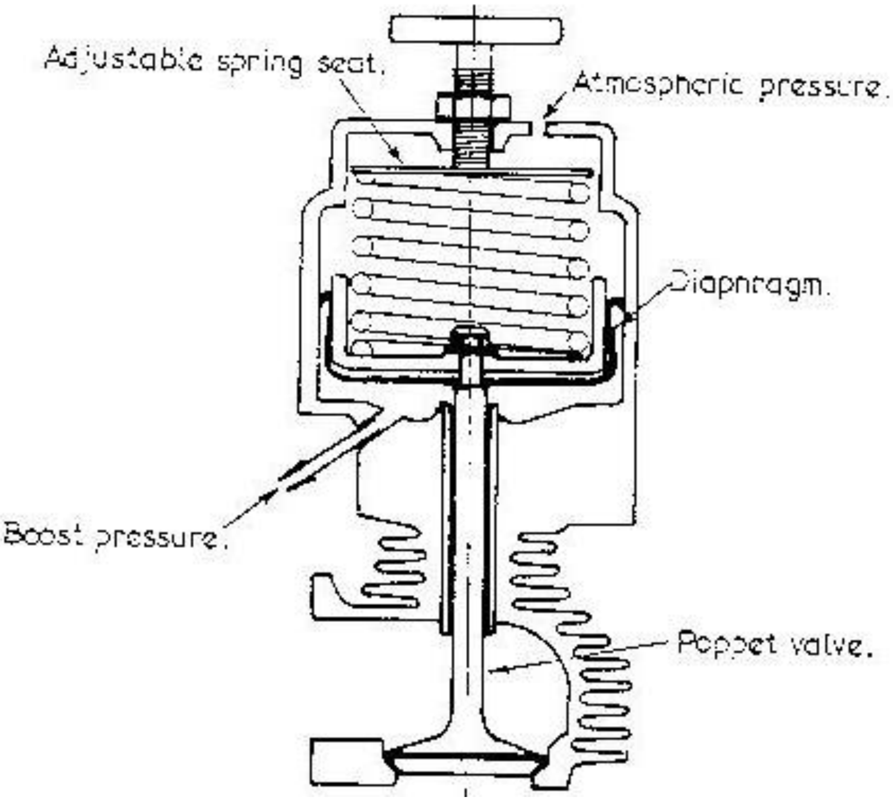

Figure 8.1 Wastegate schematic. (Retrieved from *Forced Induction Performance Tuning* by A. Graham Bell)

There are currently ways to electronically control the wastegate, but these setups also incorporate the same mechanical actuator described above. They mainly focus on manipulating the pressure signal to wastegate. Controlled leaks, mechanically or electronically, in the signal line can basically fool the wastegate into thinking that the boost pressure is lower than it really is, delaying the cracking of the poppet valve. A different approach is to manipulate the pressure differential across the wastegate spring. To do this, the atmospheric vent shown in Fig. 8.1 is replaced with a second signal line that is attached to an electronic or mechanical air regulator. The operator can then control the pressure differential across the wastegate spring, which is what actually causes the wastegate to open and close. The team was unable to find any kind of purely electronic actuation, though, that does not involve using a spring to open and close the wastegate.

Wastegates that are located inside the turbocharger are known as integral wastegates. The Garrett GT12 has an integral wastegate. An integral wastegate is a bypass valve inside the turbine which is controlled by a push rod, which is controlled by a spring in the actuator. Integral wastegates do not require any additional components be purchased as they are part of the turbocharger itself. They are thus cheap and simple to use. However, they are not ideal for performance cars. Including the bypass in the turbine housing and then dumping the exhaust right in front of the turbine outlet creates all kinds of flow problems, causing exhaust manifold pressure to rise. Some integral wastegates like the GT12's come with their own outlet, which can then have its own dump pipe. The wastegate dump pipe can be merged with the turbine dump pipe far enough downstream of the turbine to virtually eliminate the flow problems normally associated with integral wastegates.

A wastegate that is an independent device that needs to be connected into the exhaust system via dedicated pipes is known as a remote wastegate. Figure 8.1 is actually a schematic of a remote wastegate. After all the exhaust primaries have merged into a single pipe, a wastegate feed pipe can be spilt off of the main pipe to the turbine. The wastegate dump pipe can then be merged with the turbine dump pipe well downstream of the turbine. The wastegate itself just serves as a valve controlling the flow through these pipes. Remote wastegates are generally superior to integral wastegates because they cause fewer flow problems and have can have their boost pressure levels more easily adjusted.

8.2 – Boost Control Concepts

The boost control system for this turbocharged system needs to be able to effectively control over boost as well as help prevent the vacuum created by the restrictor. Ironically, limiting the creation of excessive boost and vacuum can both be accomplished by slowing the turbocharger. The boost control system must thus slow the turbocharger down when maximum boost is reached and when the restrictor chokes and begins to form a vacuum. Several different design concepts were developed to achieve this objective.

8.2.1 – Last Year's Concept

The team began its quest to attain a working boost control system by studying the options considered by last year's turbo MQP team. The design they decided on involved an air tank and a solenoid. Basically under normal boost a solenoid would route a pressure signal directly from the intake piping to the wastegate actuator of the GT12, but when the restrictor choked at a certain RPM, the ECU would switch the solenoid such that pressurized air stored in a tank would be used to open the wastegate. The tank would be filled with pressurized air from the intake pipes via a one way check valve. A slightly modified version of this design is shown in Fig. 8.2. The only difference between their design and the one pictured in Fig. 8.2 is the inclusion of the pressure regulator. Their designed system had no pressure regulator, so when the solenoid was activated, all the pressure in the air tank was sent to the wastegate actuator, completely opening the wastegate. The pressure regulator was added to allow the wastegate to be opened and closed more gradually.

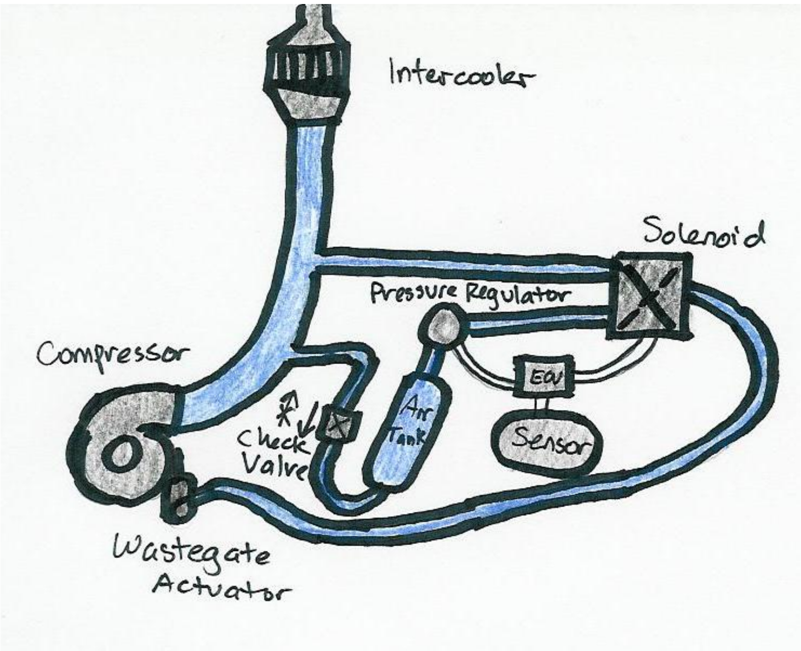

Figure 8.2 Modified version of last year's wastegate design. (Drawn by D. Curran)

After reviewing the above design, the team realized that the addition of the pressure regulator would make the solenoid unnecessary. Pressurized air could flow from the intake pipe through the check valve into the air tank, and the pressure regulator could control the pressure signal sent to the wastegate by releasing just the right amount of pressure. The pressure regulator could be controlled by the ECU, and the pressure it sends to the wastegate actuator could be linked to engine speed, boost pressure or manifold pressure. This updated version of last year's design is shown in Fig. 8.3.

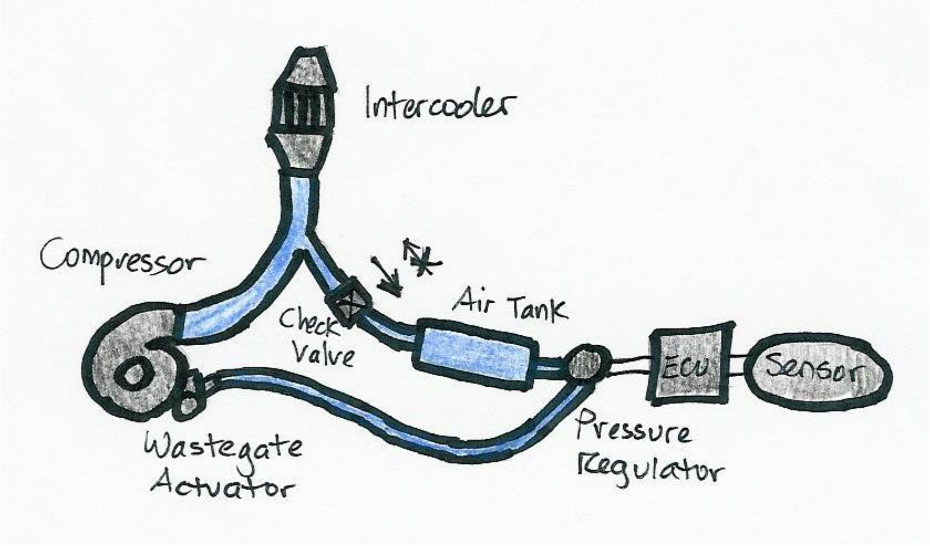

Figure 8.3 Design Concept 1: Air Tank. (Drawn by D. Curran)

8.2.2 – Motorized Concepts

The team considered two different ways of trying to use motor to somehow drive the poppet valve of the wastegate. The idea behind this was to completely divorce the wastegate actuation from the physical pressure in the intake system. Since the wastegate valve would be opened and closed via an electrical motor whose operation could be controlled by the ECU, the wastegate actuation could be linked to anything the ECU gets a reading from. The team believed that linking the wastegate actuation to the engine speed was the best way to go. The engine could then be tested on a dynamometer to determine precisely what RPM the wastegate should open and close at.

The first concept for accomplishing this purely electrical actuation, shown in Fig. 8.4, was to connect the poppet valve stem from a wastegate directly to a motor, like a linear actuator. The motor would then be powered by a battery and controlled by the ECU. The poppet valve would be pushed forward to close the valve and moved backward to open the valve. This designed system would take the place of a remote wastegate and could be programmed to open to prevent excess boost and vacuum.

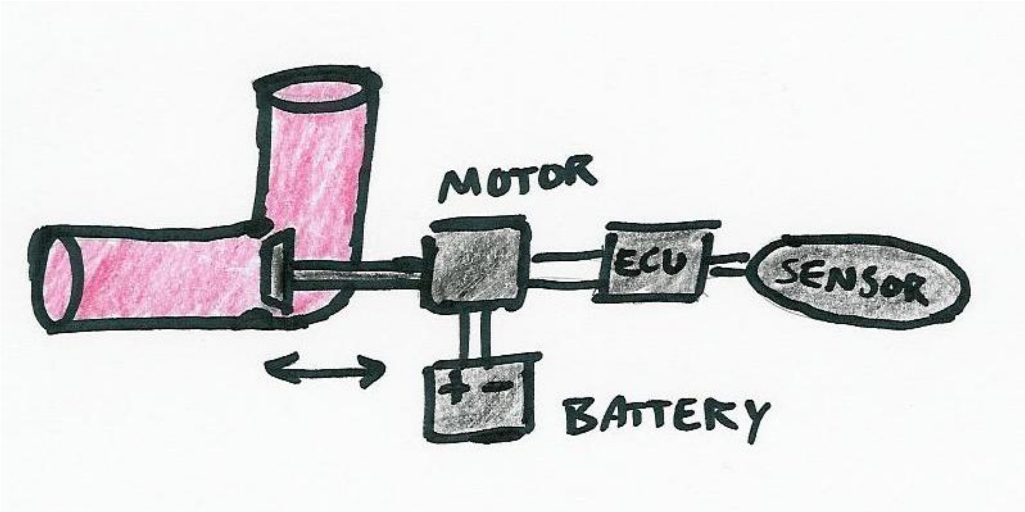

Figure 8.4 Design Concept 2: Linear Actuator. (Drawn by D. Curran)

The second idea, shown in Fig. 8.5, involved using a camshaft to open and close the wastegate. The camshaft, shown in green, would be driven by a motor controlled by the ECU. The camshaft would cause the spring, shown in yellow, to compress, which would cause the blue poppet valve to move backwards and allow exhaust to flow through the pipe. The spring would be doing all the work to keep the valve in place when the wastegate was closed. The camshaft would then be working in the same direction to compress the spring and open the valve. The rotation of the camshaft could be linked to engine speed or some other parameter measure by the ECU.

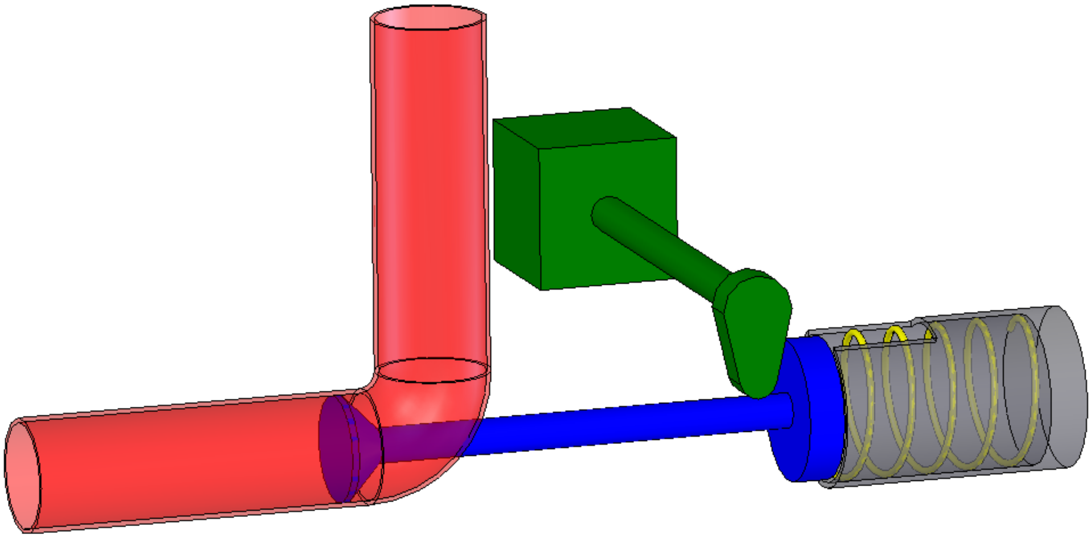

Figure 8.5 Design Concept 3: Camshaft (Modeled by D. Curran)

8.2.3 – Mechanical Concepts

There were two purely mechanical ideas floating around. Neither of them really rises to the point of being a design concept, but they are worth mentioning. The main idea here was to apply the principles of a throttle body to a wastegate, so instead of using a poppet valve, a butterfly valve (Fig. 8.6) or throttle plate (Fig. 8.7) would be used. In both cases, the valve could be opened and closed via a cable and series of linkages. This would allow the driver to have direct control over the wastegate. Of course this also means that the driver would need to fully understand the right time to open the wastegate and how much to open it.

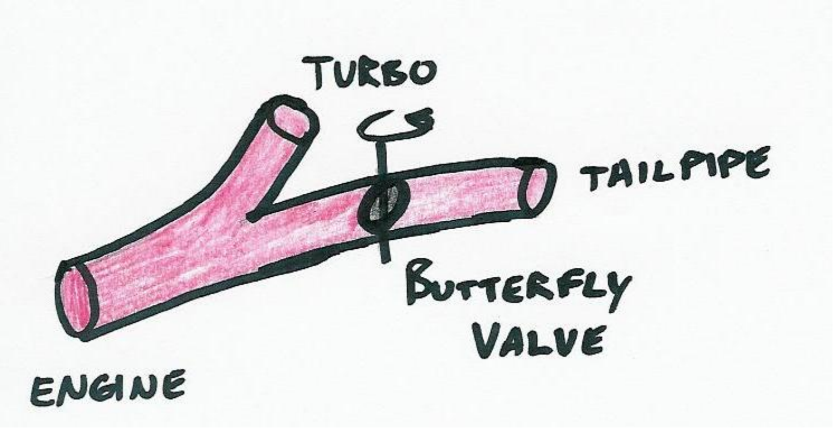

Figure 8.6 Butterfly valve wastegate. (Drawn by D. Curran)

Figure 8.7 Throttle plate wastegate. (Drawn by D. Curran)

8.2.4 – Dual Wastegate Concept

The next idea that the team considered was using two wastegates to accomplish the two separate goals of the boost control system. One wastegate would be used as the primary wastegate to prevent over boost. The other wastegate would be used as the secondary wastegate to prevent vacuum effects. This could be accomplished by reverse actuating the secondary wastegate. This basically just means reversing the direction of the pressure differential across the spring such that the wastegate will open under vacuum but remain closed under boost. A remote wastegate with two pressure ports would be needed, and the order in which signal lines were connected would be switched to accomplish the reverse actuation. A couple different variations were bounced around like using an internal or remote wastegate for the primary wastegate and whether to use the aforementioned mechanical valves or a remote wastegate for the secondary wastegate. Since the team was going to be using the GT12, it settled on an integral primary wastegate and decided to use a dual port remote wastegate for the secondary wastegate, see Fig. 8.8.

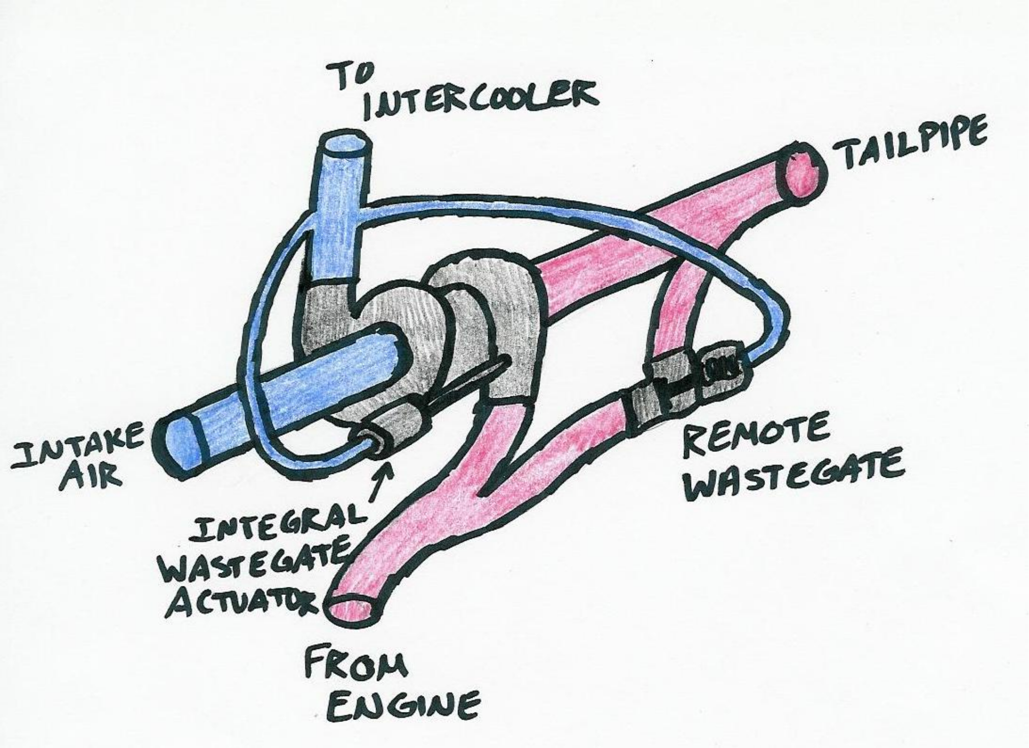

Figure 8.8 Design Concept 4: Dual Wastegates. (Drawn by D. Curran)

8.2.5 – Blow-off Valve Concept

Sticking with idea of two separate devices to accommodate the two separate needs, the team developed a concept that involved a blow-off valve and a wastegate, shown in Fig. 8.9. The blow-off valve would be placed between the compressor and intercooler, and its spring would be set to the desired maximum boost. The blow-off valve would thus open when the maximum boost was reached. This would limit the boost into engine, but it would do nothing to slow down the turbocharger, which is not a good thing. The vacuum effects would be controlled the same way as in the dual wastegate concept. A remote wastegate would be reverse actuated to open under vacuum.

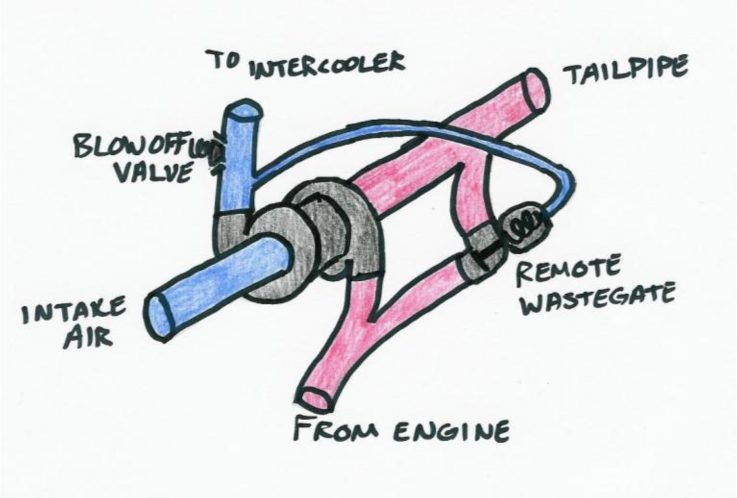

Figure 8.9 Design Concept 5: Blow-off valve. (Drawn by D. Curran)

8.2.6 – New Actuator Concepts

The final two ideas that the team came up with do not really qualify as full design concepts as they are really just a few ideas about how a custom actuator might be designed to accommodate the needs of the system. They were never fully developed. The first of these ideas is shown in Fig. 8.10. It basically consists of a throttle plate with two holes in it in between two pressure chambers. One pressure chamber would contain a spring and be vented to the atmosphere. The other pressure chamber would be empty and be connected to the intake pipes via a signal line. The pressure plate would be attached to the spring. The idea was that under boost, the spring would compress and bring the right hole into line with the exhaust pipe allowing flow. Then under a vacuum, the spring would be stretched to bring the left hole into line with the exhaust pipe allowing flow.

There would obviously be problems with setting up a spring that would yield these results.

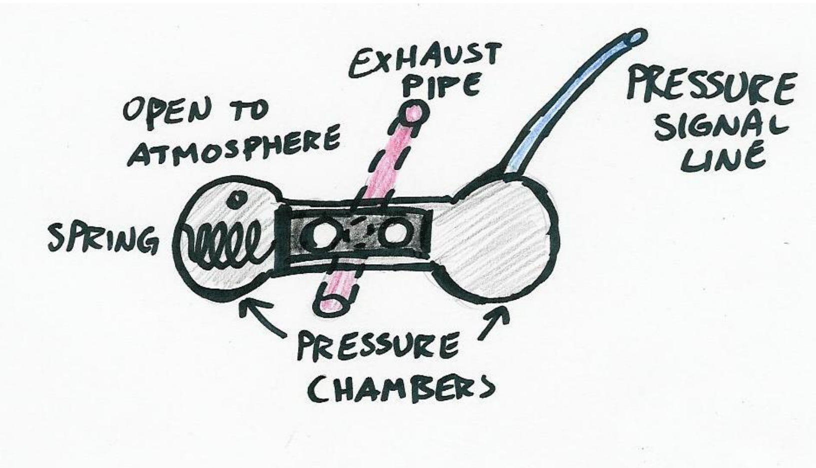

Figure 8.10 Pressure chamber idea. (Drawn by D. Curran)

The second idea was to modify the actuator of a remote wastegate as shown in Fig. 8.11. The poppet valve stem would be connected to special base with two springs, and the actuator would have two pressure inputs controlled by a solenoid. Under boost, the pressure signal would be sent to port A, which would cause the large yellow spring to compress, opening the valve. Under a vacuum, the solenoid would switch the signal to port B, and the vacuum would cause the smaller green spring to compress. The special base would be designed to allow the smaller spring to compress without the larger spring compressing. Thus the large spring and port A could be used to prevent over boost, and the small spring and port B could be used to slow the turbocharger under vacuum conditions.

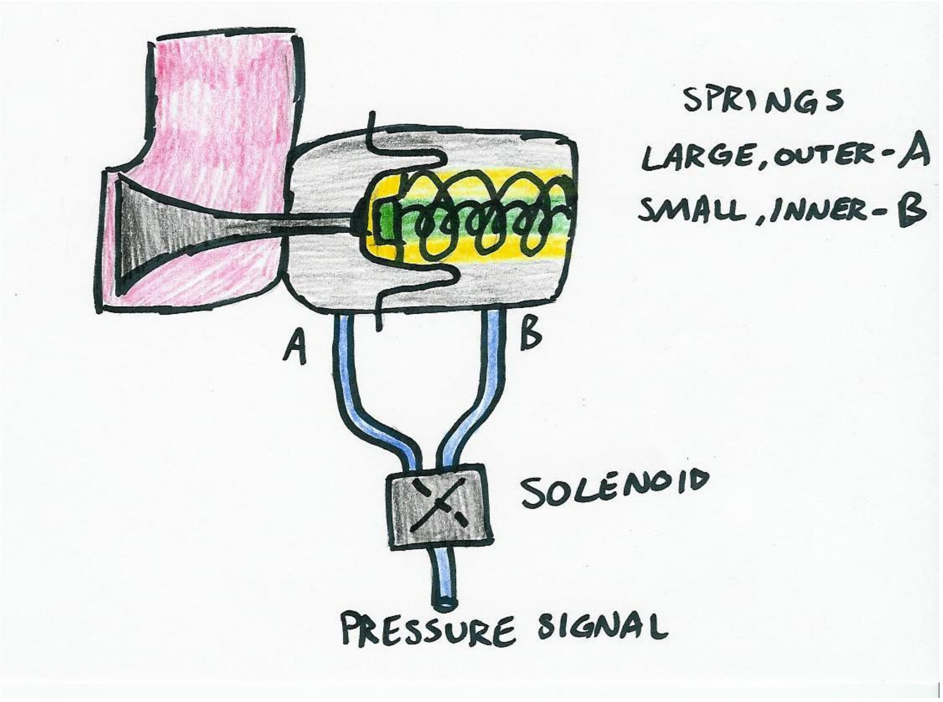

Figure 8.11 Dual spring actuator idea. (Drawn by D. Curran)

8.3 – Selection of the Final Design

Out of all the ideas and concepts described in the previous section, five potential design concepts emerged as worthy of further consideration. These design concepts were the air tank (1), linear actuator (2), camshaft (3), dual wastegate (4) and blow-off valve (5). Each of these will now be discussed briefly before the final design is selected.

The air tank design was modified from last year's turbo MQP, and as such this was really their idea. The team was skeptical but wanted to give it its due consideration. There was a lot of discomfort with the entire wastegate system depending on an air tank, especially when testing last year showed the presence of leaks in their prototype. Furthermore, it seemed exceedingly difficult to develop the ECU-controlled pressure regulator system, which was essential in pulling off this idea. Ultimately, this concept was deemed to just be too complicated with too many potential problems.

The linear actuator design had two major problems. The first was that exhaust gases would be exerted a very strong force on the poppet valve when the wastegate was closed. This means that the motor would constantly have to be exerted a strong counter force to keep the valve closed, which would require a lot of power and could burn out the motor quickly. The other problem involved the electrical and programming aspects of the design. There would need to be a relatively sophisticated wiring scheme connecting the motor, battery and ECU in order to set up the system such that the ECU could control the operation of the motor when neither component was meant to interface with the other. The electrical complexity aside, the risk of burning out the motor and losing the wastegate was too much to overlook.

The camshaft design was actually a big improvement over the linear actuator. The spring would be opposing the forces of the exhaust on the poppet valve, so it would simply be a matter of determining the magnitude of these forces and choosing a spring with the appropriate stiffness to completely oppose the force of exhaust. The spring could be chosen, though, that it could stand the added force of the camshaft. The cam would thus have to exert relatively little force on the spring to compress it, and when that force was removed, the spring would close the valve. The movement of the camshaft could then be linked to the engine speed or to boost pressure to prevent over boost and limit vacuum effects. However, this design would require a lot of wiring and programming.

The dual wastegate approach required a bit more plumbing than some of the other designs but very effectively addressed both boost control needs in a purely mechanical manner. Unfortunately, this meant that wastegate actuation had to be linked to pressure in the intake and could not be linked to engine speed. The flip side of this is that there was no complicated wiring or programming involved at all. The integral wastegate could be set to the maximum desired boost pressure and used to prevent over boost, as long as a separate dump pipe was used to prevent back pressure. The remote wastegate could easily be reversed actuated and set up with a weak spring to open under vacuum. Both wastegates would slow the turbocharger down to prevent over boost or vacuum.

The blow-off valve design would be decent approach for a turbocharger without an integral wastegate. However, since the GT12 does have an integral wastegate, adding a blow-off valve to accomplish the same thing the integral wastegate would (and does in the dual wastegate design) just does not make a lot of sense. Furthermore, a blow-off valve is inferior to a wastegate because it is less accurate and does not actually slow the turbocharger down. It just wastes boost.

After careful consideration, the team decided that the camshaft approach would be best overall design for the system. However, due to its complexity, it was deemed impossible to actually complete on top of everything else. The team thus moved on to its second option, the dual wastegate approach. The dual wastegate design is a simple mechanical design that effectively accomplishes its objectives and was certainly something that could be completed.

8.4 – Location of Pressure Sources

With the wastegate design determined, the location of the pressure sources could be considered. The location of the pressure source is the spot in the intake system at which the wastegate will control the pressure. There are two wastegates in use in the chosen design, and thus there are two pressure signals needed. The integral wastegate is intended to prevent over boost, and as such its pressure signal should come from the place in the intake system with the highest pressure to make sure that maximum boost is never exceeded. This spot would be right after the compressor. Intake air is at atmospheric pressure entering the compressor, and the intake air tends to lose some boost pressure across the intercooler, making the hose between the compressor and intercooler the high water mark for pressure. The remote wastegate is intended to open under a vacuum to slow the turbocharger and decrease the vacuum effect. As such its pressure signal should be sourced to the spot in the manifold that will experience the strongest and earliest vacuum. The team's instinct was that this was after the restrictor but before the compressor. Kevin Luchanski, a WPI alum and former FSAE team member, was gracious enough to assist the team here. He suggested sourcing the remote wastegate approximated 4 inches after the throat of the restrictor in order to get the strongest vacuum signal. Figure 8.12 shows the signal lines from the intake hoses to the wastegates.

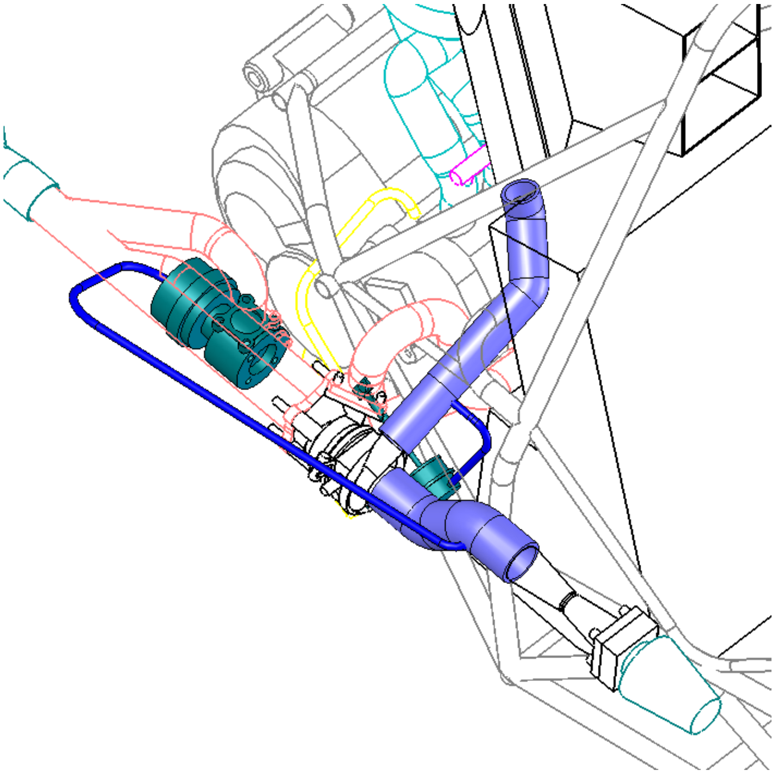

Figure 8.12 Wastegate signal lines. (Modeled by D. Curran, engine/chassis by FSAE team)

8.5 – Obtaining the Necessary Components

Several piece of hardware had to be purchased to complete the designed wastegate system. First and foremost, a remote wastegate had to be purchased. After extensive searching the TiAL F46 remote wastegate was found. This particular wastegate, shown in Fig. 8.13 with its flanges and gaskets, has the distinct advantage of coming with two pressure signal. This means that the signal line arrangement would just have to be switched to reverse actuate the wastegate, which saved the team from having to drill a new signal port into another wastegate actuator. The TiAL F46 was purchased with a 2.9psi spring, the weakest available, to cause the wastegate to open as soon as possible under vacuum.

Figure 8.13 TiAL F46 remote wastegate with flanges and gaskets.

The next component that needed to be considered was the integral wastegate actuator. The GT12 came equipped with a 7psi actuator, but the team hoped to produce a lot more boost than that. Therefore an 18psi actuator for an integral was located and purchased, shown in Fig. 8.14. This actuator could be used to replace the 7psi actuator on the GT12 to greatly increase the maximum boost level. The team decided to hold off on actually switching the actuators. It was decided to get the system running at 7psi boost and then switch to 18psi if all went well.

Figure 8.14 18psi integral wastegate actuator.

The next consideration was the signal line ports. Both signal lines were located on silicone hoses, so a means to put a port on the silicone has to be found. Fortunately, the team was able to find a product specially designed for this purpose, Fig. 8.15. A hole can be drilled into the silicone and these ports can be inserted into it.

Figure 8.15 Wastegate signal line ports.

The final component of the wastegate system was the only not purchased. This is component is the actual signal line. Several feet of signal line would need to be purchased and cut to the write length. Some clamps for the hoses would also be needed. The line should be vacuum resistant as it will experience a strong vacuum.

8.6 – Results

A dual wastegate system was designed to prevent over boost and limit vacuum effects caused by the restrictor. The integral wastegate was initially set to 7psi and sourced after the compressor to prevent over boost. An 18psi actuator has been purchased to replace the 7psi actuator when the system is ready for the real power. The remote wastegate was reversed actuated with a 2.9psi actuator sourced after the restrictor to reduce the vacuum effects of the restrictor. All necessary components except for the signal lines and their clamps were purchased.

9 – Exhaust System

The engine's exhaust system has the job of gather scorching hot exhaust from all of the cylinders and delivering to components like the turbine. The exhaust system basically consists of everything from the exhaust ports to the tailpipe. This includes the exhaust manifold, the turbocharger, the wastegate and the muffler as well as all the intermediate plumbing. Since the turbocharger and wastegate have already been discussed in previous sections, this section will focus primarily on the exhaust manifold and the plumbing between the major components.

9.1 – Some Basics

The primary job of the exhaust system is to direct the exhaust from the exhaust ports to the turbocharger, wastegate and muffler, but this is hardly the only consideration when designing an exhaust system. The exhaust system usually will have the added responsibility of supporting the weight of these components, so some care must be taken to ensure that the pipes can handle the weight. If they cannot, then extra supports will be needed.

There are several features that make a particular exhaust system desirable. The first such feature is heat retention. The exhaust pipes should keep as much heat in the gas as possible, as this extra energy benefits the turbine's performance. The exhaust pipes should have relatively small diameters to keep the gases moving at higher velocities, which accelerate the turbine quicker. All bends should be large radius turns and angles should be as shallow as possible to keep the gas flowing smoothly. Flow obstructions lead to increased pressure in the exhaust system and can cause reversion. Exhaust systems should be made stainless steel if possible as this material provides excellent heat retention and long term durability. Mild steel and cast iron are also acceptable choices.

9.2 – Exhaust Manifold

The exhaust manifold consists of all the plumbing from the exhaust ports until the pipes coming out the ports, known as primaries, have been merged together into a single pipe. There are two basic types of exhaust manifold, shown in Fig. 9.1. The log style manifold gets its name from the fact that it consists of a single, larger diameter pipe into which very short primaries merge one after another at abrupt angles, kind of like the primaries are going into a log. The color of manifold gets darker to the left because that part of the manifold experiences four exhaust pulses (one per cylinder) for each engine cycle whereas the other end experiences one pulse for each engine cycle from the very last runner. The side that experiences more pulses tends to have durability issues. The individual runner style manifold has much longer primaries that merge together much more smoothly. The primaries are designed such that they are all the same length. The actual length of the primaries can be played with to increase the engine's performance. Individual runner style manifolds are generally considered far superior from the performance standpoint, but the log style manifold does provided an alternative for those with a small budget and an even smaller space for their exhaust manifold. Despite having a limited budget and an extremely small area to work with, the team decided to try to design an individual runner style manifold. After all, this system is for a racecar. Performance is a fairly high priority.

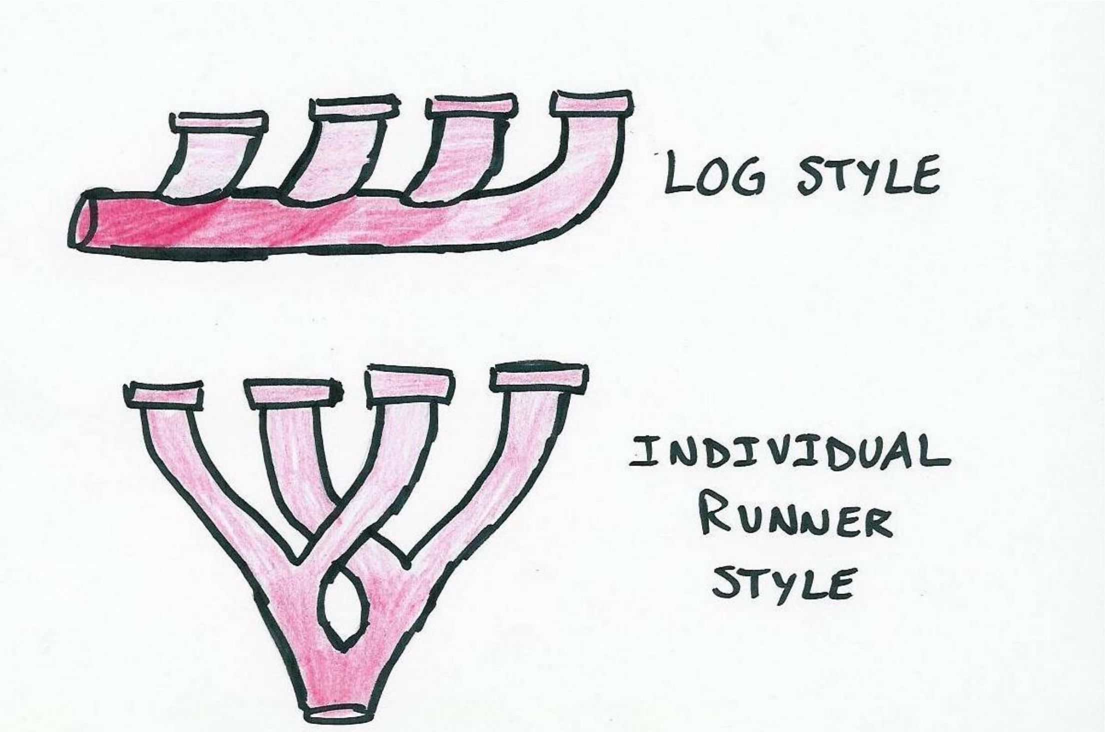

Figure 9.1 Styles of exhaust manifold. (Drawn by D. Curran)

With the decision to use an individual runner style manifold made, the team now had to decide what kind of merge system to use. For a four cylinder engine, there are basically two choices, shown in Fig. 9.2. The first type is a 4-1 merged manifold. In this type of manifold, the four exhaust primaries come together into a single merge collector that has four inlets and one outlet. The second type is a 4-2-1 merged manifold. This type of manifold uses three merge collectors with two inlets and one outlet. Each primary will merge with one other primary to form pipes called secondary pipes. The secondary pipes merge together to finally form the single pipe out of the manifold. While the figure shows the manifolds as approximately the same size, the 4-1 manifold is actually much compact as it has a third as many merge collectors and no secondary pipes. For serious engine tuners, the 4-2-1 will prove superior because it can be tuned to help promote exhaust scavenging, which is the process of timing exhaust pulse to compliment each other. If done correctly, this can result in a vacuum pulse reaching the exhaust valve of the cylinder just as it is opening, which will help draw the exhaust out of the cylinder, helping it fill with the new air/fuel mixture. A 4-1 manifold cannot be tuned to promote scavenging. Thus the performance edge goes to the 4-2-1, but limited space tends to favor a 4-1 manifold. The team went with the individual runner style manifold for performance reasons and would have like to have gone with a 4-2-1 manifold for the same reason, but the space limitations and system layout simply made that impossible. A 4-1 individual runner style exhaust manifold was thus to be designed.

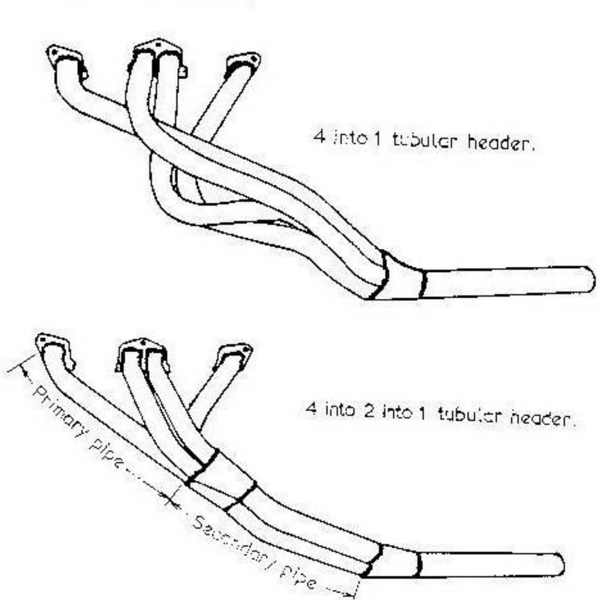

Figure 9.2 Different types of merged runner style manifolds. (Retrieved from *Forced Induction Performance Tuning* by A. Graham Bell)

The team bought a 4-1 merge collector, shown in Fig. 9.3, intended for use with 1.375 inch OD primaries and an outlet pipe of 1.625 inch OD. The merge collector was then measured and a model was created in SolidWorks. This model was used extensive as a tool in designing the primaries. The major concern in designing the primaries was simply packaging. There was very little room to work with, and whatever could fit was the best that could be done. The other two considerations were equalizing the lengths of the pipes and limiting abrupt bends as much as possible.

Figure 9.3 Merge collector, 4-1.

The primaries were modeled entirely within the system assembly by fixing the position of the merge collector and routing the primaries to it. At first the merge collector was positioned horizontally, with its outlet facing the turbocharged and its inlets about even with the second exhaust port. Much effort was expended trying to develop equal length primaries that could make it into the merge collector. This effort ultimately proved fruitless. It was impossible to equalize the lengths of the primaries with the merge collector in this position. Additionally, just getting the primaries into the merge at all resulted in very sharp bends.

It was decided that the merge collector should be moved. The merge collector was positioned vertically between the second and third exhaust ports such that its outlet was about even with the heights of the exhaust ports. The primaries were then wrapped underneath and into the merge collector. This proved to be a much more satisfactory design. The primaries were perfectly symmetric and their lengths were all about 14.2 inches. Figure 9.4 shows the model of this manifold.

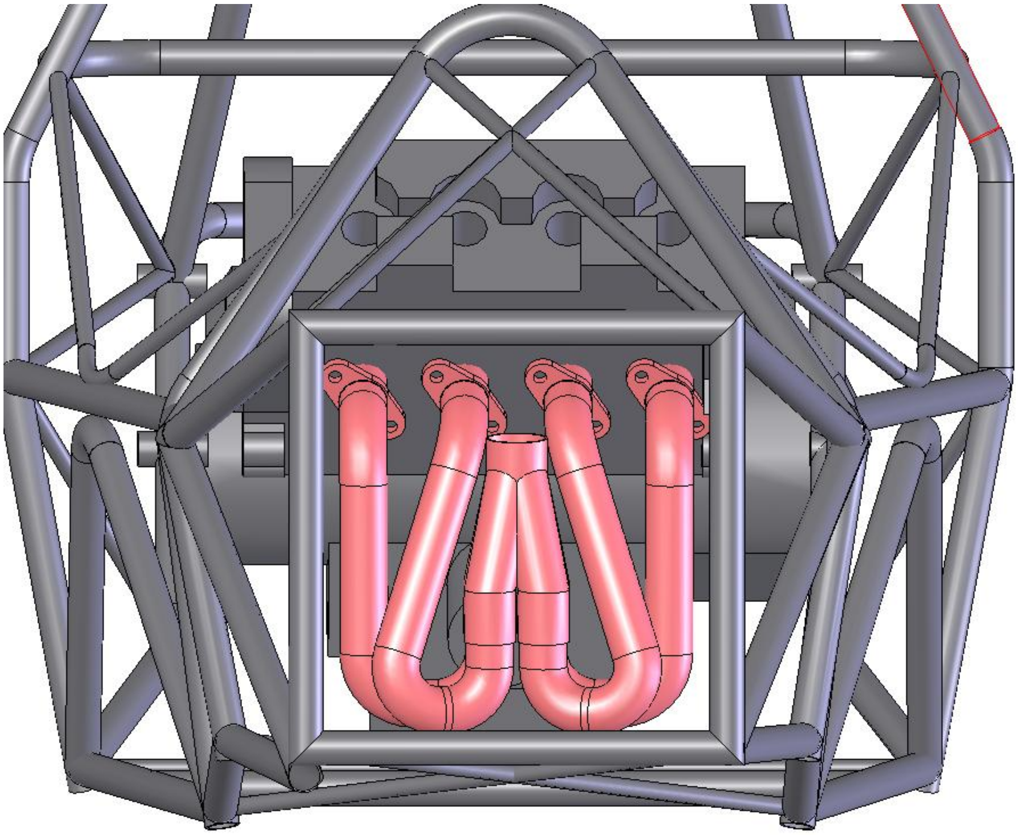

Figure 9.4 Designed exhaust manifold. (Modeled by D. Curran, engine/chassis by FSAE team)

While the design of manifold was completed, the only component that was purchased was the merge collector. Drawings for the primaries are available in Appendix K. The pipes for the primaries would have required a mandrel bender and were far too expensive to afford.

9.3 – Remaining Exhaust Pipes

The remaining exhaust pipes were designed to direct the exhaust to and from the major components of the exhaust system as smoothly as possible given the tight packaging limitations. A pipe was added coming out the merge collector to deliver the exhaust to the turbocharger as shown in Fig. 9.5. It can be seen that the pipe does not make it to the turbine inlet. A special steel reducing cone had to be designed along with a flange for the turbine inlet. A special turbine outlet flange was designed as well. The drawings for these parts are in Appendix K. Figure 9.6 shows the cone and flanges added along with the turbine dump pipe connecting the turbocharger and muffler.

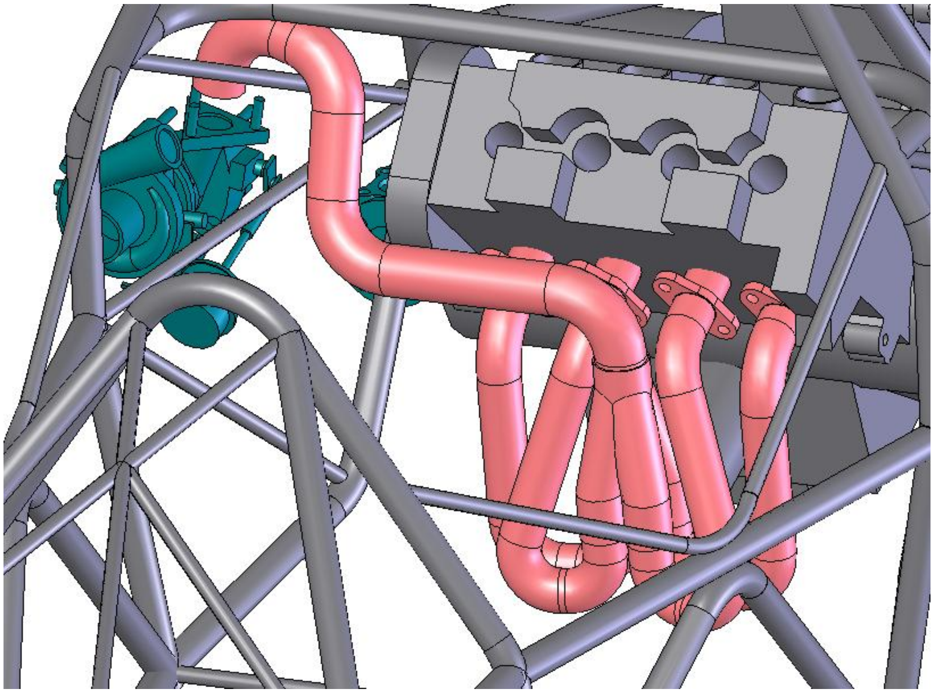

Figure 9.5 Turbocharger feed pipe added. (Modeled by D. Curran, engine/chassis by FSAE team)

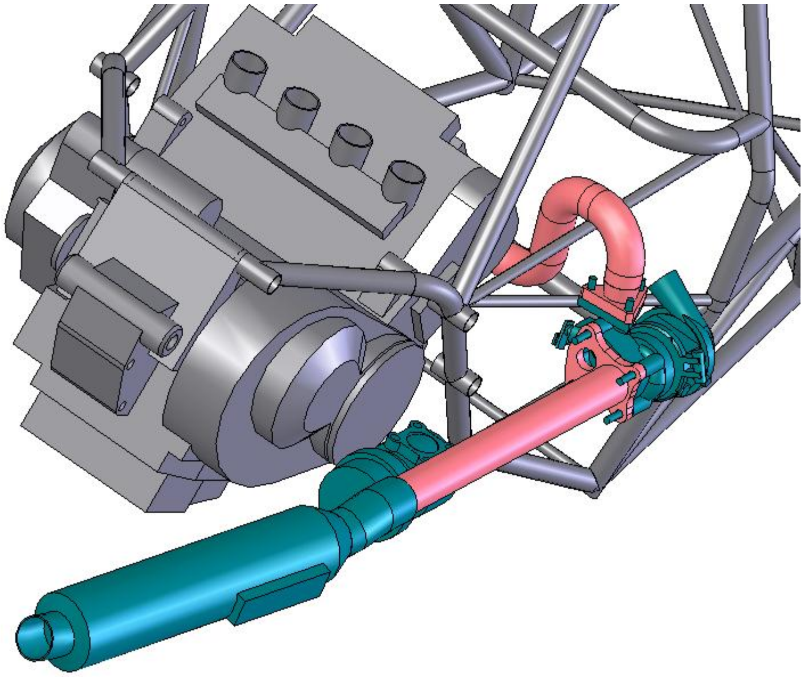

Figure 9.6 Flanges and dump pipe added. (Modeled by D. Curran, engine/chassis by FSAE team)

The next step was to add a separate dump pipe for the integral wastegate. Like the turbine dump pipe, this pipe would be welded to the turbine outlet flange. This integral wastegate pipe extends about 12 inches past the flange before merging into the turbine dump pipe as shown in Figure 9.7.

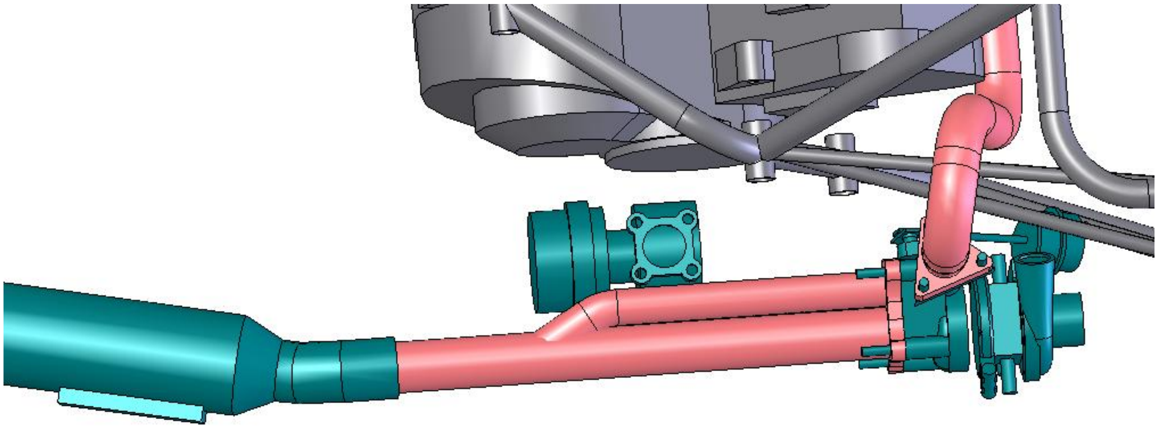

Figure 9.7 Integral wastegate dump pipe added. (Modeled by D. Curran, engine/chassis by FSAE team)

All that remains is to add the pipes connecting the remote wastegate into the system. The feed pipe needs to split off from the exhaust pipe between the merge collector and turbine at a shallow angle if possible. The dump pipe needs to be merged with the turbine dump pipe like the integral wastegate dump pipe. Figure 9.8 shows the addition of these pipes.

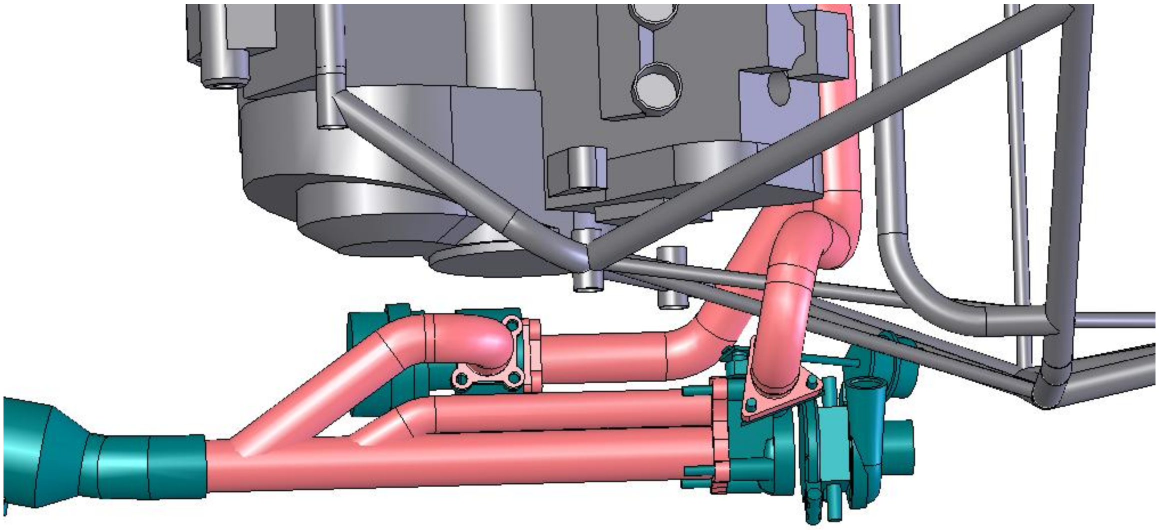

Figure 9.8 Remote wastegate plumbing added. (Modeled by D. Curran, engine/chassis by FSAE team)

The turbine inlet and outlet flanges were successfully machined with a water jet at Merchants Sheet Metal free of charge, Fig. 9.9 and Fig. 9.10 respectively. Figure 9.11 shows these two flanges on the turbocharger. The turbine inlet cone, Fig. 9.12, was partially turned on the lathe, but since it is made of steel, it damaged the tools and could not be completed. The pipes were going to be purchased with the pipes for the primaries, but they all turned out to be too expensive.

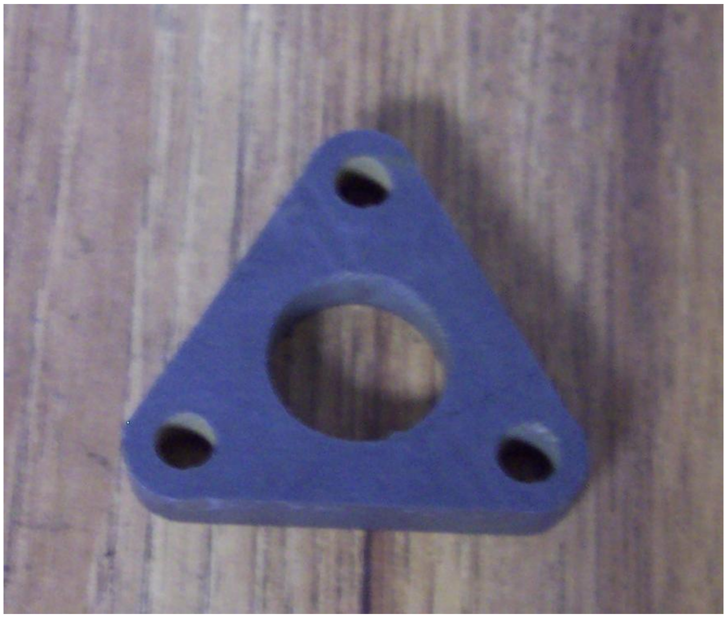

Figure 9.9 Steel turbine inlet flange.

Figure 9.10 Steel turbine outlet flange.

Figure 9.11 Flanges on turbocharger.

Figure 9.12 Partially machined turbine inlet cone.

9.4 – Results

The design of the exhaust system was completed. The model of the exhaust system is shown isolated in Fig. 9.13. All of the pipes necessary to complete this design are listed in Appendix H. None of these pipes were purchased for the same reason the intake runners were not purchased. It was simply too expensive to have the necessary pipes bent. All of these necessary flanges and gaskets were purchased or machined. The lone reducing cone could no be completed due to a lack of proper steel cutting inserts in Washburn.

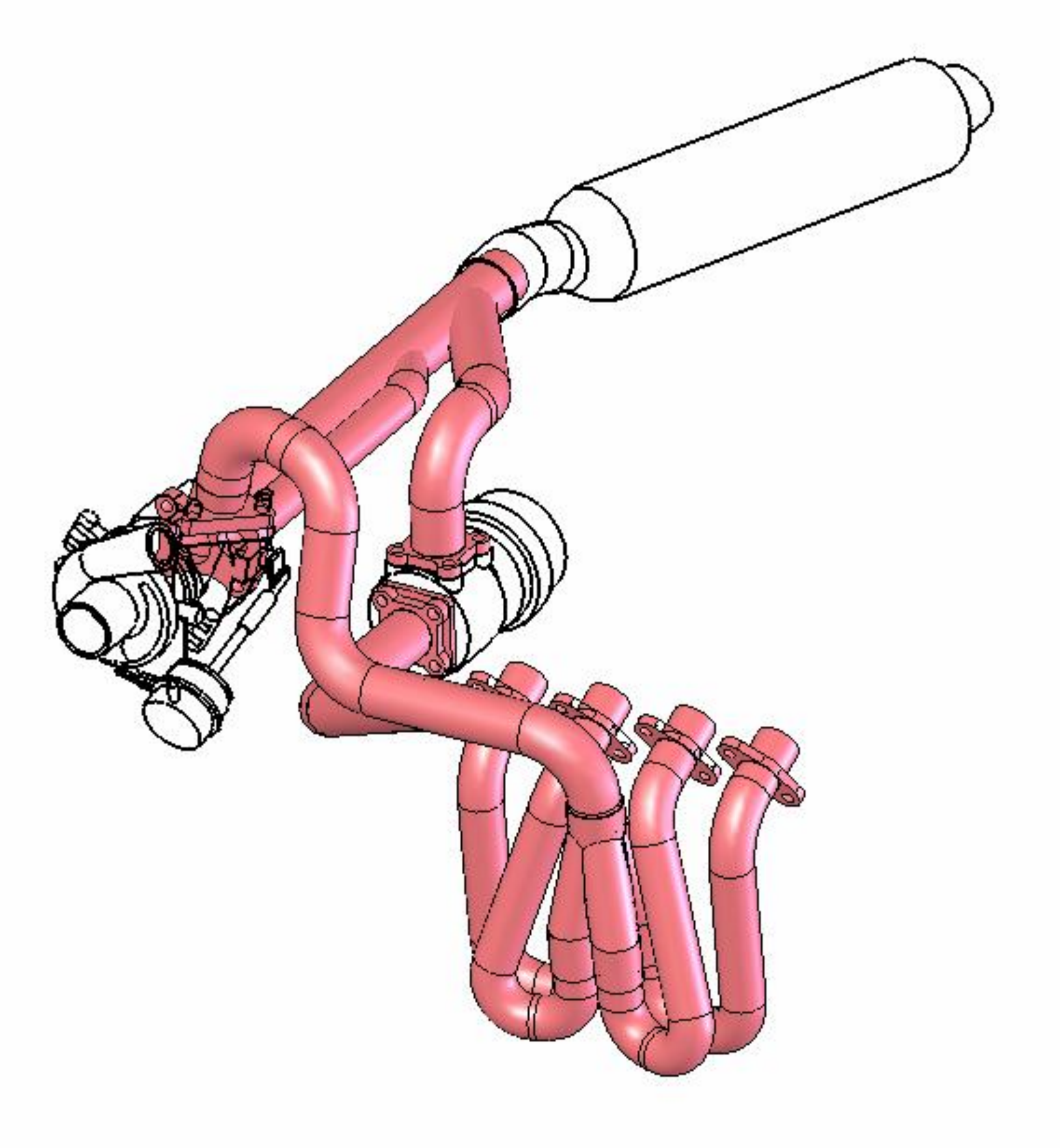

Figure 9.13 Complete exhaust model, isolated. (Modeled by D. Curran)

10 – Turbocharger Bearings

The bearings section of the turbocharger must be properly lubricated in order to keep the turbocharger working. For the GT12, this requires routing both oil and water from the engine into the bearings section. The requirements for the oil and water delivery systems will now be discussed.

10.1 – Lubrication

The turbocharger is lubricated with oil directly from the engine. Therefore the type of oil in the engine is important. Wide range, multi-viscosity oils and mineral oils should be avoided since they tend to cause coking. Synthetic oil with a straight viscosity works best. Adding the turbocharger to the oil circuit will require more frequent oil changes as the heat of the turbocharger causes the oil to break down more quickly.

Most engines create a lot of oil pressure, so there should not be a problem with the oil pump being unable to produce enough oil pressure for the turbocharger. Sometimes the engine will create too much pressure, and a restrictor or regulator will be needed to limit the pressure at the turbochargers. A turbocharger works well when it receives oil within a certain pressure range, which is put out by the turbocharger manufacturer. For the GT12, the minimum oil pressure is 12-15psi and the maximum oil pressure is 35-40psi. The pressure created by the engine needs to be determined in order to assess what needs to be done with the turbocharger oil feed.

The actual hoses delivering the oil to the turbocharger and returning it to the engine need to be able withstand high pressures and temperatures. Metal braided lines are a popular choice for these hoses, but care must be exercised in placing such hoses as they can cause damage to components they rub up against. The oil feed line delivers oil from a positive pressure location on the engine to the turbocharger bearings. This line should be at least 0.25 inches ID. The oil drain line returns the oil to the engine oil pan. This hose needs to be at least 0.5 inches ID to ensure proper draining. The oil tends to come out of the bearings section more like whipped cream than liquid oil and thus needs a large area through which to flow. If gravity is the only force acting on the oil, then the drain line should exit the turbocharger vertically and sweep smoothly into the oil pan without ever being horizontal or twisting or kinking. These requirements can be relaxed if a dedicated oil pump is added to help pump the oil from the turbocharger to the oil pan.

In this system, care was taken to place the turbocharger high enough to allow a straight gravity drain into the engine. The dedicated pump was deemed to be too much weight to add, and there did not appear to be room for it anyway. At the recommendation of the FSAE team, the oil feed is going to be taken from the oil pressure sender via a tee splitter. A metal braided hose could then be screwed onto the threaded adapters on the tee splitter and the custom oil flange. A 0.5 inch ID metal braided hose would then sweep smoothly into the engine oil pan having been screwed onto an adapter on the oil flange and another adapter welded onto the oil pan. Figure 10.1 shows a model of these oil lines.

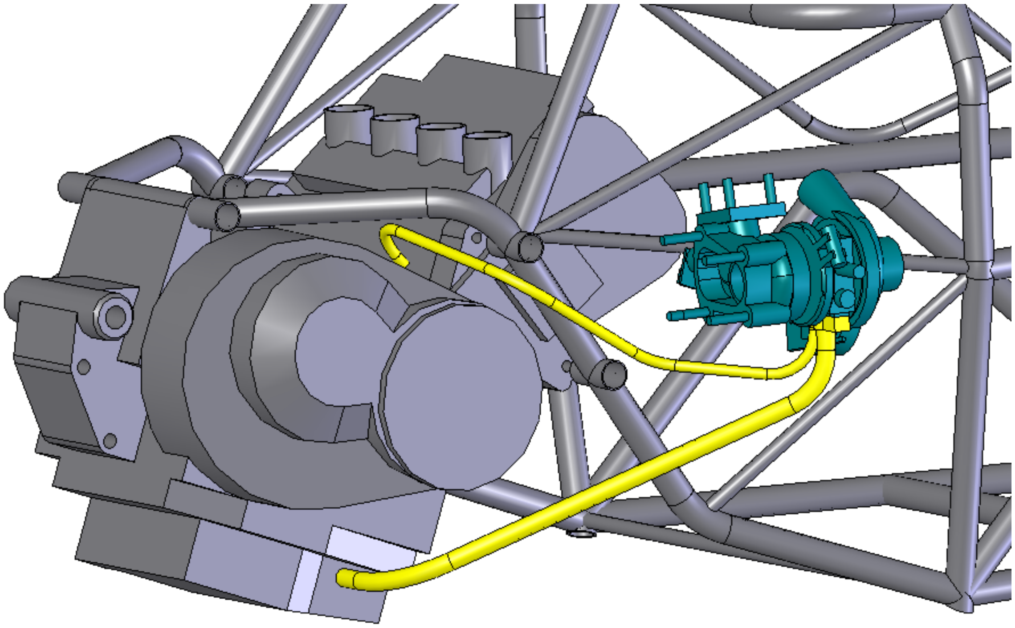

Figure 10.1 Oil feed and drain lines. (Modeled by D. Curran, engine/chassis by FSAE team)

The necessary tee splitters and adapters were purchased. Merchants Sheet Metal manufactured the custom oil flange, the drawing for which is in Appendix L, free of charge using a water jet. All of these components are shown in Fig. 10.2.

Figure 10.2 Tee splitter, adapters and oil flange.

Unfortunately there were a few problems with oil flange. Because of the poorly designed oil face on the GT12, the holes on the flange were too close together to drill threads into them. This created a major problem that was never solved. Since the problem was not solved, metal braided lines were not purchased. An additional problem with oil flange had to with a problem with the water jet. There was a leak in the compressor that required the water jet to make two passes, resulting in the choppy piece shown in Fig. 10.3.

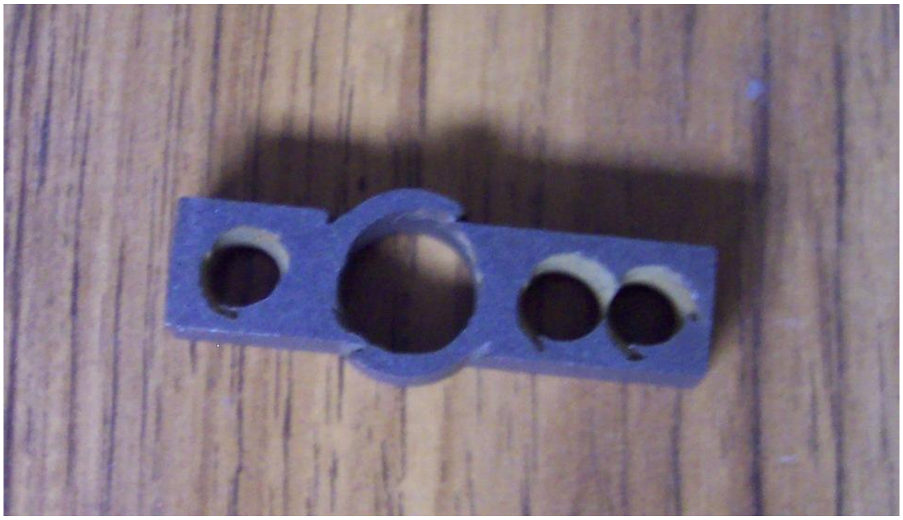

Figure 10.3 Damaged oil flange.

In an attempt to resolve the oil flange problem, a new concept was developed. Instead of screwing adapters into the oil flange, threaded tubes will come off of the flange. The hoses can then be screwed to the ends of the tubes. This might solve the oil flange problem but the team did not get a chance to test it out.

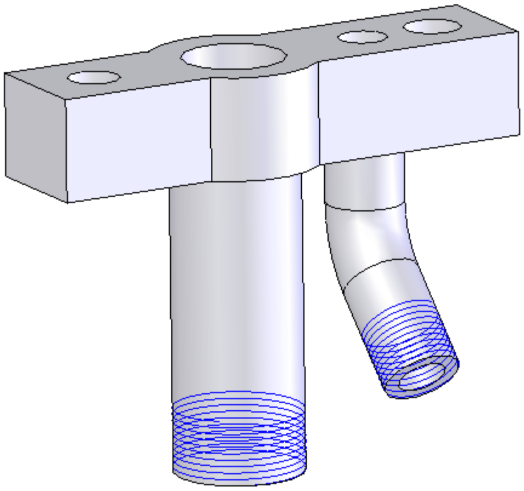

Figure 10.4 New oil flange concept. (Modeled by D. Curran)

10.2 – Cooling

The immense heat of the exhaust flowing through the turbine can cause the oil to coke, blocking the oil flow and killing the bearings. By adding a water jacket to the bearings, Fig. 10.5, the oil temperature can reduced to the point at which it will not coke, Fig. 10.6. While not all turbochargers have a water jacket, it is a very desirable feature to have and should be considered a must.

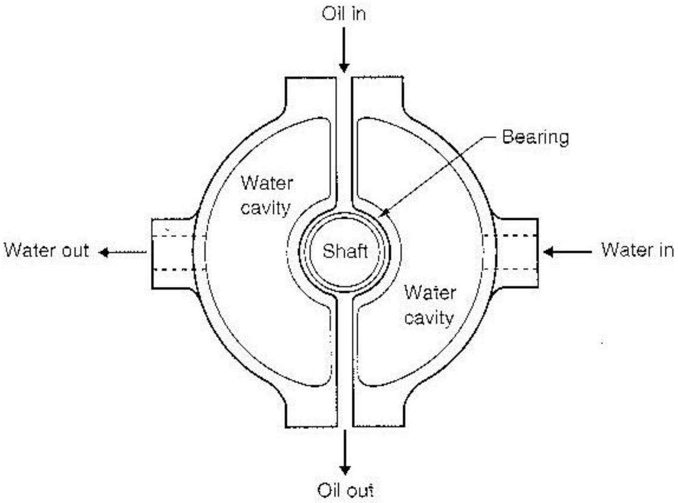

Figure 10.5 Water jacket around bearings. (Retrieved from Corky Bell's *Maximum Boost*)

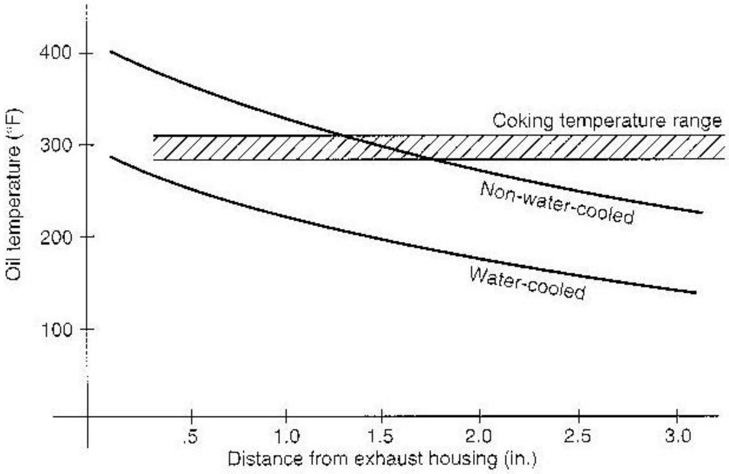

Figure 10.6 Effects of water-cooled bearings. (Retrieved from Corky Bell's *Maximum Boost*)

Fortunately, the GT12 does have a water jacket, so oil coking should not be an issue. Engine coolant can just be routed from the radiator circuit to the water ports on the bearings section. There are no flow, pressure or orientation requirements for the water jacket, so setting up the water lines here is pretty simple. The hoses and splitters were not purchased.

10.3 – Results

The general design of the lubricating and cooling systems, Fig. 10.7, was completed, but some of the details were not worked out and very few of the components were purchased. The most notable problem was with the oil flange. A solution to the oil flange problem must be found before the rest of the details can be worked out. Once the oil flange problem is solved, the hoses necessary to complete the system can be purchased. The oil pressure needs to be test at the tee in order to determine whether or not an oil restrictor or pressure regulator is needed to limit the pressure into the turbocharger.

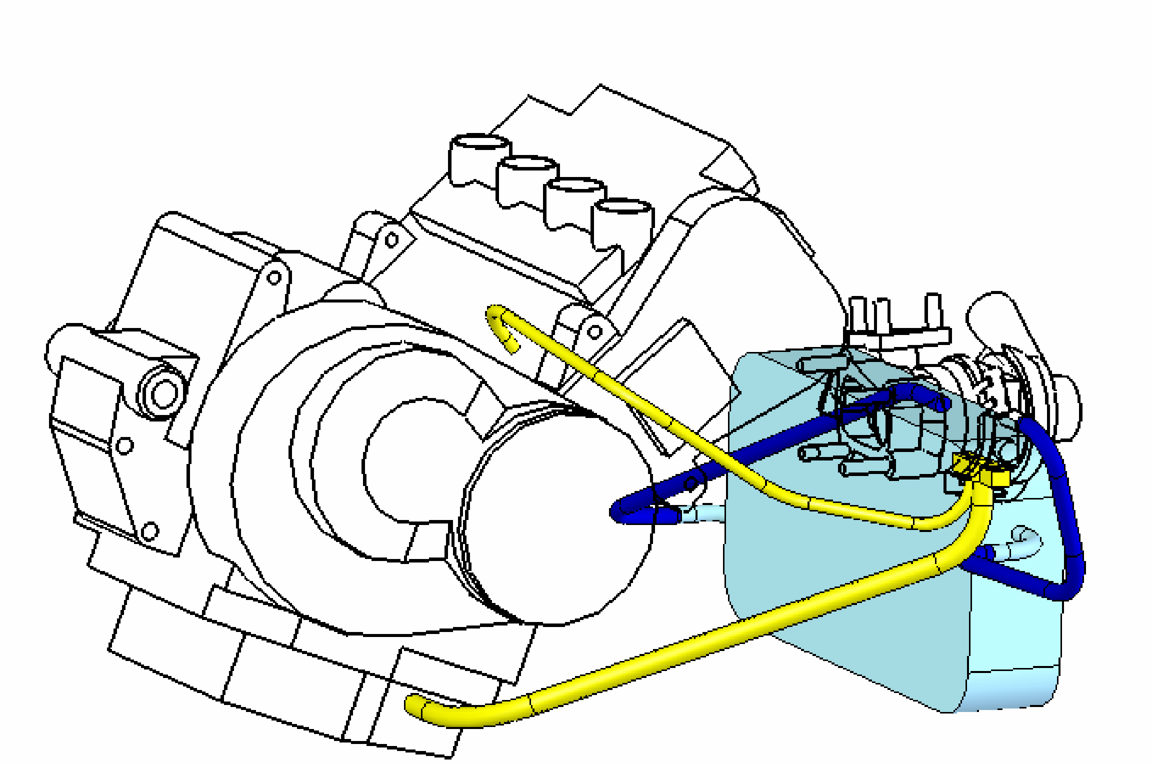

Figure 10.7 Oil and cooling lines attached to turbocharger, isolated. (Modeled by D. Curran, engine by FSAE team)

11 – Engine Modifications

Turbocharging an engine requires paying careful attention to several of the engine's critical components. Some components will need to be upgraded or replaced while others will be fine. It all depends on the desired boost pressure. Higher boost pressures mean more engine modifications.

The stock ECU will need to be replaced. Most stock ECUs cannot easily be modified to advance/retard the ignition spark and most are not capable of maintaining a correct AFR as boost pressure increases. Furthermore, any goals of electronic wastegate actuation will go unfulfilled with a stock ECU. An aftermarket ECU should be found that can control ignition timing, maintain a correct AFR at the anticipated boost levels, and control wastegate actuation. The team found several possibilities for the aftermarket ECU. The Electromotive TEC^3 and Perfect Power XMS were two such possibilities. Members of the FSAE strongly recommended using the Performance Electronics PE-ECU-1 and assured the team that it could everything that was required of it. The team decided to follow this advice and purchase the PE-ECU-1, shown in Fig. 11.1. With the addition of the new ECU, a new wiring harness will need to be built. In this case, the intake air temperature and coolant sensors had to be replaced to work with the ECU. A 3-bar MAP (pressure) sensor needed to be purchased to accommodate the high boost pressures the team was shooting for.

Figure 11.1 PE-ECU-1 aftermarket ECU.

The EFI (electronic fuel injection) system may require several modifications. The fuel injectors need to be able to provide enough fuel to maintain the correct AFR. When the engine is running an high speeds and high boost pressures, it is possible the stock injectors will not be able to provide enough fuel to maintain the correct AFR. The stock injectors would then need to be replaced with larger injectors that could provide sufficient fuel. The fuel pump would also need to be able to provide enough power to move the necessary fuel to the injectors. While it should be able to do this, it is wise to test the fuel pump and make sure a larger one is not necessary.

The ignition system is another place where modification will likely be necessary. The stock spark plugs will need to be replaced with colder plugs. Trial and error is then necessary to find exactly what heat range works best. The plug gap may need to be shortened or some kind of voltage supplement may be necessary to ensure proper ignition under high boost pressures. The timing of the ignition can be controlled by the ECU to maximize power under boost.

The timing of the intake and exhaust valves in the engine cylinder should be considered. Turbocharged systems should have little valve overlap, time when both valves are open. During valve overlap, higher exhaust manifold pressures can force exhaust back into the cylinder. Reducing the valve overlap will help prevent this reversion. This can be accomplished by changing camshaft timing or replacing the cams themselves.

For anyone who wants to create serious boost pressures, lowering the stock compression ratio should be considered. Higher compression ratios mean that less boost can be produced before the engine starts to knock. Lowering the compression ratio will allow higher boost pressures without knocking. However, this is a very complicated procedure. Power gains should be sought elsewhere before lowering the stock compression ratio.

These are some of the major engine modifications that need to be looked into when turbocharging an engine. However, there are plenty more that should also be considered. Gaskets, seals, bearings, pistons, connecting rods and dozens of other components usually could be improved, though typically with no real tangible benefit. High boost pressures will require many of these more mundane things to be seriously looked at.

12 – Results and Conclusions

A 2001 Honda CBR600 F4i was purchased and bolted onto a custom engine stand. After extensive work the engine was fully prepared to run both mechanically and electrically and did in fact turn over several times. The engine did not run because after fully preparing the engine to run the team was informed that it would take an estimated 40 hours of tuning to get the engine running, which was more time than anyone was willing to put into it since there was no data acquisition system. There is currently still no means to obtain airflow or horsepower data from the engine. The team concluded that the engine was in good condition and would in fact run if given the proper tuning attention, but that lacking anyway to get data from the engine it was not worth the time needed to tune the engine, not when there was so much other work to do.

After an exhaustive search, the team really was not able to find much in the way of small engine displacement turbochargers. What little was found generally did not have proper documentation. Furthermore, without airflow data, the compressor could not properly be mapped. Since a turbocharger could not accurately be selected without a proper compressor mapping, the team decided that they could not justify spending the money to replace the current turbocharger. The Garrett GT12 was thus chosen by default.

The system layout was determined by sketching concepts on paper and then discussing them with the FSAE team. The finalized layout was completely modeled in SolidWorks inside of the FSAE chassis to finalize the detailed position of all of the components and plumbing. The final system layout is probably the best that could be done with very small space available on this particular chassis. Unfortunately it was very clear that this chassis was not designed with a turbocharger in mind, and as such some of the component placement and certainly the plumbing required some serious compromises in the name of packaging.

The intake manifold and entire intake system was almost completely designed and modeled in SolidWorks to be made of aluminum with silicone hoses connecting several of the key components. The team never received the FSAE throttle body and could not afford to purchase a new throttle body off the shelf, so that one particular component was not in fact completely designed. Otherwise the design was finished. Several of the components needed for the intake were machined on the CNC lathe, and all the necessary silicone hoses were purchased. A few parts still need to be finished being machined, and the runner pipes need to be purchased. Perhaps redesigning the runners to have larger bend radii may make it possible obtain the necessary pipes for less money than the current anticipated cost.

The intercooler was chosen to have an air/air core with dimensions of 15.1x6x2.25 inches. This core should be large enough to provide good heat transfer while limiting flow losses through the intercooler, and the air/air core added less weight to the chassis than the alternative air/water core. Custom end tanks were designed to maximize and equalize airflow into and out of the intercooler. Bell Intercoolers custom built the intercooler assembly for the team. A ducting concept was developed to maximize cooling airflow through the core but was never finalized or constructed.

The best way to control the boost of system while addressing the vacuum problem created by the restrictor would be to implement the design concept involving a computer controlled camshaft that compressed a spring to open the wastegate while allowing actuation to be linked to any engine sensor reading, like engine speed. However, the team decided it was not capable of implementing such a design. Therefore, the team went the next best option, a dual wastegate system. The integral wastegate on the GT12 would be sourced after the compressor to limit boost pressure to either 7psi or 18psi. A remote wastegate would be reverse actuated and sourced right after the restrictor to begin to crack at the first sign of a vacuum. This simple and purely mechanical design should effectively address the issues of excessive boost and vacuum formation but without the precision control of an electronic system like the camshaft. All of the necessary components for this system aside from the signal lines were purchased.

The exhaust system was completely designed in SolidWorks. Serious compromise had to be made to fit the system within the FSAE chassis. The exhaust manifold is a 4-1 equal length, individual runner design. Plumbing after the merge collector effectively, if not smoothly, connects the turbocharger, wastegate and muffler together. The custom flanges for the turbine were designed and successfully machined. The turbine inlet cone could not successfully be machined due to a lack of steel cutting tools in the machine shop. The pipes could not be obtained due to their very high price tag. Designing the manifold without the FSAE chassis as a constraint would allow for a simpler design with cheaper pipes. However, any manifold that fits in that engine compartment will likely require a mandrel bender and some serious money.

The turbocharger bearings will work well if they receive the right amount of oil flow from the engine. A water jacket filled by engine coolant hoses will eliminate the biggest concern in this system, which is oil coking. The next biggest concern can be addressed by providing a smooth, vertical drain hose into the engine oil pan, allowing the oil flow out of the bearings unobstructed. There is a big problem producing the necessary oil flange due to a design flaw with the GT12. A solution to the oil flange problems needs to be found before the remaining components can be purchased to complete the system.

There are dozens are potential engine modifications that need to be considered when adding a turbocharger to an engine. An aftermarket ECU and new MAP sensor will almost always be necessary. Most other modifications are linked to desired boost pressure. For a 7psi or less boost system, many of the modifications can probably be put off. Testing or trial and error will determine whether larger injectors or colder spark plugs will be necessary for proper ignition and a correct AFR. An 18psi system would require extensive modification, though. The fuel injectors would have to be replaced with larger ones to maintain the correct AFR. Colder spark plugs would definitely be needed, and a voltage supplement would probably be needed. The valve timing would certainly benefit from some alteration at this kind of performance level. The stock compression ratio would probably need to be lowered to prevent engine knocking. Finally, an assortment of pumps, seals, gaskets and the like would need to be tested under these rigorous conditions to determine whether or not they need to be replaced. The team concluded that it should try to get the turbocharged system running at 7psi without any engine modifications beyond the new ECU before getting into the hornet’s nest that would be required at 18psi boost.

Overall, the vast majority of the design work for the system was completed. The designed system meets all FSAE rules and could be placed on this year’s car were it physically completed. The total weight of the system was limited by opting for simpler intercooler and oil system designs that did not incorporate dedicated pumps, sumps, reservoirs, etc. Also the use of silicone hoses in the intake system helped reduce its weight. The only thing the team could do to increase power was to design the system to produce as much boost as possible. Ideally, much more could have been done to increase power of a broad engine speed range if only airflow and horsepower data could have been obtained from the engine with a restricted intake. Many of the components were purchased or manufactured, but the implementation fell short of being completed. Thus the turbocharged engine was not operational. Limited money prevented the team from obtaining the necessary pipes for manifolds or from buying a new throttle body. The team ran out of time trying to solve the oil flange problem. With more time and money, the implementation could have been completed, and the team is confident the turbocharged system would have worked had it been completed.

13 – Recommendations

The following are a series of recommendations agreed upon by team.

- Tune the naturally aspirated engine and get it running on an-engine stand with as much stock equipment as possible.
- Purchase the stock starter switch to help get the engine running quickly and avoid many of the problems experienced this year with engine's electrical systems.
- Test the stock unrestricted NA engine to acquire torque, horsepower and airflow data over the entire rev range.
- Replace the intake on the naturally aspirated engine and get it running with a restricted FSAE intake instead of the stock air box.
- Test the restricted NA engine to acquire torque, horsepower and airflow data over the entire rev range.
- Compare the torque, horsepower and airflow data over the entire rev range for the unrestricted and restricted NA engines to determine the effects of the intake restrictor on the NA engine's power output and air consumption.
- Modify the engine stand to turn it into a test stand that is capable of accommodating or housing testing equipment necessary to obtain data from the running engine.
- Acquire a data acquisition system by installing an airflow sensor on the intake pipes and finding a way to export data, possibly via the ECU, from the airflow sensor that can matched to corresponding engine speed data.
- Acquire an engine dynamometer to obtain torque and horsepower data for engine while it is running on that stand. A makeshift dynamometer could be designed and

built for the engine. Other alternatives are to find an engine dynamometer to purchase or rent time on.

- Contact Al Smyth of Portatree Timing Systems in Uxbridge, Massachusetts once the engine is ready to test. He may be able to find an engine dynamometer to rent time on.
- Purchase several pressure and temperature sensors and install them on the turbocharged engine system once it is operational. Use these sensors to acquire pressure and temperature data at several points in the turbocharged system. These points should include but are not limited to before and after the compressor and after the intercooler. Use this data to then try to improve the design.
- Once the turbocharged system is operational, put it on the engine dynamometer and test it for airflow, horsepower and torque data over the entire rev range.
- Use the airflow data from the turbocharged engine to determine at what RPM the restrictor chokes and how much airflow it actually permits at that point.
- Use all of the acquired data from the turbocharged system to assess design decisions and the effectiveness of the components. Attempt to improve the design based on the conclusions thereof.
- Find a better sized turbocharger for the restricted engine by using the acquired data to better size the compressor and turbine.
- Develop an ideal compressor map based on the airflow data that can be used to find a better sized compressor.
- Find a turbocharger that does not have an integral wastegate.
- Find a turbocharger that allows full 360° clocking of all of its sections.

- Find a turbocharger that has its oil inlet and outlet on opposite faces of the bearings section rather than side by side on the same face.
- Consider using the acquired data and knowledge about desirable turbocharger features to custom design a turbocharger for this application. This could possibly be an MQP.
- The FSAE chassis on which the turbocharged engine must be placed should be designed with the turbocharged system in mind.
- The chassis should be much bigger, particularly the area for the exhaust manifold.
- The turbocharger should be placed as high as possible in the chassis to facilitate a gravity drain oil system, even higher than it currently is would be advisable.
- The intake plumbing to the compressor from the restrictor should be in a straight line with no bends or cross section changes.
- The size of the opening above the diver's seat should be increased to provide sufficient space to duct the intercooler such that it receives enough ambient airflow.
- Either design or purchase a throttle body approximately 2 inches in diameter. A simple butterfly valve would probably be the best bet.
- Find and purchase a high-flow air filter rated for the engine's airflow and horsepower that will fit onto the throttle body.
- Finish machining the restrictor outlet cone using smaller and longer tools.
- Cut the extra pieces of stock off of the restrictor cones and weld them together at their throats.

- Determine how to connect the restrictor inlet to the throttle body. The working assumption of this team was that the restrictor would be welded to a flange for the throttle body.
- Attempt to improve the design of the restrictor via computer analysis of the flow through the restrictor.
- Machine the designed sleeve for the compressor to allow the 2 inch ID hoses to be clamped to it, or find a silicone reducer section that can be used to clamp onto the compressor.
- Assembled all of the silicone hoses together using the joiners and clamps, and clamp the assembled hoses to the proper components.
- Spend time trying to design a better plenum. Particularly, try to determine exactly what the plenum volume should be and try to find a way to create the desired bell mouth inlet and outlets.
- Machine the designed bell mouth inlet and outlets from the stock aluminum in the inventory and weld the designed plenum together.
- Spend time trying to determine the best length and diameter of the intake runners.
- Use computer analysis to try to maximize the flow through the plenum and runners.
- Purchase the necessary pipes for the designed intake runners.
- Cut the extra stock off the runner cones and weld them together with pipes to form the intake runners.
- Weld the runners to the plenum.
- Cut the extra stock off of the plenum cone and weld it to the plenum.

- Assembled all of the intake components to form a complete intake system.
- Perform a more detailed analysis air/air versus air/water intercoolers to determine which one is really the best choice for the system.
- Use the experimentally acquired airflow and horsepower data to size the intercooler core.
- If the layout of the intake system is changed, consider changing the design of the intercooler end tanks to facilitate plumbing.
- Finalize the design for the intercooler duct.
- Build the designed duct and attach it the intercooler using either glue or tape and a rubber strip.
- Complete the detailed design of the camshaft electronic wastegate system. This could be an MQP.
- Try to link the wastegate actuation to engine speed or electronically to pressure.
- Use acquired pressure data from the turbocharged engine to determine the best places to source the wastegates in the current dual wastegate design.
- Install the punch port wastegate signal ports in the appropriate locations on the silicone hoses.
- Buy the wastegate signal lines and their clamps.
- Replace the integral wastegate actuator on the GT12 with 18psi actuator.
- Test the wastegates to determine the cracking pressures and wide open pressures of each.
- Purchase the necessary pipes for the exhaust primaries and all the other exhaust plumbing.

- If redesigning the exhaust manifold, use an individual runner style manifold instead of a log style manifold.
- Attempt to design and tune a 4-2-1 manifold to replace the 4-1 manifold.
- Try to optimize the lengths of the primaries to foster scavenging.
- Use stainless steel as the material for all the exhaust pipes.
- Find a way to integrate the remote wastegate more smoothly. Use a shallow angle to split off from the main pipe and to merge the dump pipes.
- Purchase or custom produce gaskets for the turbine inlet and outlet.
- Determine whether the manifold will be able to support the weight of the turbocharger, wastegate and muffler. If it cannot, try redesigning it to support their weight or add some kind of supplemental support system for the components.
- Determine what the oil pressure is at the oil pressure sender. Based on the guidelines for oil pressure for the turbocharger, determine whether the oil pressure is too low or too high. If it is too low, then the oil pump may need to be replaced. If it is too high, an oil restrictor or pressure regulator of some kind will be needed to limit the excessive pressure.
- Determine how to solve the oil flange problem for the GT12. Either attempt to implement the design concept included in the oil section or try to come up with a completely different solution. It is important to recognize that the turbocharger cannot be operated with lubrication, so the system cannot run until this problem is resolved.

- Purchase appropriately sized oil feed and drain lines. Always use at least 0.5 inch ID hose for the drain line. Metal braided hose should really be used as well.
- Drill a hole in the oil pan and attach a connection for the oil drain line.
- Consider changing the type of oil in the engine to be more turbocharger friendly.
- Determine where the engine coolant is going to be diverted from for the bearings water jacket and determine where it is going to be returned.
- Purchase necessary pipe splitters and hoses for the coolant lines.
- Use the after market ECU to modify the ignition timing to maximize power output.
- Confirm that the ECU and EFI are able to maintain a correct AFR.
- Try to use the ECU to control the wastegate actuation.
- Build a new wiring harness for the aftermarket ECU.
- Hook up the 3-bar MAP, IAT, TPS and coolant sensors to the appropriate places.
- Use the anticipated boost pressure of the system to determine what engine modifications are necessary.
- Get the engine running at 7psi boost to make sure everything works. Then increase the boost pressure to 18psi and complete the necessary engine and system modifications. Get the engine running at 18psi boost and attempt to optimize.
- Buy larger fuel injectors for 18psi boost system and test them to make sure they can supply enough fuel to maintain the correct AFR at maximum boost.
- Test the fuel pump to make sure it can pump enough fuel to the injectors under boost.

- Make sure that the injectors operate at a duty cycle below 80%. If they do not, then even larger injectors will be necessary.
- Make sure the engine does not run lean or rich.
- Use the highest octane gas available.
- Look into using E85 as an alternative to gasoline.
- Add a knock sensor to engine to detect detonation.
- Replace the stock spark plugs with colder ones and modify the heat range based on trial and error.
- Make sure that the voltage is sufficient to spark under maximum boost. A voltage supplement may be needed.
- Replace the stock cams or alter the camshaft timing to prevent reversion.
- Lower the stock compression ratio to allow higher boost pressures.
- Fully inspect the engine block and components for any damage prior to running the high boost system.
- Determine the maximum boost that the pistons, con rods, gaskets and seals can handle.
- Once the final turbocharged system has been tested and optimized on the stand, install it on the FSAE chassis. Test the chassis on a dynamometer to compare NA and turbocharged power curves. Also complete road testing.
- Start the process of learning how to machine in B or C term, since the shops are often in use and it is difficult to learn how to machine and get it all done in a single term.
- Talk to Neil Whitehouse for assistance machining. He is extremely helpful.

- Check the available tools in the shop well before the anticipated machining date. The currently available tools really leave something to be desired, so special tool may need to be ordered to machine all the parts. This is especially true for steel parts.
- Really stress manufacturability and what is commercially available early on in the design process. Do some background research into these areas before starting to design the system. It will save a lot of time later on and will increase the odds of developing an operational system.
- Locate a work area in which productive work on the project can be accomplished without distraction or interference. Also try to find a place in which the equipment is secure enough that things will not be taken off of the engine or cut to pieces mysteriously.
- Model all the components in Solidworks and create an assembly to work on the system layout.
- Model parts to be machined in SolidWorks and import them into GibbsCAM to write the tool paths.
- Find computer analysis programs that can be used to predict and analyze airflows, temperatures, stresses, etc. for the designed system.
- After thoroughly reading this report, use Appendix M to conduct an inventory of the components. Reuse whatever can be reused, and try to come up with a budget for any other necessary components.
- Develop a schedule and try to stick with it.

- Try to get the turbocharged engine running as designed in the project seeing as so much of the work has already been done. It will then be easier to try to improve parts of the design.

Comments

This was a really interesting project that provided a really educational experience in engineering. Working through the entire design process from initial concepts to detailed models to completed parts was a particularly valuable experience. The importance of considering manufacturability in preliminary designs is made much clearer after going through some of the problems that were encountered in machining. The necessity of making tough design decisions that balance performance with attainability was also something that is obvious.

It is unfortunate and very frustrating the turbocharged engine was not completed. There are probably two major reasons why the team was unable to finish the turbocharged engine. The first is that the NA engine monopolized so much of the team’s time. For the first couple months of the project, it was the top priority of the entire team. Even after a few months, one of the three people in the team was still spending all his time on the NA engine. Having to spend so much time on the NA engine really cut short the amount of time the team had to work on the turbocharged system. The other big problem was having to design the system to go on this year’s FSAE chassis, which was clearly never intended to have a turbocharger on it based on its very small size. The manifold and plumbing in particular became extremely tight and expensive as a result of the packaging issues created by the chassis size. Had the goal been to design a turbocharged system with a restricted intake that would just run on the stand, these packaging problems and expensive manifolds would not have been such an obstacle.

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<http://en.wikipedia.org/w/index.php?title=Internal_combustion_engine&oldid=207769666>.

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Rules & Important Documents Formula SAE. 24 Apr 2008. <http://students.sae.org/competitions/formulaseries/rules/>

Appendix A – Relevant FSAE rules

The following is a list of FSAE rules quoted from the 2008 FSAE rule book available at the FSAE website (http://students.sae.org/competitions/formulaseries/rules/). These rules were deemed relevant to turbocharging either directly or indirectly.

3.2.2 Ground Clearance

The ground clearance must be sufficient to prevent any portion of the car (other than tires) from touching the ground during track events, and with the driver aboard there must be a minimum of 25.4 mm (1 inch) of static ground clearance under the complete car at all times.

3.5.1.1 Engine Limitations

The engine(s) used to power the car must be four-stroke piston engine(s) with a displacement not exceeding 610 cc per cycle. The engine can be modified within the restrictions of the rules. If more than one engine is used, the total displacement can not exceed 610 cc and the air for all engines must pass through a single air intake restrictor (see 3.5.4.3, "Intake System Restrictor.")
Hybrid powertrains utilizing on-board energy storage are not allowed.

3.5.1.6 Coolant Fluid Limitations

Water-cooled engines must only use plain water, or water with cooling system rust and corrosion inhibitor at no more than 0.015 liters per liter of plain water. Glycolbased antifreeze or water pump lubricants of any kind are strictly prohibited.

3.5.2 Fuels

The basic fuel available at competitions in the Formula SAE Series is unleaded gasoline with an octane rating of 93 (R+M)/2 (approximately 98 RON). Other fuels may be available at the discretion of the organizing body.
Unless otherwise announced by the individual organizing body, the fuel at competitions in the Formula SAE Series will be provided by the organizer.
During all performance events the cars must be operated with the fuels provided by the organizer at the competition.
Nothing may be added to the provided fuels. This prohibition includes nitrous oxide or any other oxidizing agent.
Teams are advised that the fuel supplied in the United States is subject to various federal and state regulations and may contain up to ten percent (10%) ethanol. The exact chemical composition and physical characteristics of the available fuel may not be known prior to the competition.
Consult the individual competition websites for fuel types and other information.

3.5.2.1 Fuel Temperature Changes – Prohibited

The temperature of fuel introduced into the fuel system may not be changed with the intent to improve calculated fuel economy.

3.5.2.2 Fuel Additives – Prohibited

No agents other than fuel (gasoline or E85), and air may be induced into the combustion chamber. Non-adherence to this rule will be reason for disqualification. Officials have the right to inspect the oil.

3.5.3.8 Fuel Injection System Requirement

The following requirements apply to fuel injection systems.
A. Fuel Lines – Flexible fuel lines must be either (i) metal braided hose with either crimped-on or reusable, threaded fittings, or (ii) reinforced rubber hose with some form of abrasion resistant protection with fuel line clamps per 3.5.3.7. Note: Hose clamps over metal braided hose will not be accepted.
B. Fuel Rail – The fuel rail must be securely attached to the engine cylinder block, cylinder head, or intake manifold with brackets and mechanical fasteners. This precludes the use of hose clamps, plastic ties, or safety wire.
C. Intake Manifold – The intake manifold must be securely attached to the engine block or cylinder head with brackets and mechanical fasteners. This precludes the use of hose clamps, plastic ties, or safety wires. The use of rubber bushings or hose is acceptable for creating and sealing air passages, but is not considered a structural attachment.

3.5.3.9 Air Intake and Fuel System Location Requirements

All parts of the fuel storage and supply system, and all parts of the engine air and fuel control systems (including the throttle or carburetor, and the complete air intake system, including the air cleaner and any air boxes) must lie within the surface defined by the top of the roll bar and the outside edge of the four tires (see figure 8).[Fig. A.1]
All fuel tanks must be shielded from side impact collisions. Any fuel tank which is located outside the Side Impact Structure required by 3.3.8, must be shielded by structure built to 3.3.8. A firewall must also be incorporated, per section 3.4.10.1. Any portion of the air intake system that is less than 350 mm (13.8 inches) above the ground must be shielded by structure built to 3.3.8.

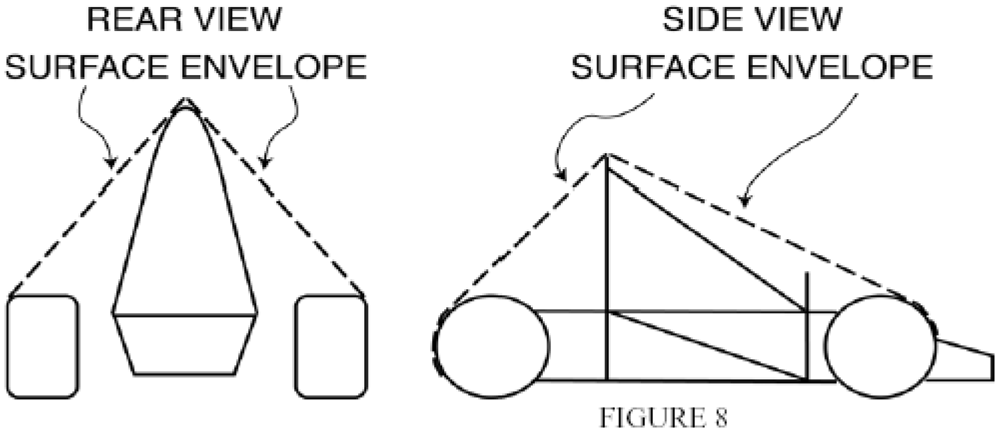


Figure A.1 FSAE roll envelope.

3.5.4.2 Throttle Actuation

The throttle must be actuated mechanically, i.e. via a cable or a rod system. The use of electronic throttle control (ETC) or “drive-by-wire” is prohibited.
The throttle cable or rod must have smooth operation, and must not have the possibility of binding or sticking.
The throttle actuation system must use at least two (2) return springs located at the throttle body, so that the failure of any component of the throttle system will not prevent the throttle returning to the closed position.
Note: Throttle Position Sensors (TPS) are NOT acceptable as return springs.

Throttle cables must be at least 50.8 mm (2 inches) from any exhaust system component and out of the exhaust stream.
A positive pedal stop must be incorporated on the throttle pedal to prevent over stressing the throttle cable or actuation system.
The use of a push-pull type throttle cable with a throttle pedal that is capable of forcing the throttle closed (e.g. toe strap) is recommended.

3.5.4.3 Intake System Restrictor

In order to limit the power capability from the engine, a single circular restrictor must be placed in the intake system between the throttle and the engine and all engine airflow must pass through the restrictor. Any device that has the ability to throttle the engine downstream of the restrictor is prohibited.
The maximum restrictor diameters are:
- Gasoline fueled cars - 20.0 mm (0.7874 inch)
- E-85 fueled cars – 19.0 mm (0.7480 inch)

The restrictor must be located to facilitate measurement during the inspection process. The circular restricting cross section may NOT be movable or flexible in any way, e.g. the restrictor may not be part of the movable portion of a barrel throttle body.
If more than one engine is used, the intake air for all engines must pass through the

one restrictor.

3.5.4.4 Turbochargers & Superchargers

Turbochargers or superchargers are allowed if the competition team designs the application. Engines that have been designed for and originally come equipped with a turbocharger are not allowed to compete with the turbo installed.
The restrictor must be placed upstream of the compressor but after the carburetor or throttle valve. Thus, the only sequence allowed is throttle, restrictor, compressor, engine.
The intake air may be cooled with an intercooler (a charge air cooler). Only ambient air may be used to remove heat from the intercooler system. Air-to-air and water-to-air intercoolers are permitted. The coolant of a water-to-air intercooler system must comply with Rule 3.5.1.6.

3.5.5.1 Muffler

The car must be equipped with a muffler in the exhaust system to reduce the noise to an acceptable level.

3.5.5.2 Exhaust Outlet

The exhaust must be routed so that the driver is not subjected to fumes at any speed considering the draft of the car.
The exhaust outlet(s) must not extend more than 60 cm (23.6 inches) behind the centerline of the rear axle, and shall be no more than 60 cm (23.6 inches) above the ground.
Any exhaust components (headers, mufflers, etc.) that protrude from the side of the body in front of the main roll hoop must be shielded to prevent contact by persons approaching the car or a driver exiting the car.

Appendix B – Purchases

The following table contains a list summarizing all of the team's purchases. The listed cost is the actual cost of the item and does not contain shipping and handling or applicable taxes.

Item Number	Item Description	Vendor	Item Cost	Quantity
1	2001 Honda CBR600 F4i motorcycle engine	Cycle Pros Salvage	$699.99	1
2	2001 Stock Engine Harness (CBR600 F4i)	Drumhill Cycle	$30	1
3	2001 Stock Throttle Body (CBR600 F4i)	Drumhill Cycle	$300	1
4	2001 Stock Fuel Pump (CBR600 F4i)	Drumhill Cycle	$150	1
5	2001 Stock Air Box (CBR600 F4i)	Drumhill Cycle	$25[1]	1
6	Complete 2001 Stock Wiring Harness (CBR600 F4i)	Argo Cycles and Auto	$120	1
7	Miscellaneous Stock Gaskets, Banjo Bolts, Nuts, & Hoses (CBR600 F4i)	Honda	$550	1
8	2001-04 User's Manual (CBR600 F4i)	EBay	$36	1
9	2001 Stock Exhaust Manifold (CBR600 F4i)	EBay	$65	1
10	2001 Stock Starter Relay (CBR600 F41)	EBay (Richard Cox)	$36	1
11	12V Battery	Wal-Mart	$40[2]	1
12	Custom Steel Engine Stand (material and labor)	Merchants Sheet Metal	$800[3]	1
13	Custom Steel Battery Box (material and labor)	Merchants Sheet Metal	$100[3]	1
14	Custom Aluminum Fuel Tank (material and labor)	Merchants Sheet Metal	$300[3]	1
15	TiAL 46mm Remote Wastegate	Full Race	$449.00	1
16	TiAL 46mm Wastegate Inlet Flange	Full Race	$35.00	1
17	TiAL 46mm Wastegate Dump Flange	Full Race	$35.00	1
18	TiAL 2.9psi Wastegate Spring	Full Race	$29.50	1

19	18psi Wastegate Actuator	ATP Turbo	$59.00	1
20	PE-ECU-1 Aftermarket ECU Package Including Harness	Performance Electronics	$998.00[4]	1
21	3-bar MAP Sensor	Performance Electronics	$90.00	1
22	3-bar MAP Sensor Connector	Performance Electronics	$5.00	1
23	Air Temperature (IAT) Sensor	Performance Electronics	$30.00	1
24	Air Temperature (IAT) Sensor Connector	Performance Electronics	$5.00	1
25	4-1 Merge Collector, 1.375 inch Primaries	S&S Headers	$83.50	1
26	Aluminum 6061, 3 feet of 2 inch Round Stock	McMaster-Carr	$57.52	1
27	Aluminum 6061, 3 inches of 4.5 inch Round Stock	McMaster-Carr	$39.97	1
28	Aluminum 6061, 1 foot of 3.5 inch Round Stock	McMaster-Carr	$56.86	1
29	Steel 4140, 1 foot of 2.125 inch Round Stock	McMaster-Carr	$39.13	1
31	Steel 4140, 12x4x0.5 inch Rectangular Bar Stock	McMaster-Carr	$28.30	1
32	Pipe Tee, 10mm	C.A.P.	$3.65	1
33	Threaded Male/Male Adapter, 10mm	C.A.P.	$1.28	3
34	Hex Nut, 8mm, 2 pack	Home Depot	$0.32	4
35	Flat Washer, 8mm, 4 pack	Home Depot	$0.42	2
36	Hex Bolt, 6mm x 40mm	Home Depot	$0.40	2
37	Flat Washer, 6mm, 4 pack	Home Depot	$0.38	1
38	Threaded Male/Male Adapter, 0.5 inch	True Value	$0.79	1
39	Quick Tap Wastegate Signal Ports	ATP Turbo	$19.50	2
40	Stainless Steel Joiner for 2 inch Silicone Hoses	Silicone Intakes	$7.99	1
41	Clamp for 2 inch Silicone Hose	Silicone Intakes	$2.60	4
42	2 inch 45° Silicone Elbow Hose	Silicone Intakes	$16.99	2
43	1.5 inch 45° Silicone Elbow Hose	Silicone Intakes	$16.99	2
44	1.5 inch 90° Silicone Elbow Hose	Silicone Intakes	$16.99	3
45	2 feet of 1.5 inch Straight Silicone Hose	Silicone Intakes	$39.99	1
46	Clamp for 1.5 inch Silicone Hose	Silicone Intakes	$2.60	18
47	Type 304 Stainless Steel Pipe, 3 feet of 1.38 inch ID/1.50 inch OD	McMaster-Carr	$18.64	1

48	Air/Air Intercooler with Custom End Tanks, 15.1x6x2.25 inch Core	Bell Intercoolers	$434.00[5]	1
49	Turbine and Oil Flanges (labor)	Merchants Sheet Metal	$300[3]	1

Notes:
1. Donated by JohnPaul Piccolomini.
2. Donated by David Lamoureux, Jr.
3. Donated by David Lamoureux, Sr. and Ronnie Withers of Merchants Sheet Metal.
4. Performance Electronics provided a 20% discount. Final price was $798.40.
5. Bell Intercoolers provided a 20% discount. Final price was $347.20.

Table B.1 Summary of purchases.

Appendix C – Pictures of Engine and Stand

The following are pictures of the naturally aspirated Honda CBR600 F4i on the custom built engine stand.

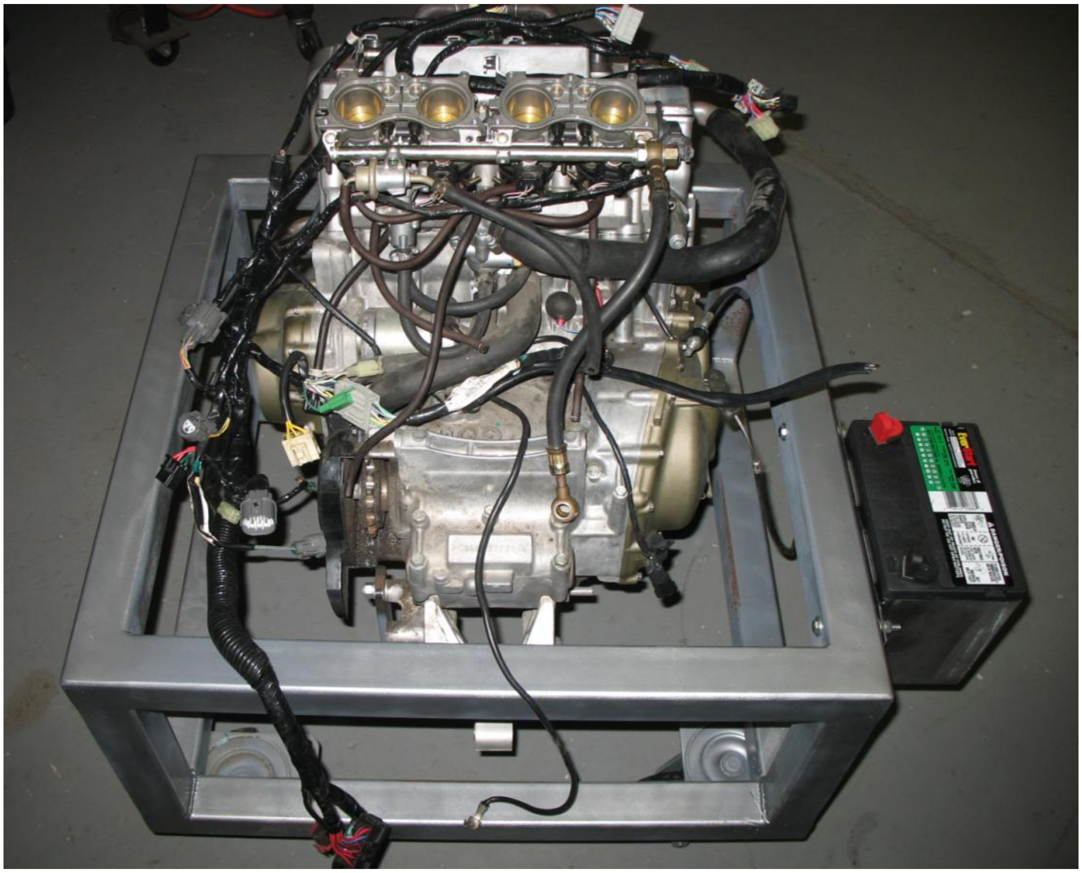


Figure C.1 Engine in stand, view 1.

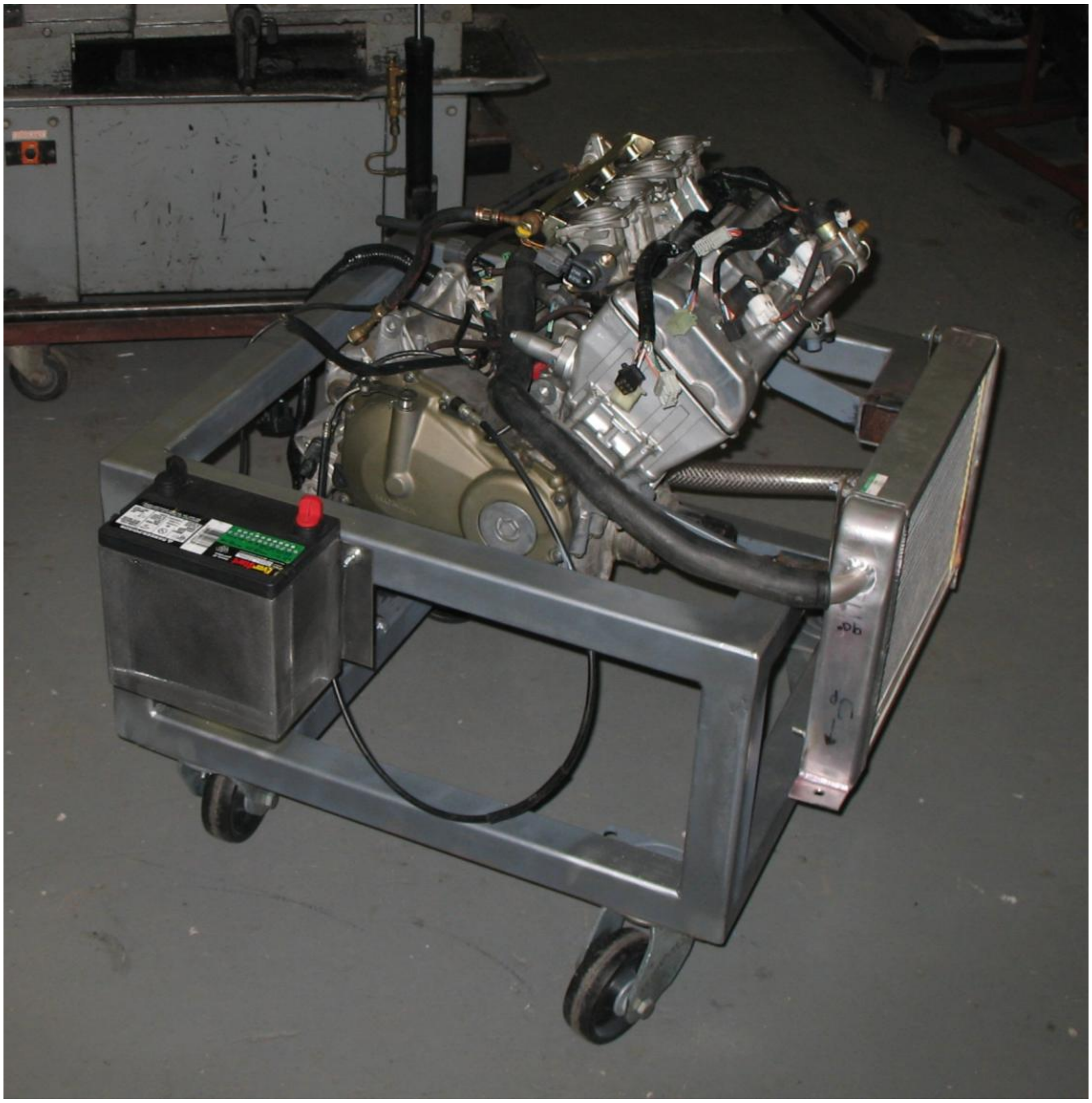


Figure C.2 Engine in stand, view 2.

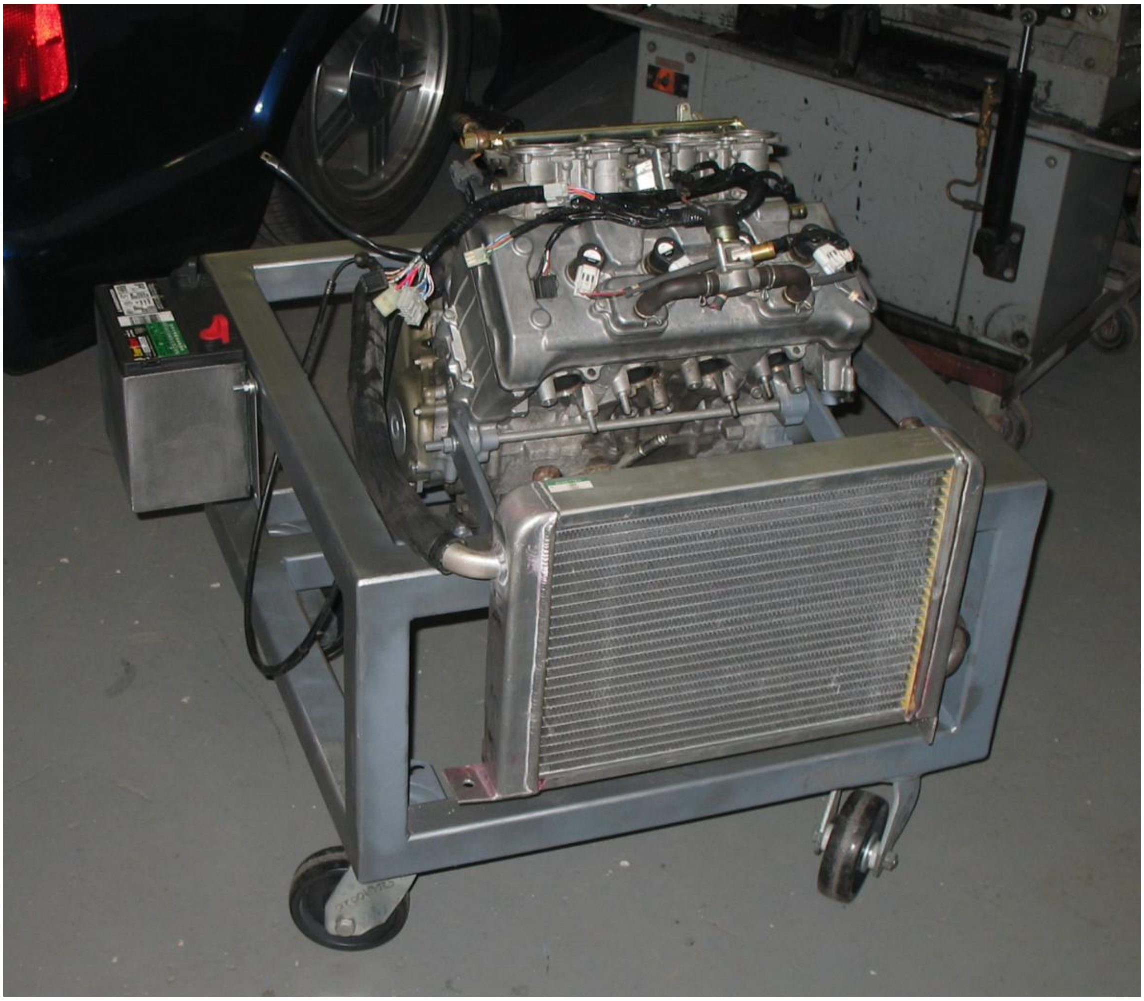


Figure C.3 Engine in stand, view 3.

Appendix D – Garrett GT12 Turbocharger

Detailed pictures, the compressor map and the turbine map for the GT12 are provided.

Figure D.1 GT12 Turbo, view 1.

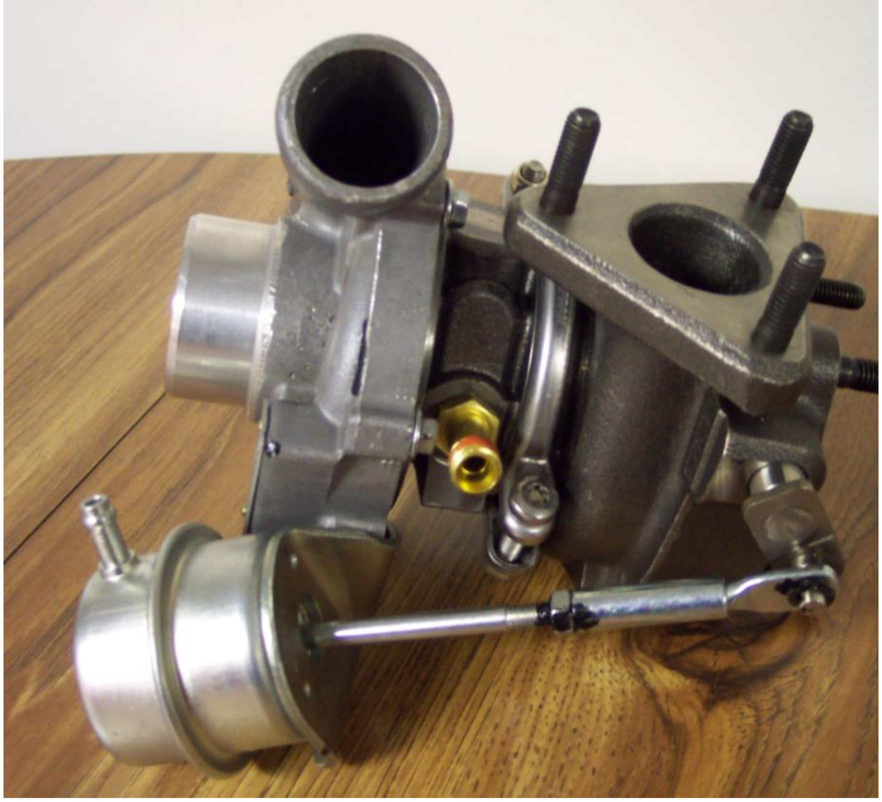


Figure D.2 GT12 Turbo, view 2.

Figure D.3 GT12 Turbo, compressor inlet.

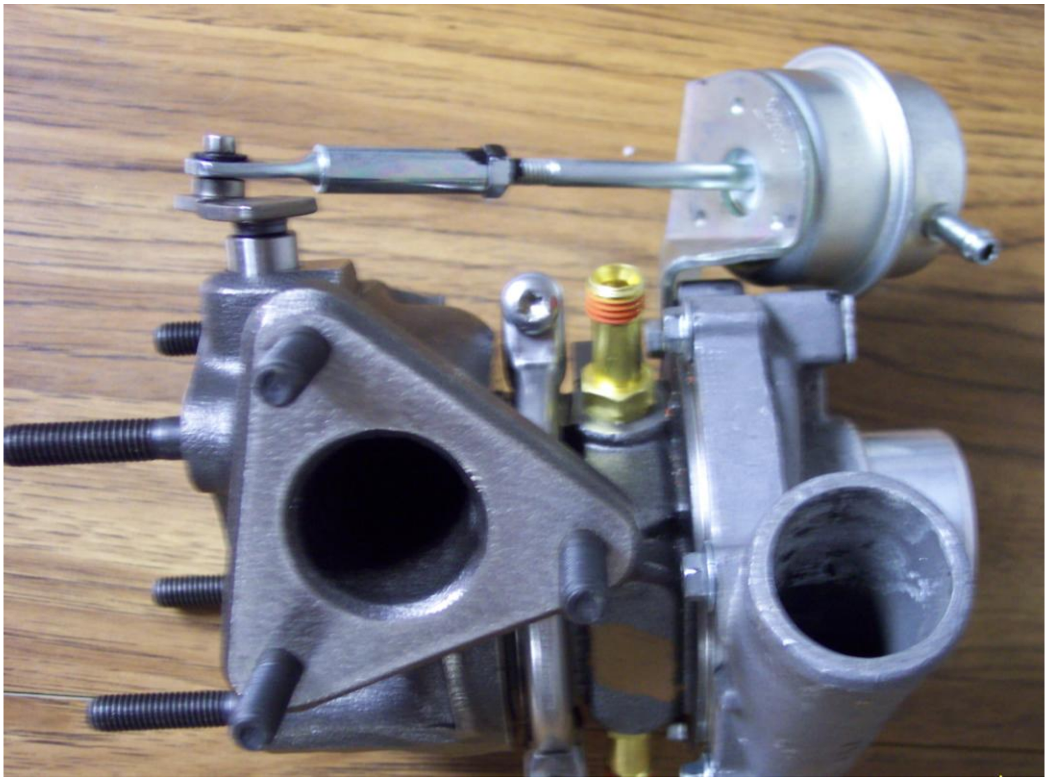


Figure D.4 GT12 Turbo, compressor outlet (right) and turbine inlet (left).

Figure D.5 GT12 Turbo, turbine and wastegate outlets.

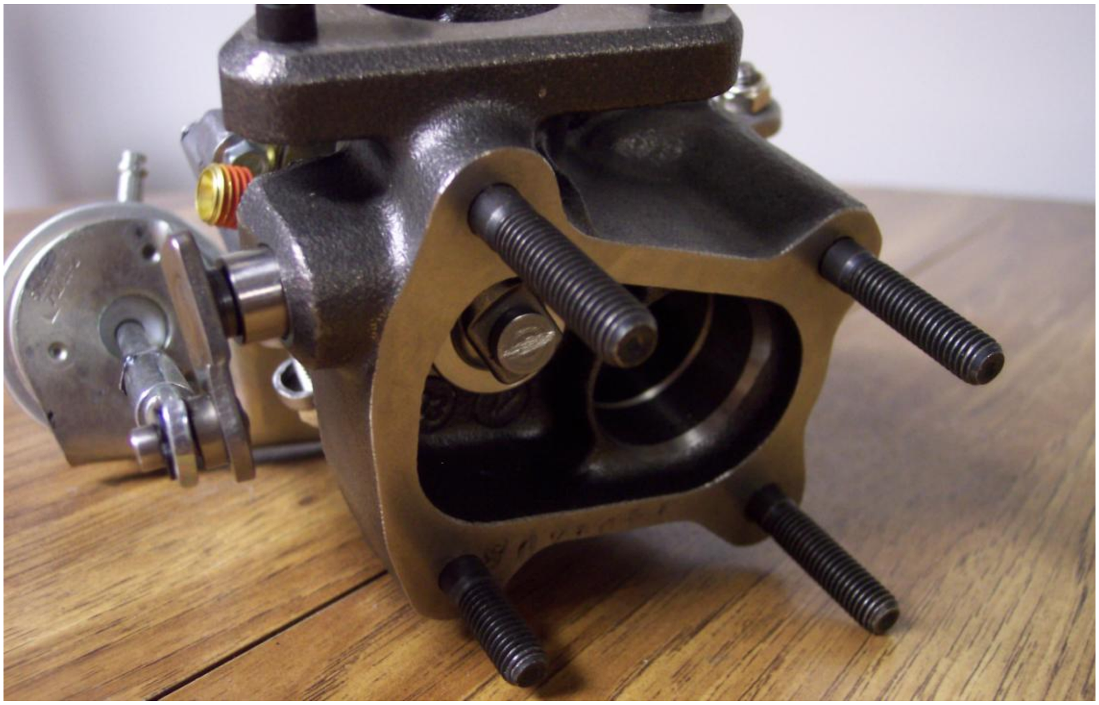


Figure D.6 Gt12 Turbo, wastegate linkage to outlet.

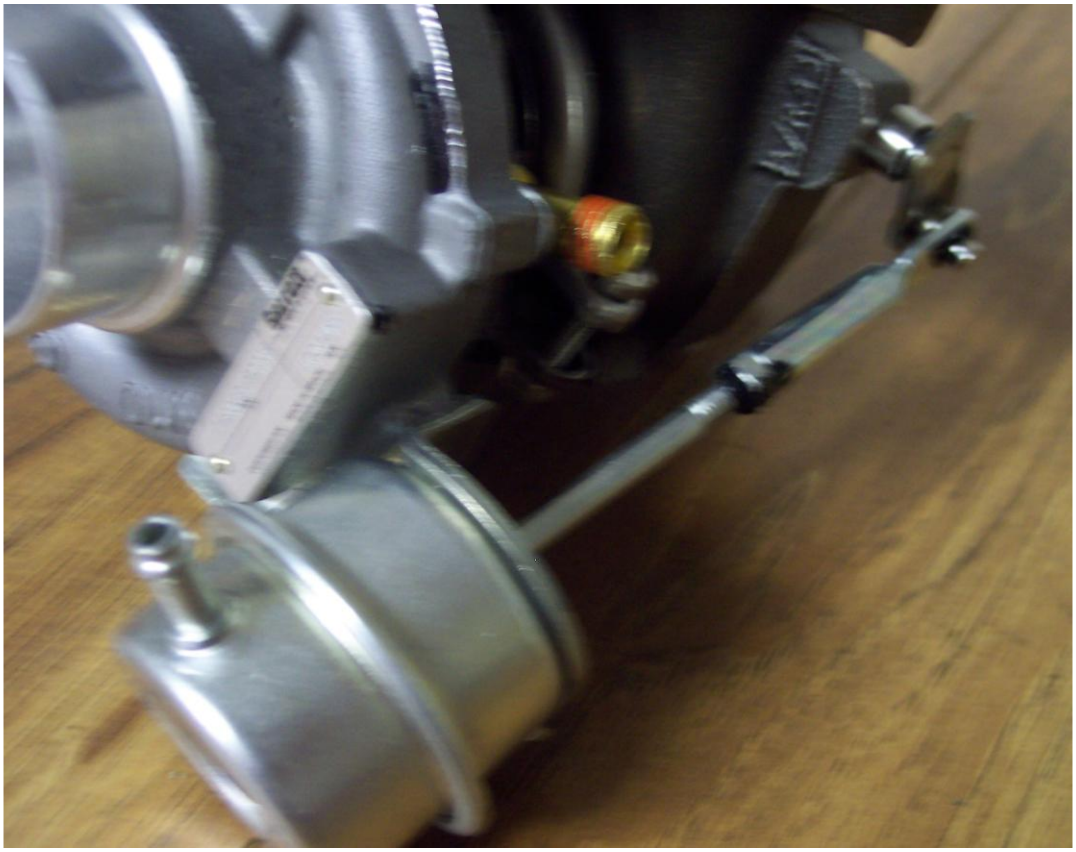


Figure D.7 GT12 Turbo, wastegate actuator.

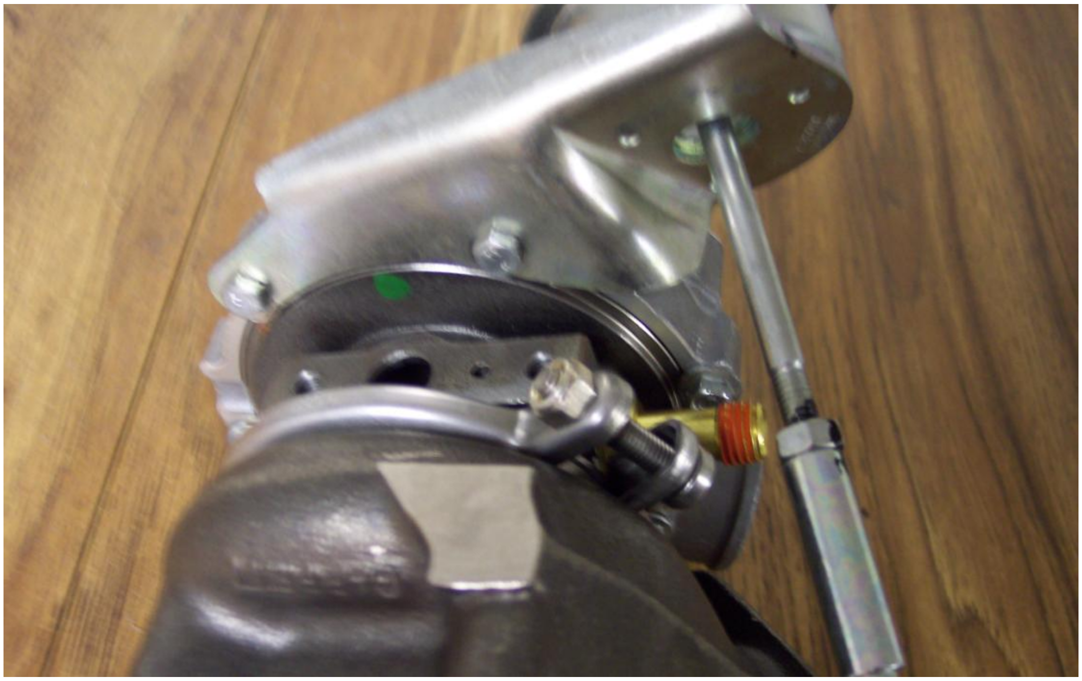


Figure D.8 GT12 Turbo, wastegate housing bolted to compressor.

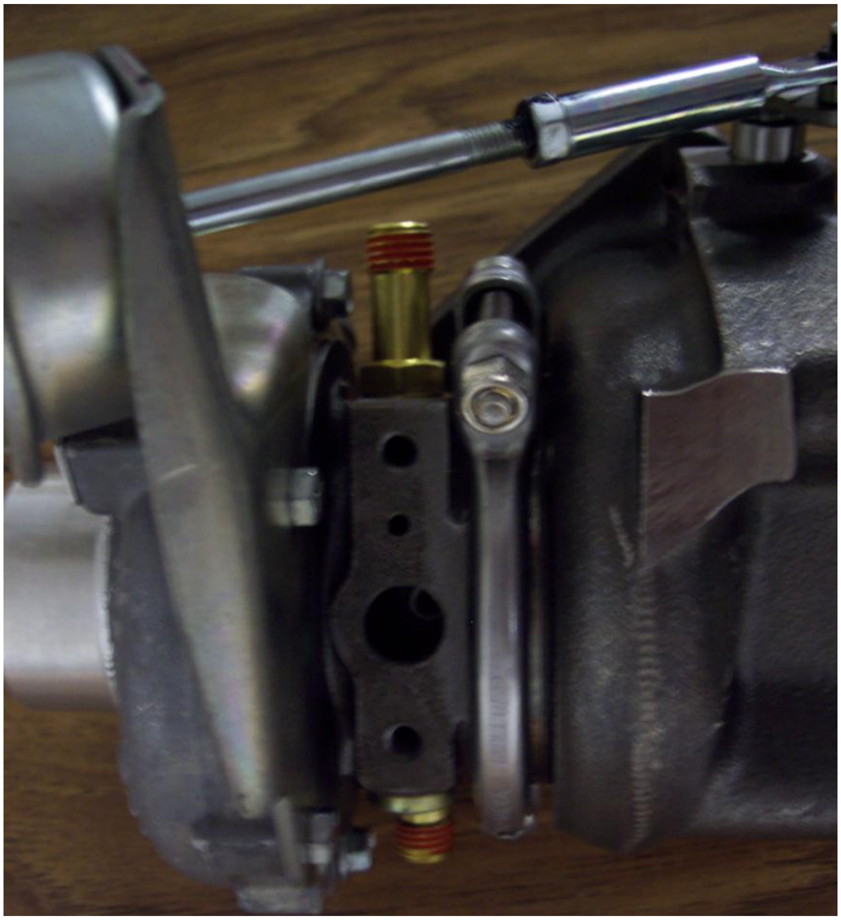


Figure D.9 GT12 Turbo, oil inlet and outlet.

Figure D.10 GT12 Turbo, coolant ports from both sides of bearings.

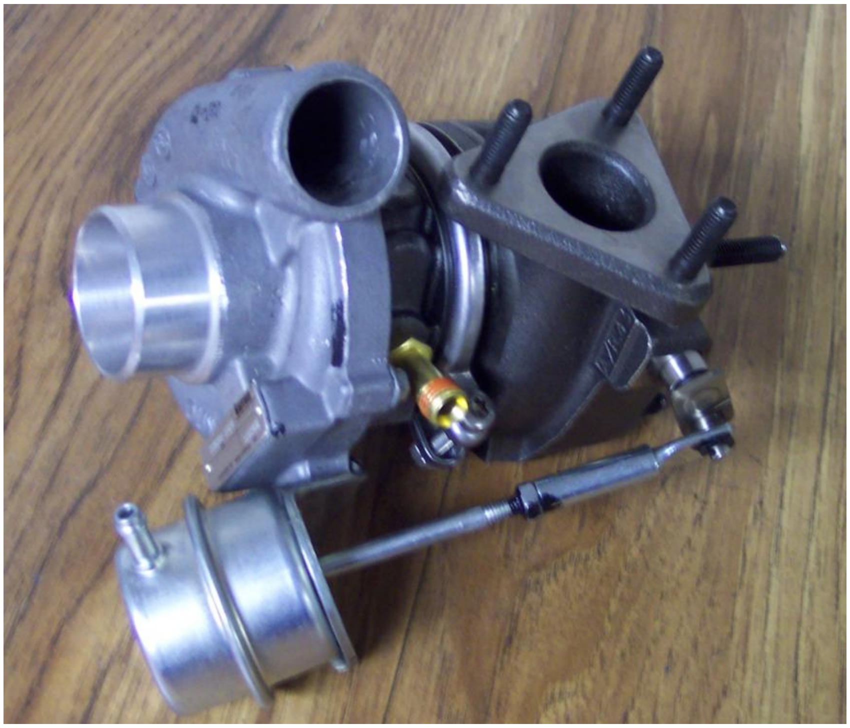


Figure D.11 GT12 Turbo, overview.

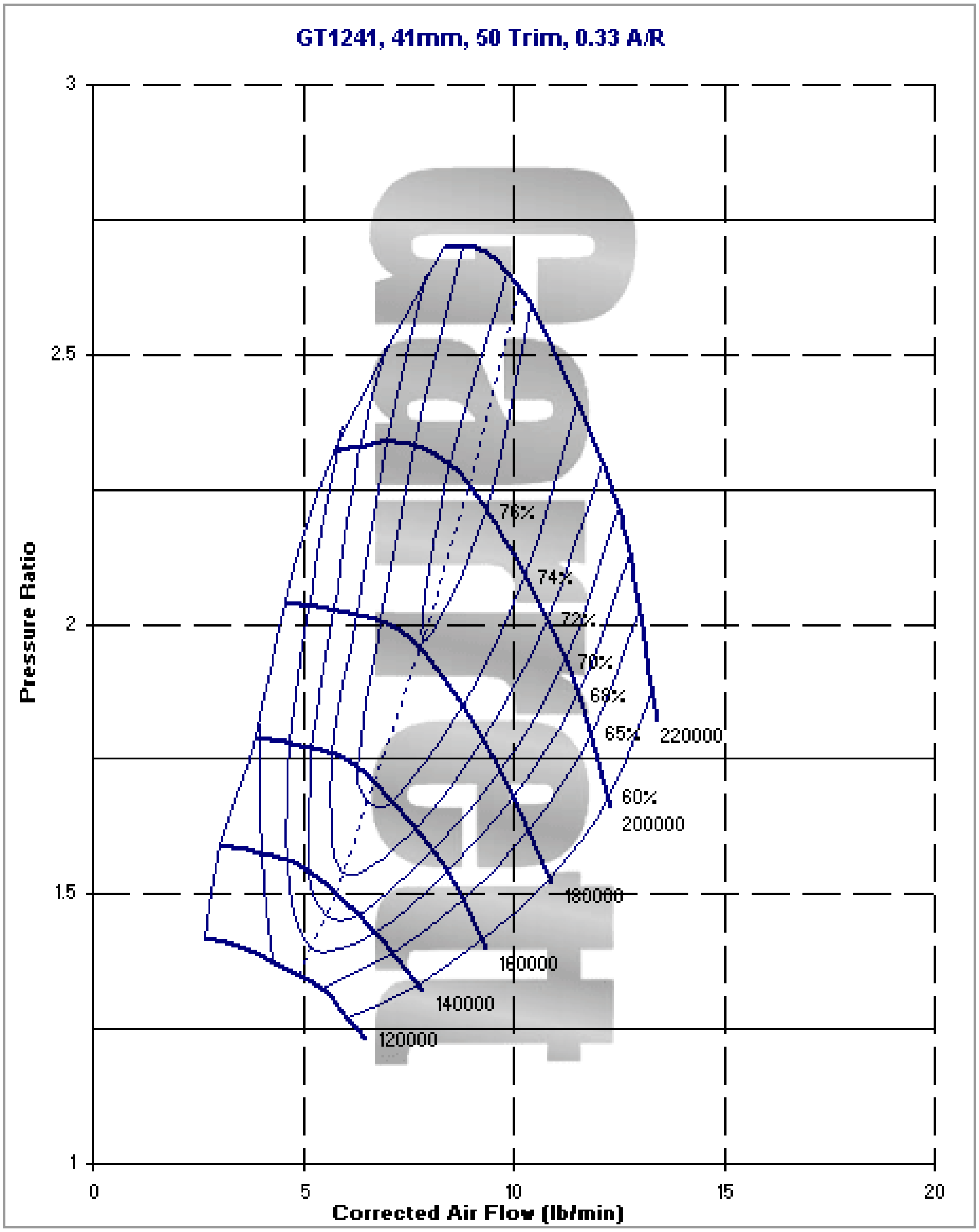


Figure D.12 GT12 Turbo, compressor map. (Retrieved from http://www.turbobygarrett.com/turbobygarrett/catelog/Turbochargers/GT12/GT1241_756068__1.htm on April 23, 2008)

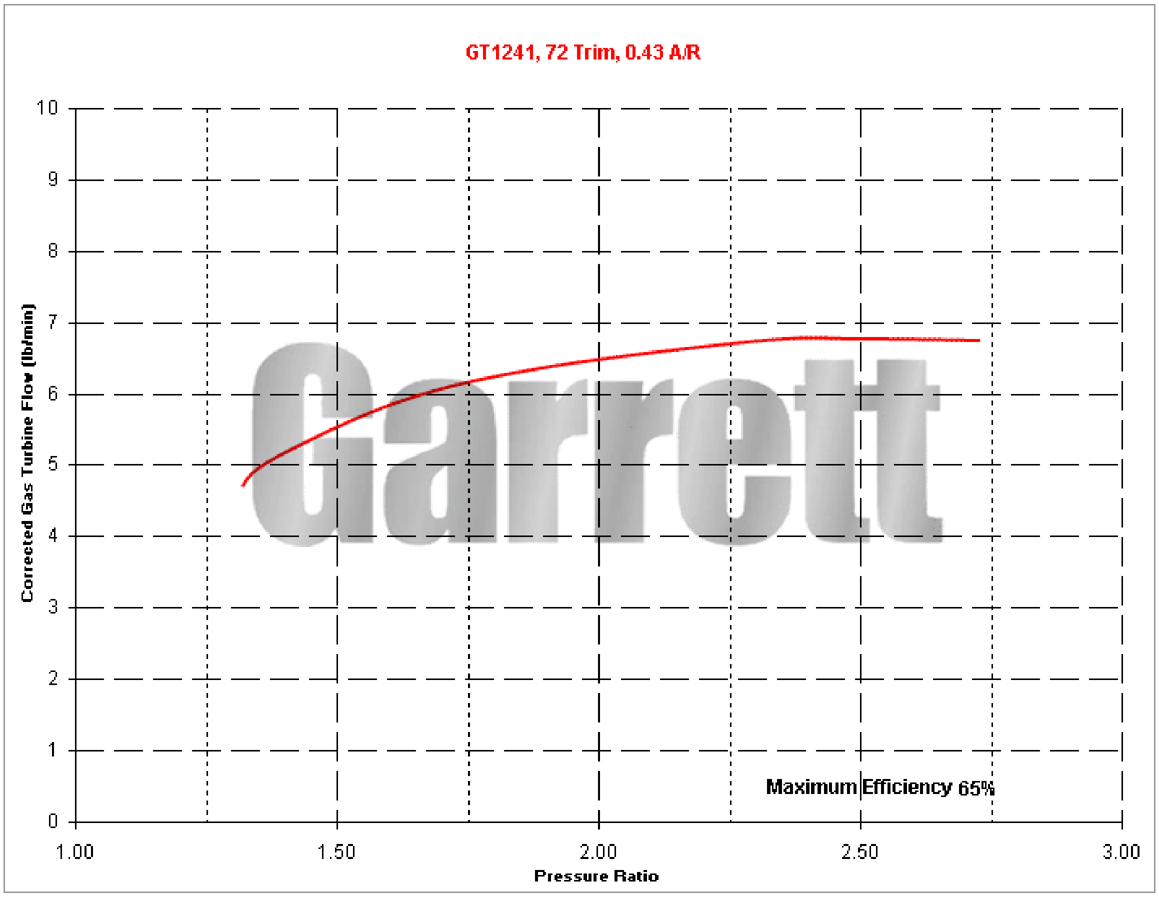


Figure D.13 GT12 Turbo, turbine map. (Retrieved from http://www.turbobygarrett.com/turbobygarrett/catelog/Turbochargers/GT12/GT1241_756068__1.htm on April 23, 2008)

Appendix E – Final System Model

The following figures are screenshots of the final system model from SolidWorks. The chassis and engine were modeled by the FSAE team. All other components and plumbing were modeled and assembled by David Curran.

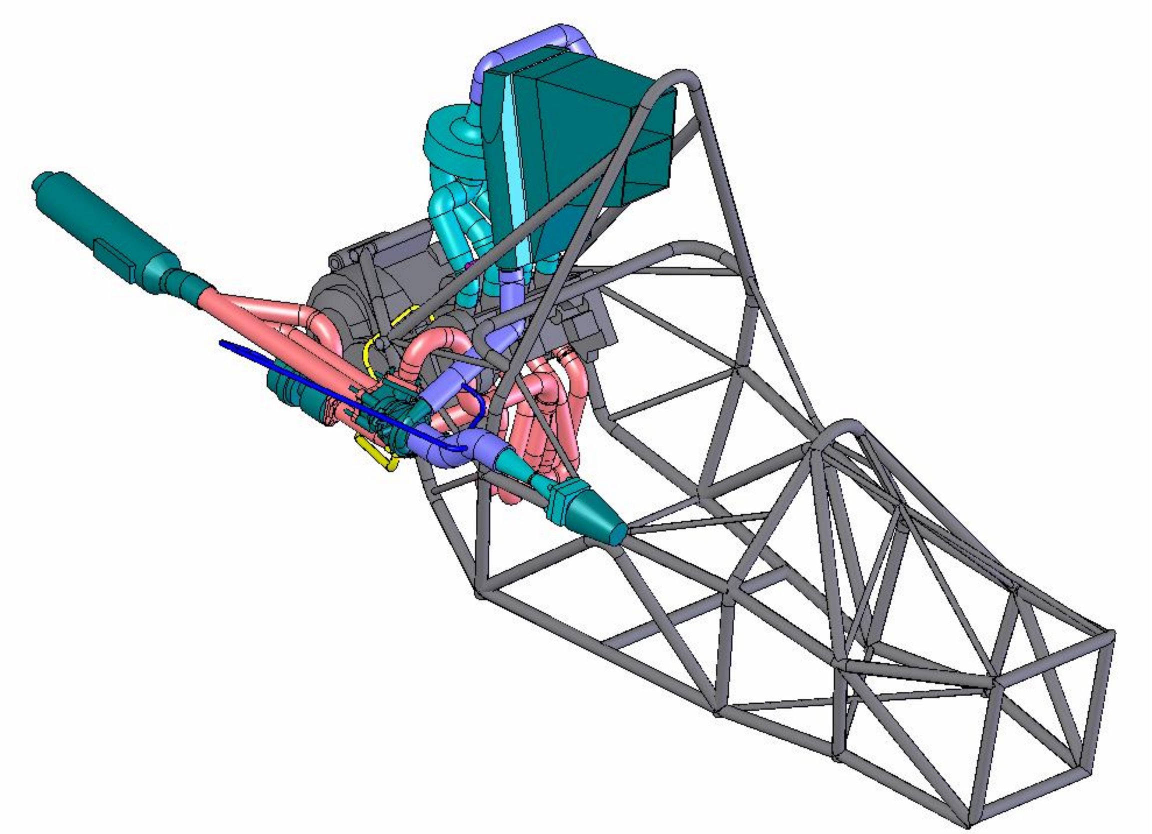


Figure E.1 Final system model, Project Presentation Day view.

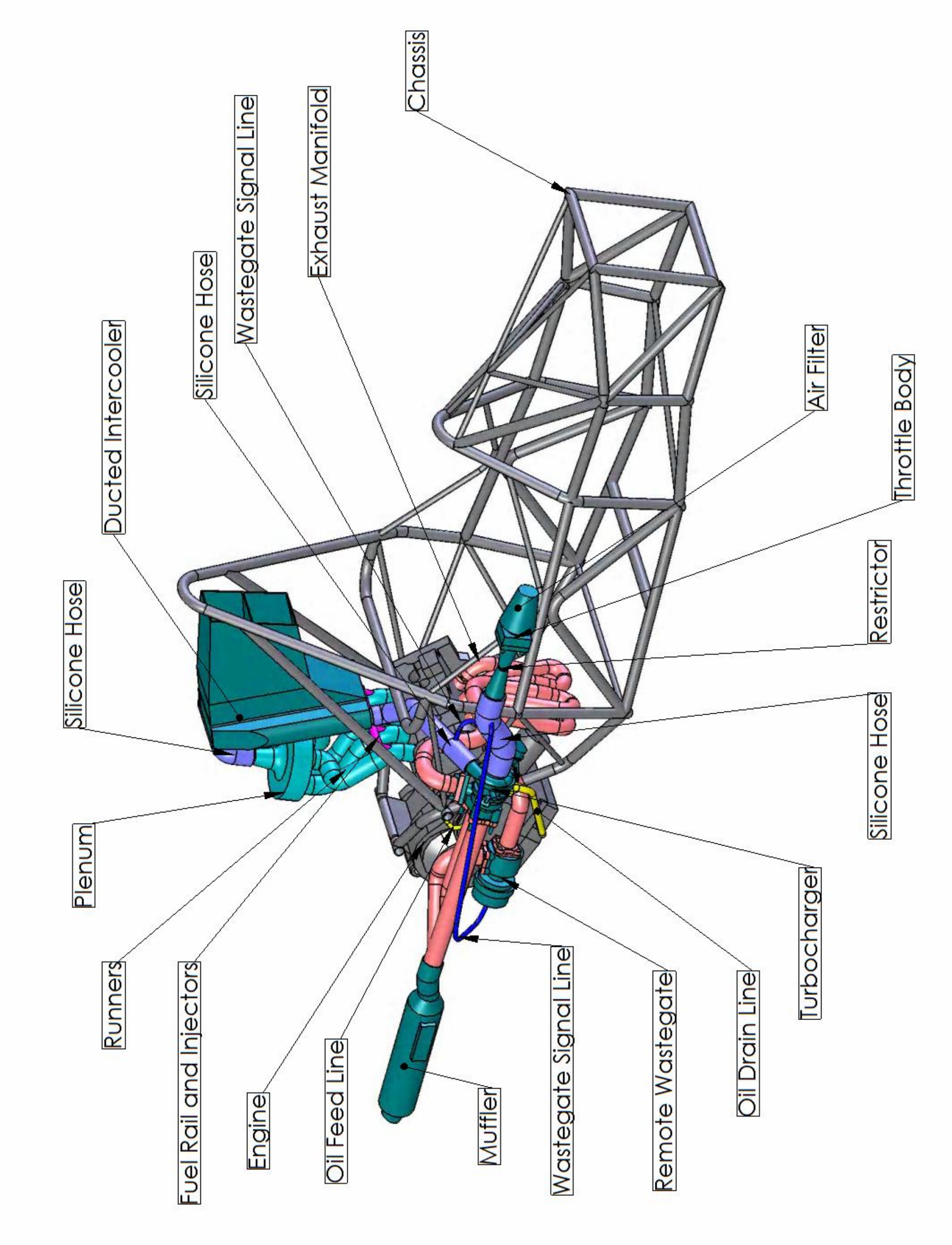


Figure E.2 Final system model, components labeled.

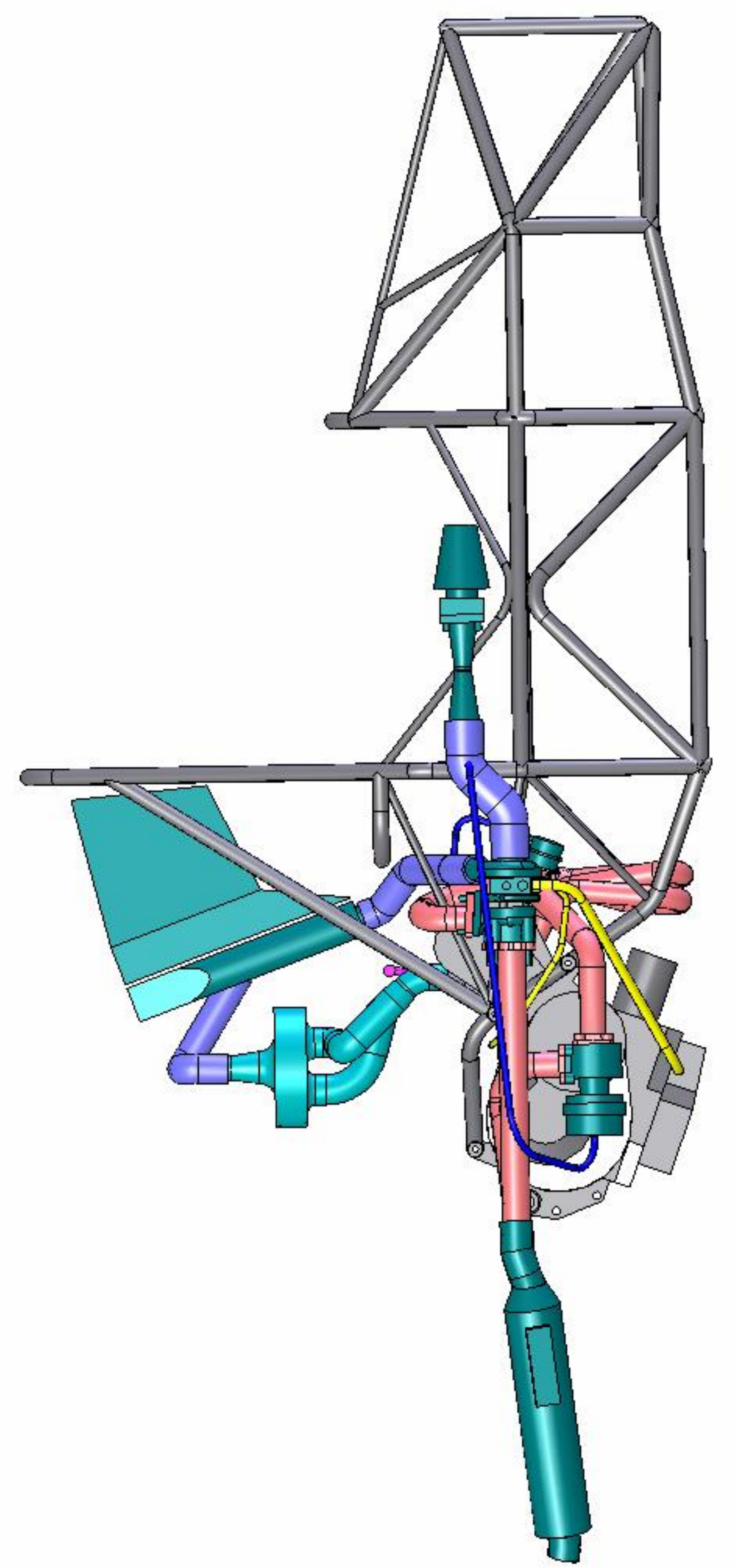


Figure E.3 Final system model, right view.

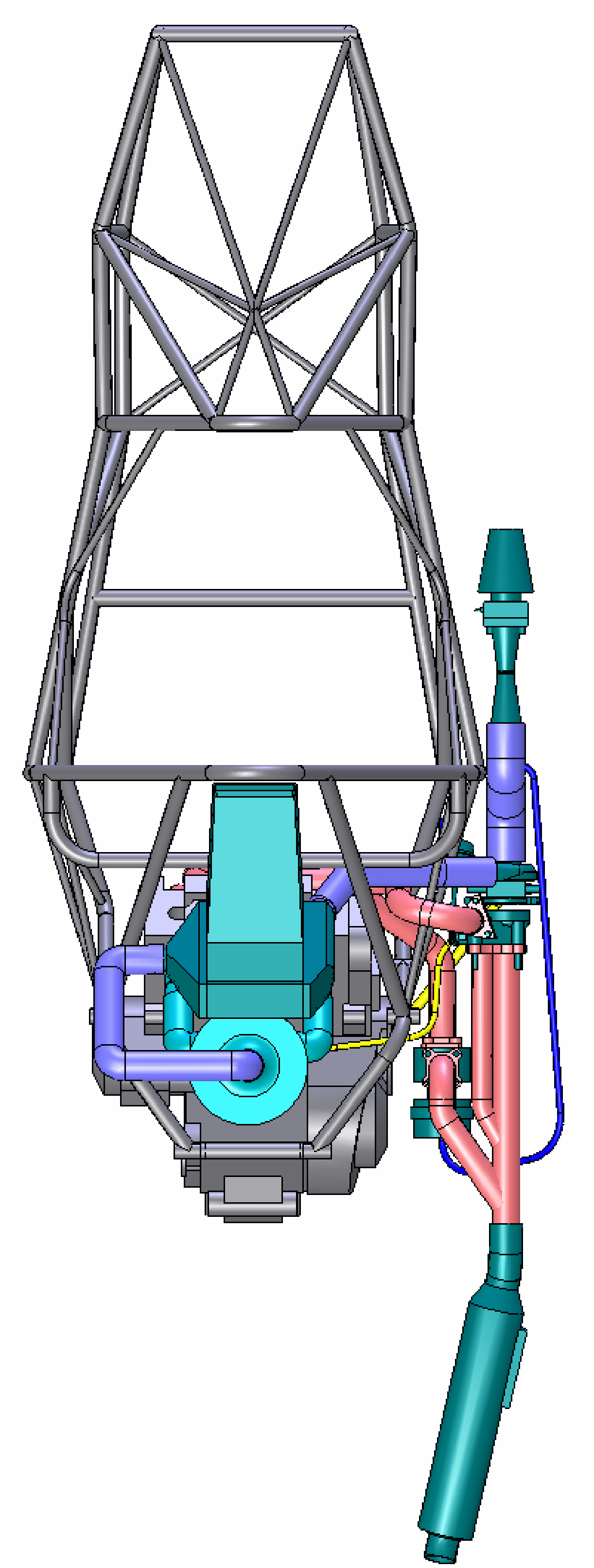


Figure E.4 Final system mode, top view.

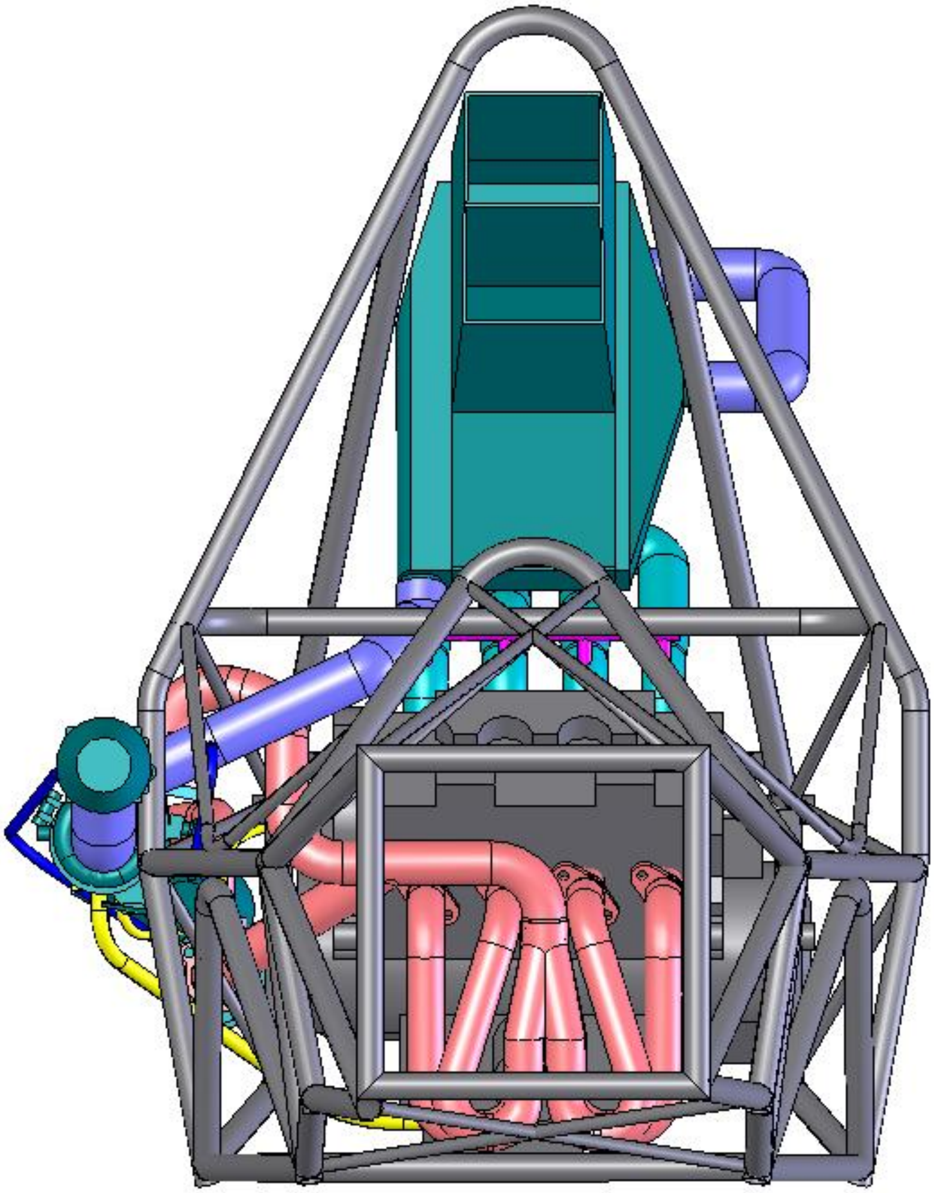
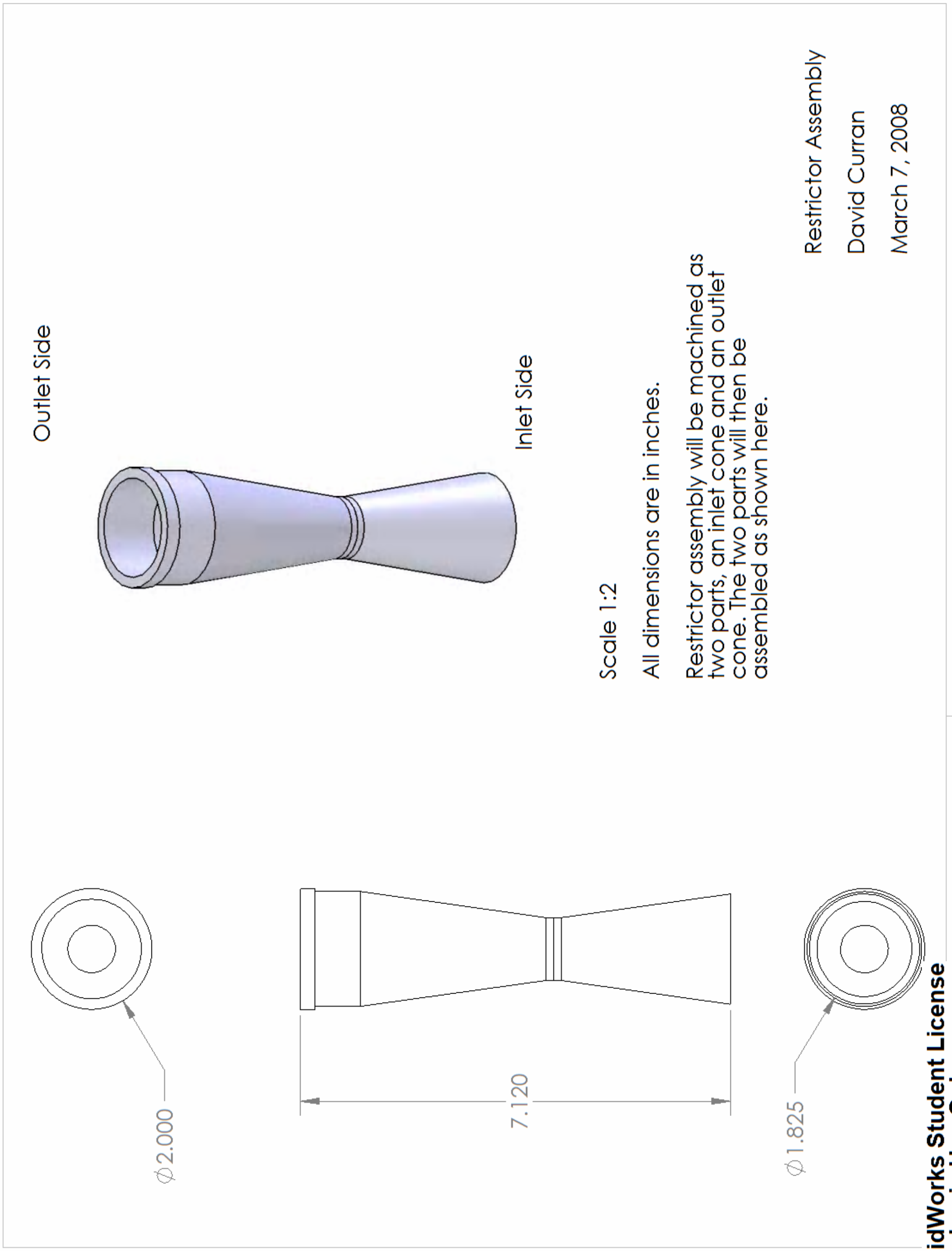


Figure E.5 Final system model, front view.

Appendix F – Intake Drawings

All of the intake components were modeled in SolidWorks. Drawings were created for each component that had to be machined or custom built. All models and drawings were done by David Curran. All drawings are scaled to 75% of actual size to accommodate the word processor’s formatting.



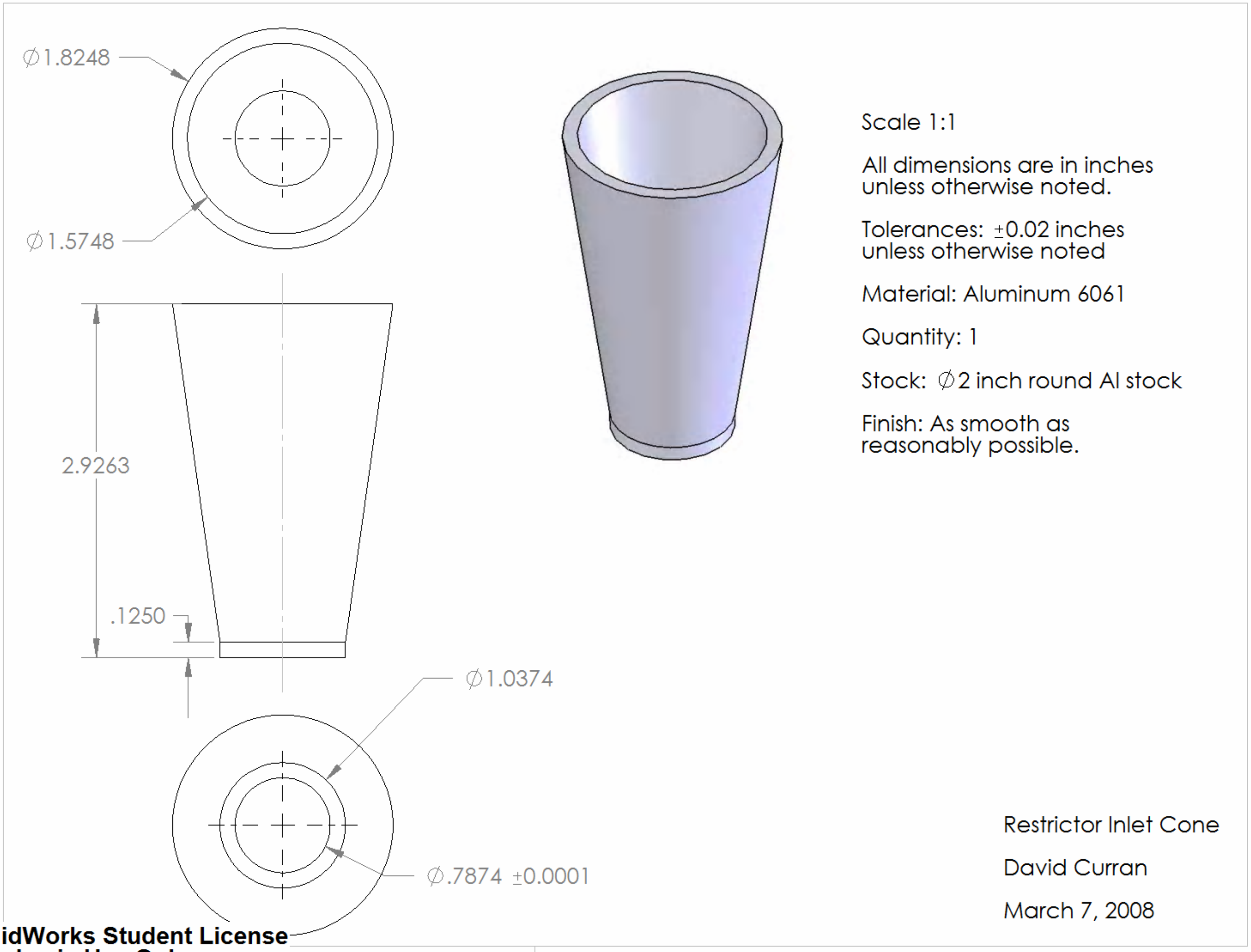
Scale 1:2

All dimensions are in inches.

Restrictor assembly will be machined as two parts, an inlet cone and an outlet cone. The two parts will then be assembled as shown here.

Restrictor Assembly
David Curran
March 7, 2008

Figure F.1 Restrictor assembly drawing.



Scale 1:1

All dimensions are in inches unless otherwise noted.

Tolerances: ±0.02 inches unless otherwise noted

Material: Aluminum 6061

Quantity: 1

Stock: ∅2 inch round Al stock

Finish: As smooth as reasonably possible.

Restrictor Inlet Cone

David Curran

March 7, 2008

Figure F.2 Restrictor inlet cone drawing.

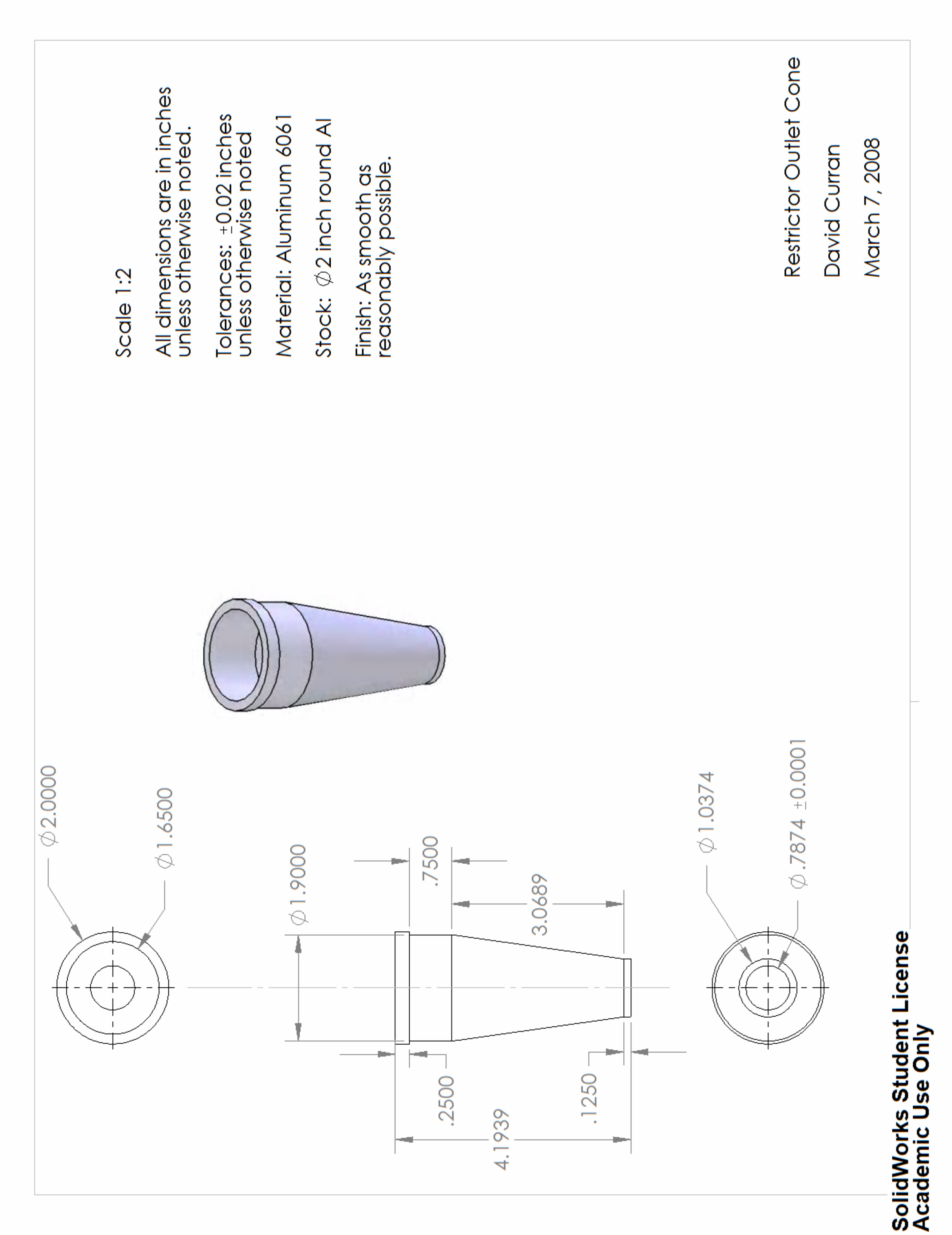


Figure F.3 Restrictor outlet cone drawing.

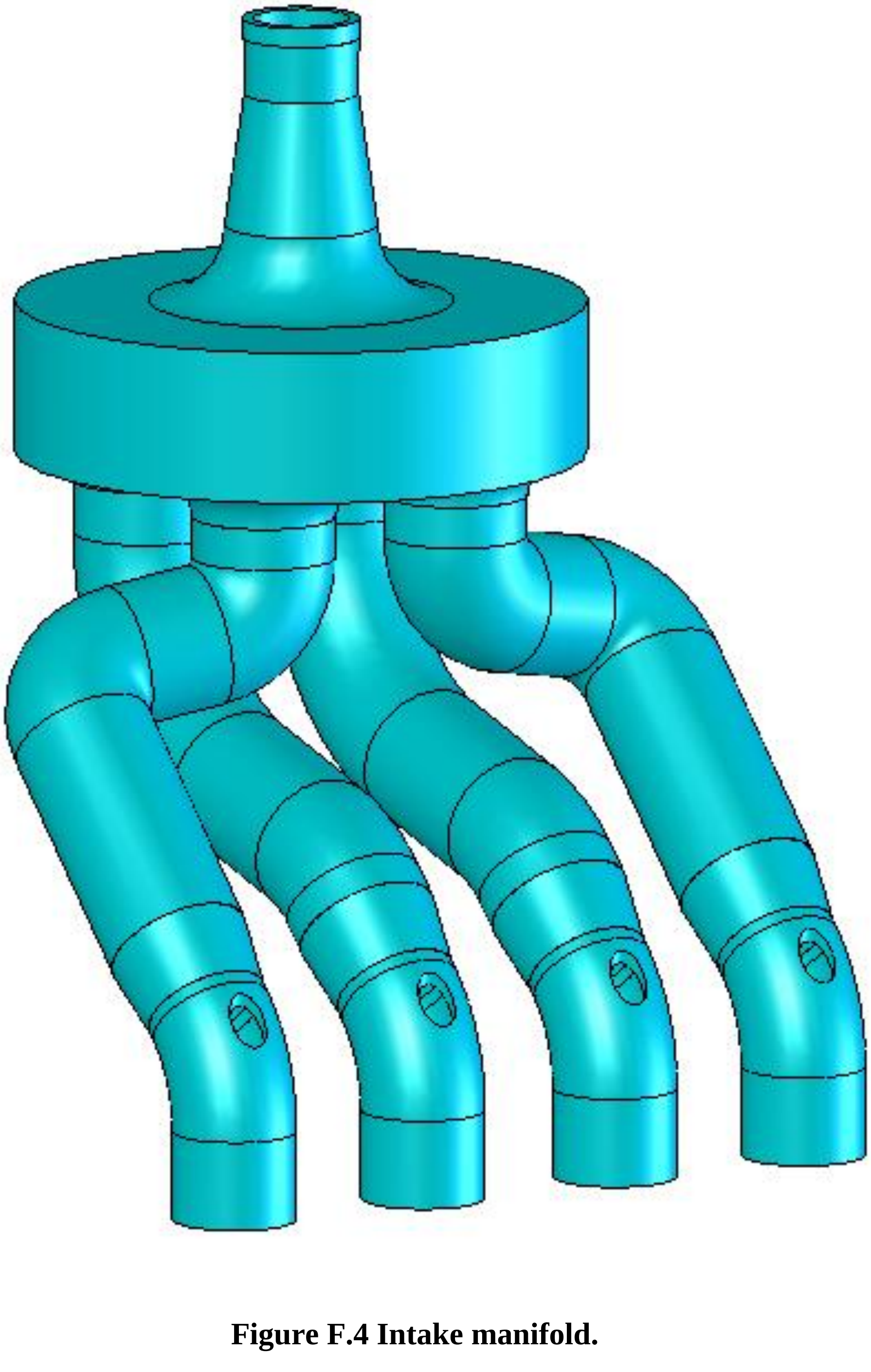


Figure F.4 Intake manifold.

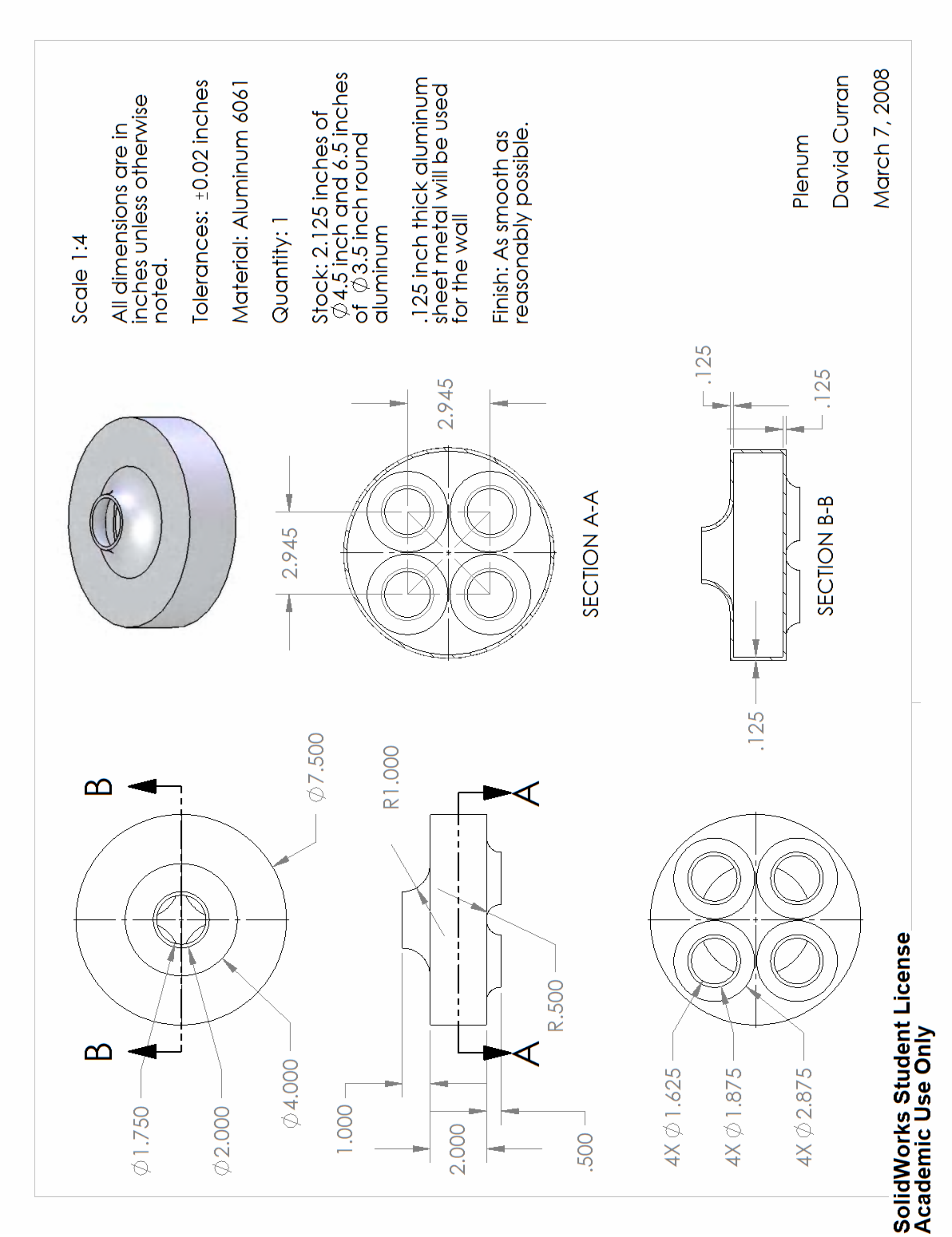


Figure F.5 Plenum drawing.

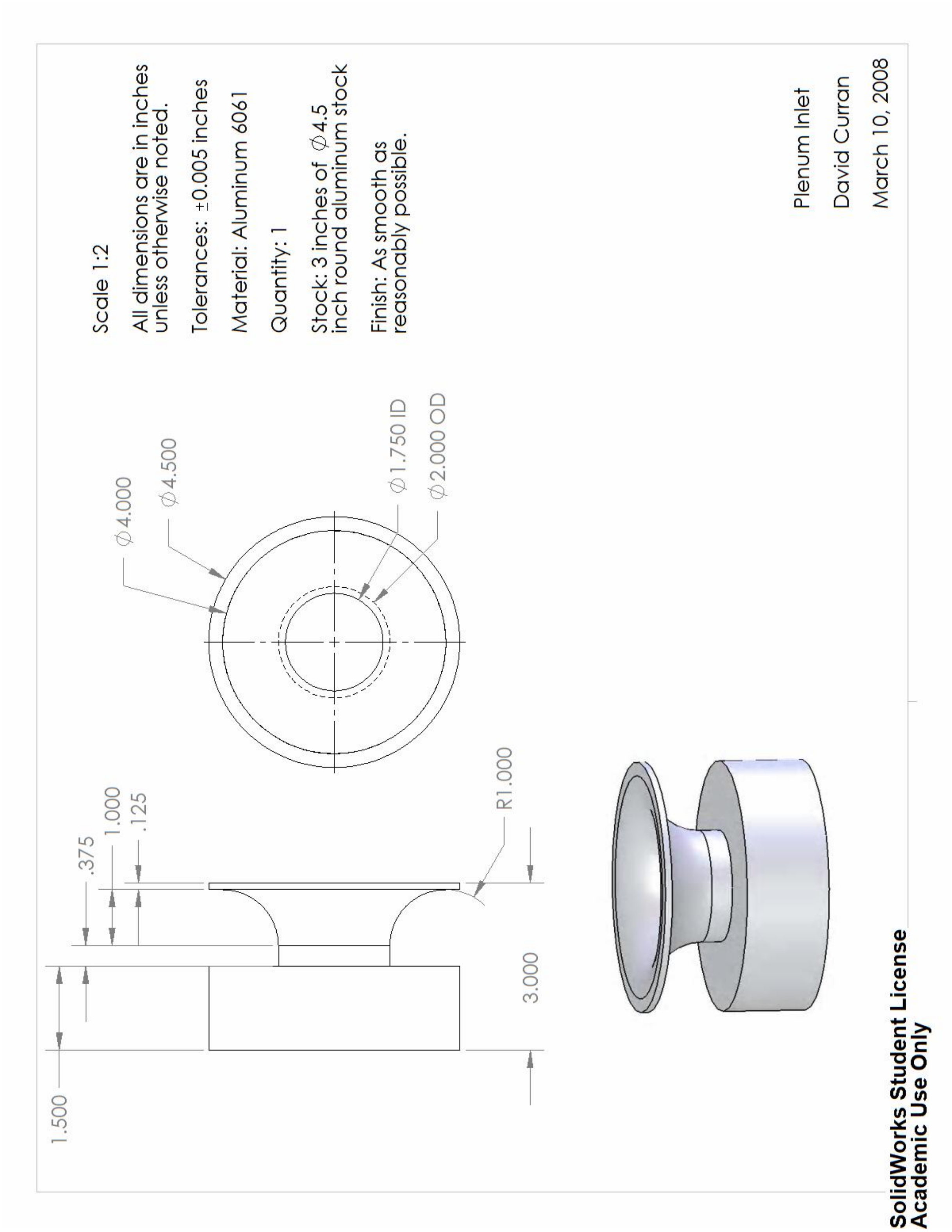


Figure F.6 Plenum inlet machining drawing.

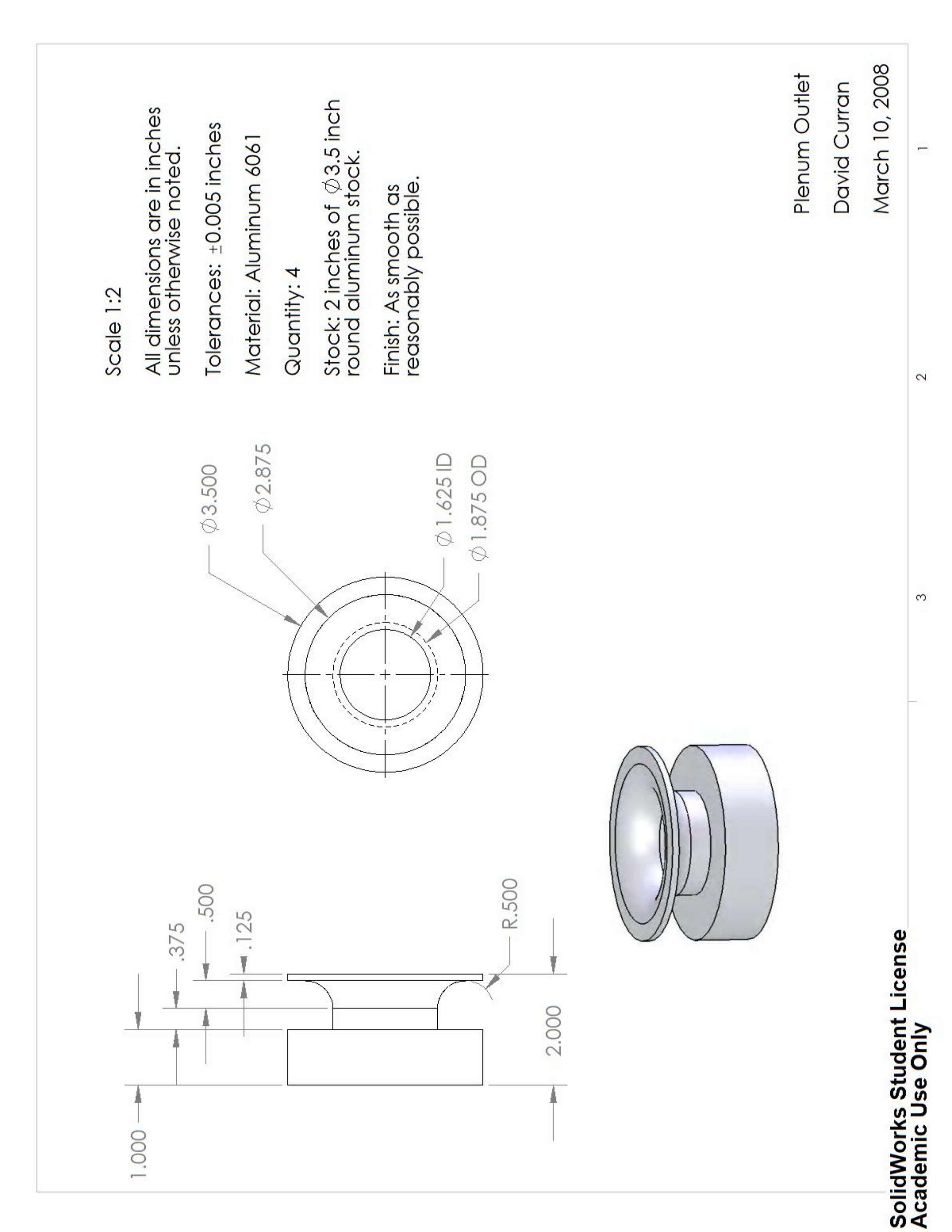


Figure F.7 Plenum outlet machining drawing.

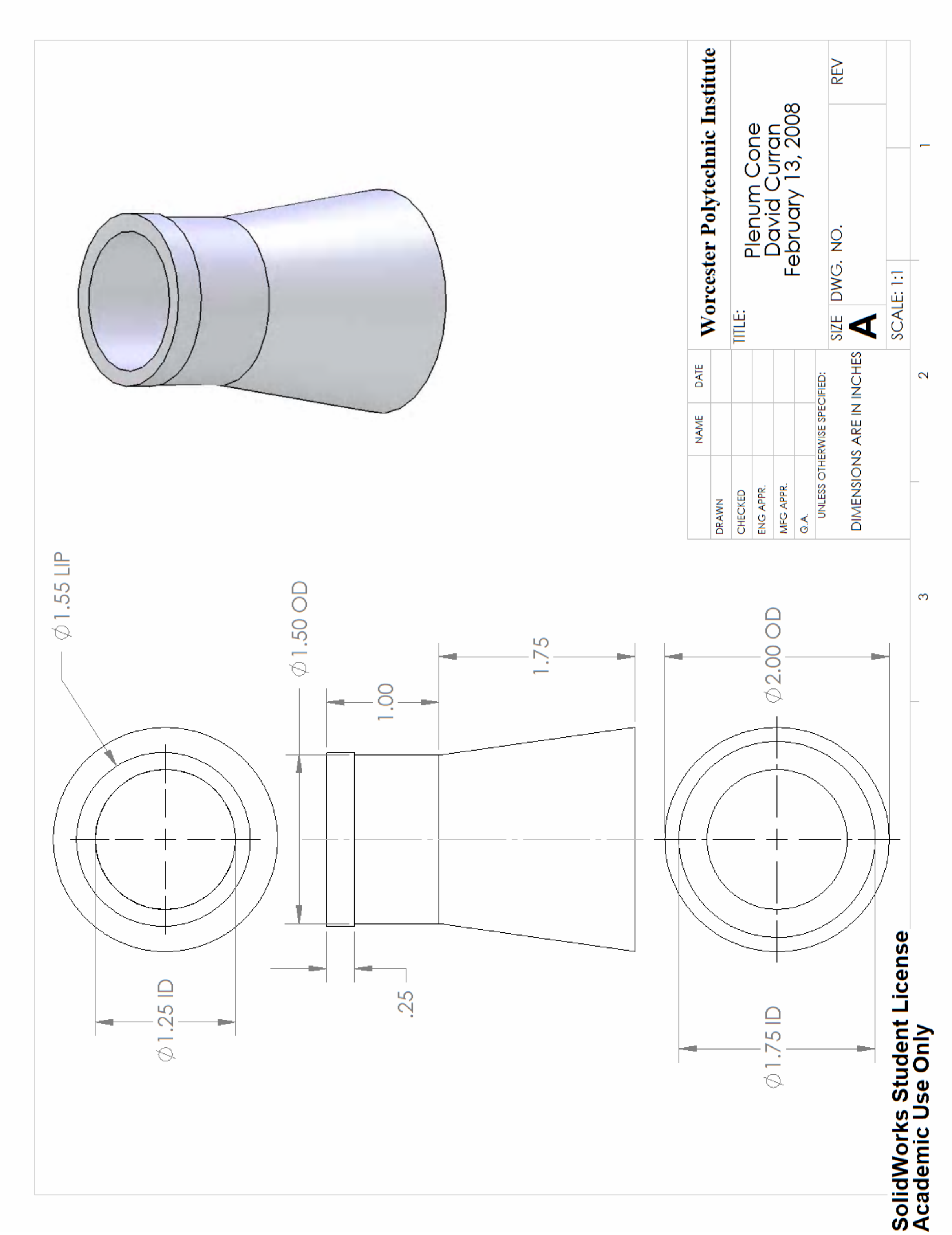


Figure F.8 Plenum cone drawing.

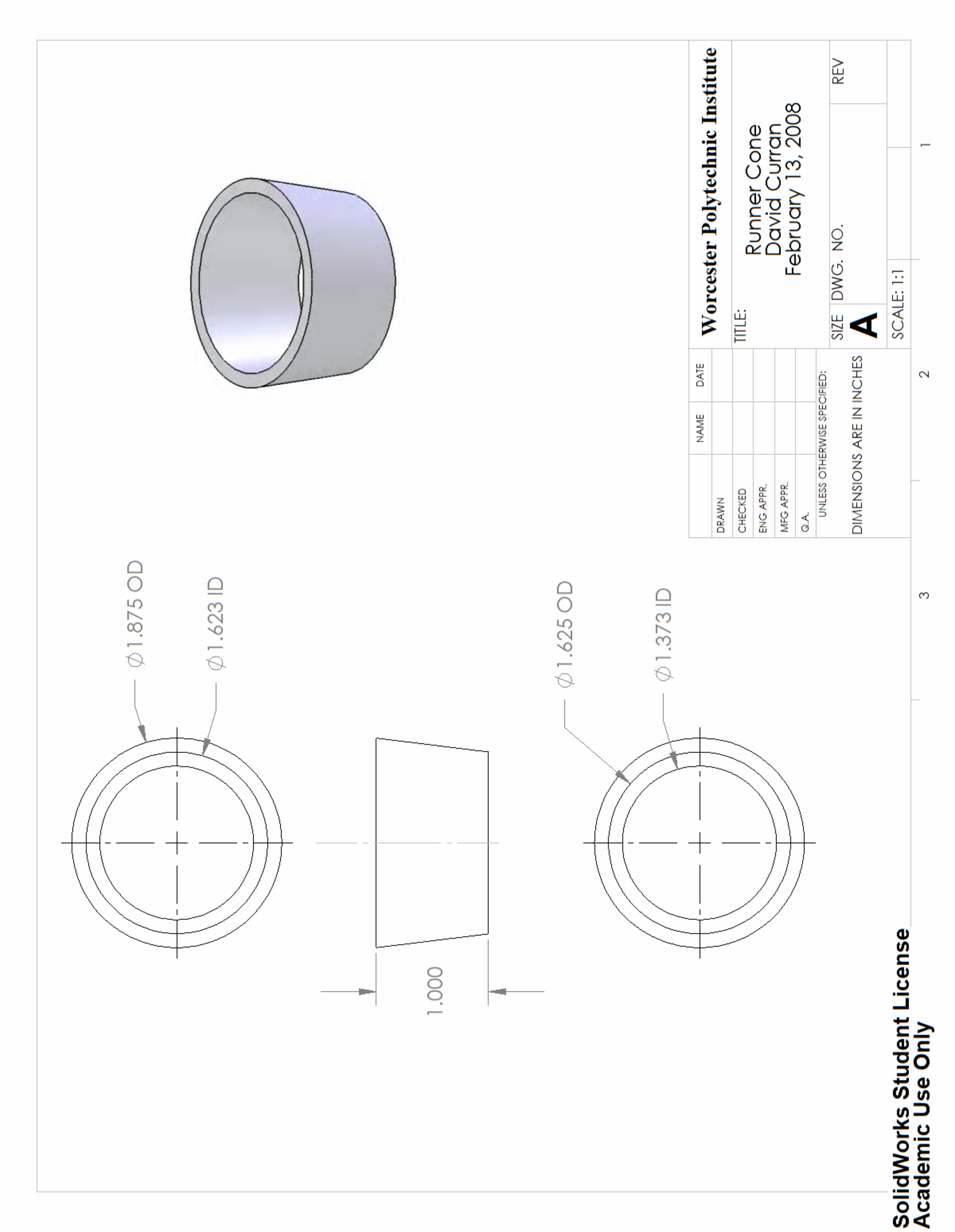


Figure F.9 Runner cone drawing.

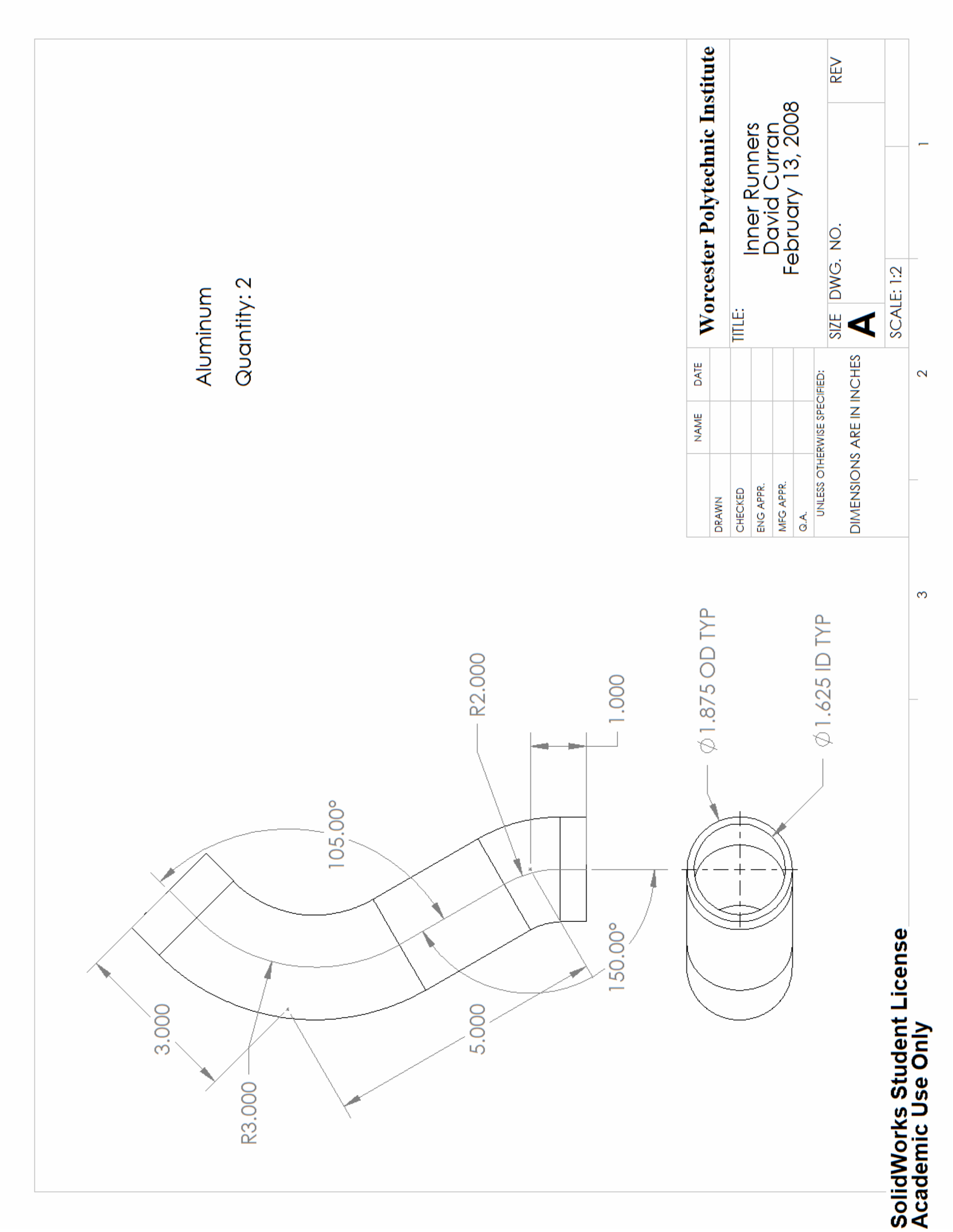


Figure F.10 Inner runner drawing.

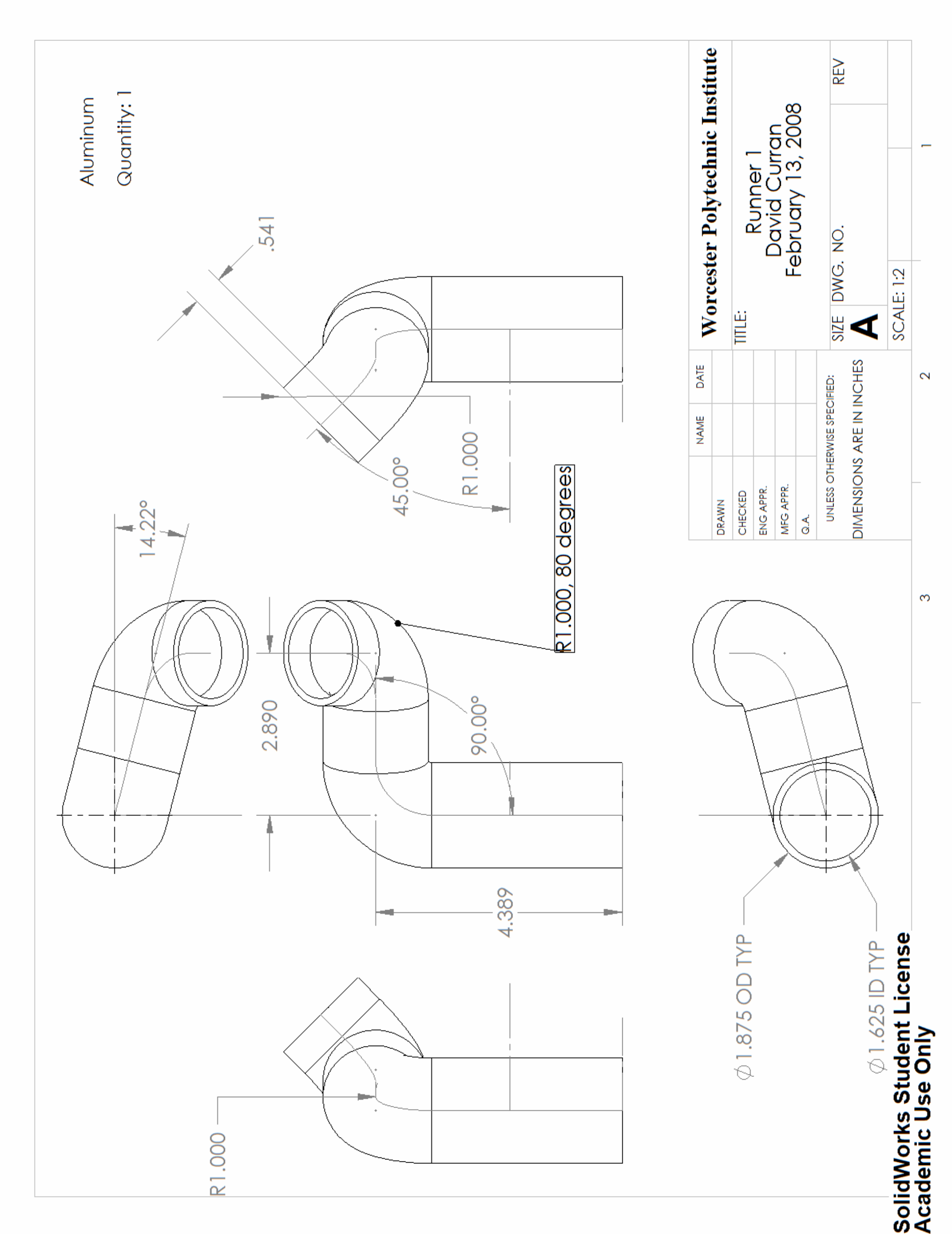
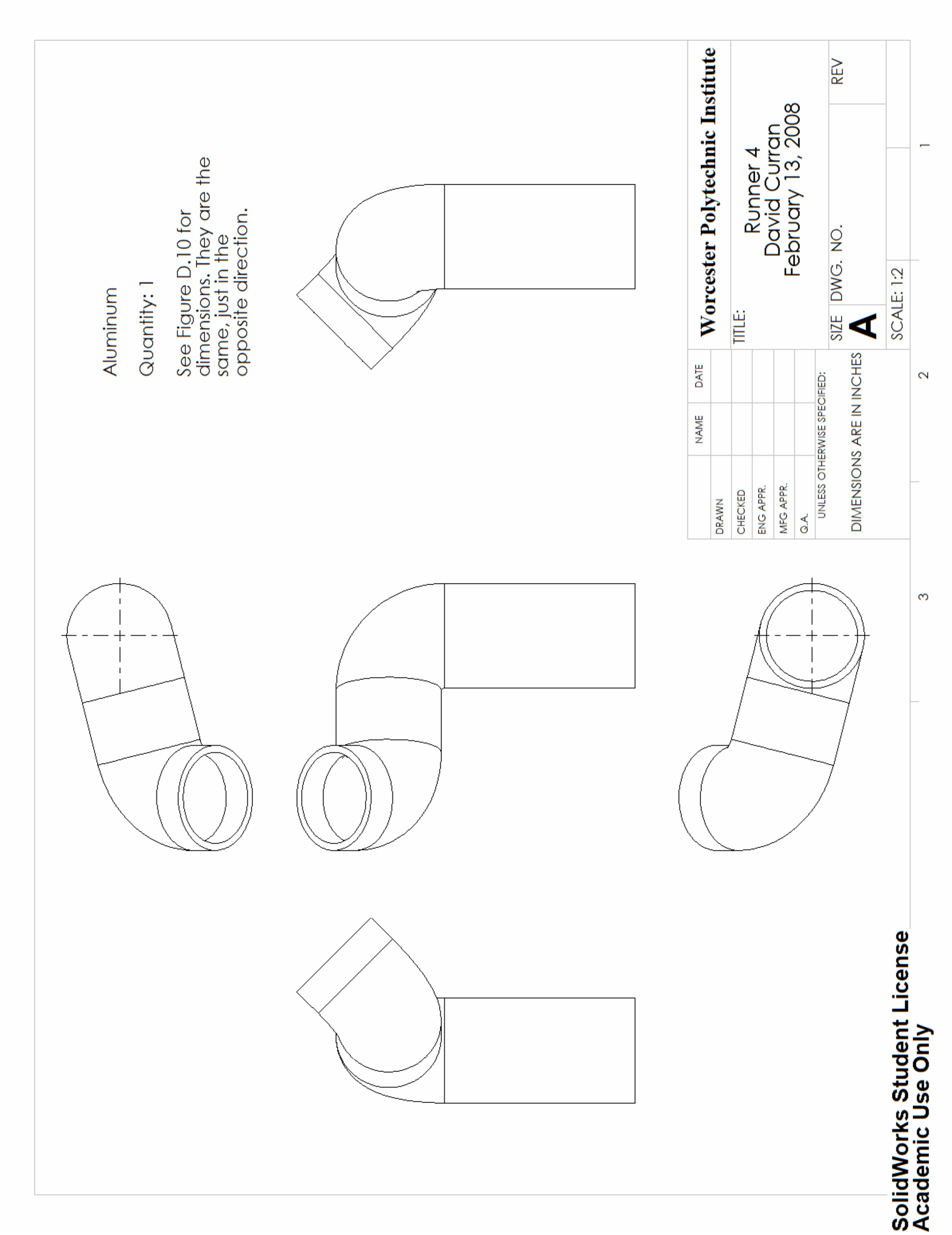


Figure F.11 Right outer runner drawing.



Aluminum

Quantity: 1

See Figure D.10 for dimensions. They are the same, just in the opposite direction.

Figure F.12 Left outer runner drawing.

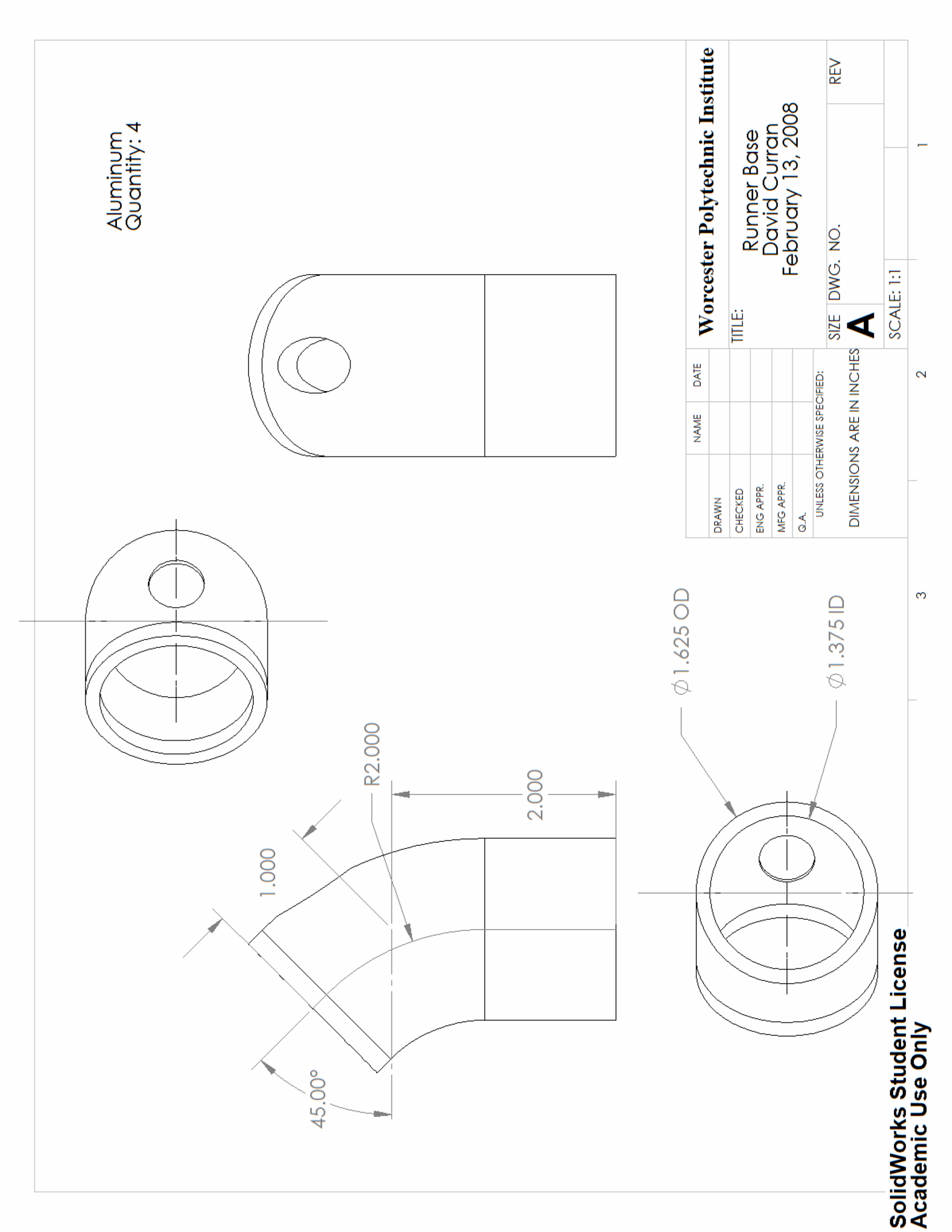


Figure F.13 Runner base drawing.

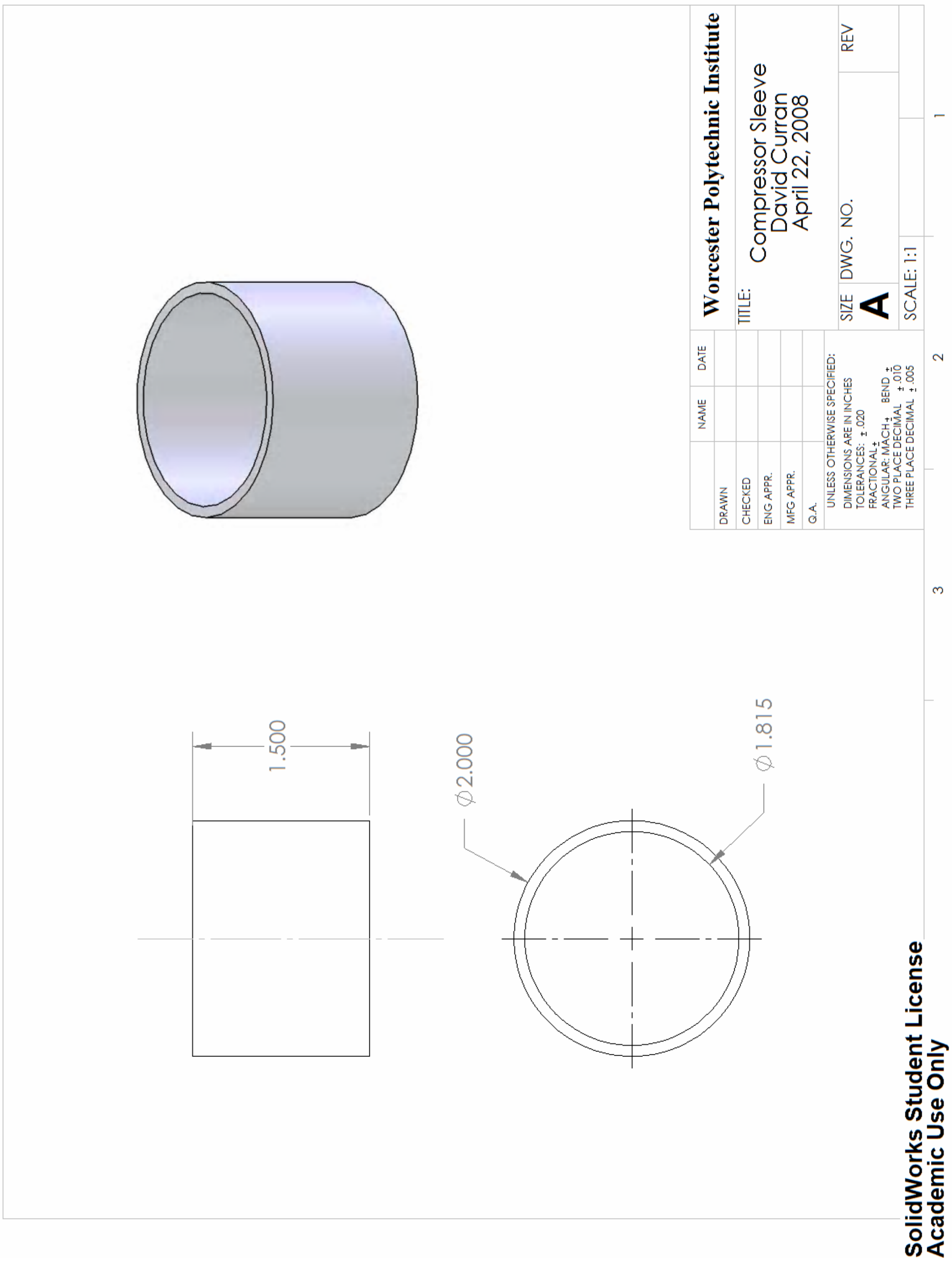


Figure F.14 Compressor sleeve drawing.

Appendix G – Machining Process

The plenum runners and the restrictor inlet and outlet were successfully machined by the team at the Washburn Shops. Raw stock aluminum and steel rods were purchased online to accommodate their machining needs. Since the team had no idea how to use any of the CNC machines or Gibbscam software, they were faced with a rather difficult challenge of machining their own parts. Due to the budget concerns, it was deemed not practical to have these parts custom machined elsewhere because the cost would be too great. With the help of Neil Whitehouse, the team was taught how to use the lathes along with Gibbscam. The team designed their parts in Solidworks, and then imported them into Gibbscam. In addition, each part needed to be made with roughly 2 inches of additional raw stock attached to it to allow for clamping while turning and machining. This excess stock was to be cut off the part and discarded once machining was done. In the case of the restrictor inlet and outlet, each piece would be cut from its clamping stock and then the pieces would be welded together at the middle to form the restrictor. This strategy made machining easier because it presented two smaller pieces of similar geometry to be machined rather than one large piece. It would be very difficult and complicated to machine an entire restrictor out of a single solid piece of stock. The geometry is too complicated and the longer the stock, the more tool chatter would happen which would affect the quality of the machining. In addition WPI does not have a great selection of machining tools, especially long ones, so it was easily determined that machining the restrictor out of a single piece of metal was not a good idea.

The team machined all their parts on a lathe using the same procedure as taught to them from Neil Whitehouse. The first step was to find the correct tools to machine with. For all surface contouring, a triangular-shaped diamond cutter was used. Holes were drilled by using both a center/spot drill and also a drill bit of the desired diameter. Internal contouring was done using a boring bar with the proper sized fitting. These tools were then installed into the lathe by screwing them into the proper tool holders. Care was taken to ensure that tools were not too close to each other and that each tool had a properly aimed nozzle that could spray coolant on the tool while machining.

With all the desired tools properly and securely installed into the lathe, the next step was to probe each tool. This was done using the lathe's drop-down probing bar. To do this, one must gently drop the bar down into the probing position. If not done gently and carefully, the machine will beep as a warning. It does this because the probing tip is extremely sensitive to the slightest touches and if not handled properly with the right hands, could break and become unusable. With the probing bar in the probing position, one must probe each tool one at a time. This is done by selecting the tool to be probed. Once selected, the user must manually jog the tool over very closely to the probing tip. Once very close (but not touching) the user then decreases the increment of movement and holding down the direction button they want to move in, they allow the tool to gently touch the probing tip. Once the tool touches the probe tip, the machine stops and beeps. This beep indicates the tip's location on the x-z axis in respect to the tool selected and the value is stored in the program to be run. This ensures machining accuracy and quality. All tools to be machined need to be probed individually. Certain tools such as drills are only probed in the z-direction because that is the only direction in which they cut. Other tools such as the boring bar need to be probed in the z and x axis because they cut in both those directions.

Once the tools are properly installed and probed, the next step is to adjust the clamping chuck to the size of the stock to be turned. It is important that the three clamping bars of the chuck be securely screwed in place at exactly the same distance from the center of the chuck. To check for this, one has to count the number of teeth showing at which the chuck is screwed down. All three clamps must be screwed in with the same number of teeth showing. This ensures accuracy and safety in machining allowing for a strong and uniform clamp on the part from all 3 clamping arms. Once a desired setting is attained, the clamp is opened using the footswitch and the part to be machined can be put in.

Once the preparation work for the lathe is ready, the machining can begin. The program built in Gibbscam gets saved onto a disk and then the disk is inserted into the lathe itself. The computer on the lathe reads the disk drive and the user opens the program that they desire. The program is written in G-code and the user must make sure that for the lathe it is G-36 (the first time the team machined they did not know this and the wrong G-code caused the tool to collide with the spinning chuck). Once the program is loaded, the z-offset can be set. With the stock clamped into the chuck, allow for the spindle to spin (for aluminum, the team used 1000 RPM). With the spindle spinning, select the diamond cutter and manually jog it over just to the very edge of the stock. The idea is to shave off a very slight layer of metal to both level off the end of the stock and to set the z-offset. Once the z-offset is correct, machining can begin. The team ran all of their operations with the lathe at 5% rapid. This causes the lathe to move its tools much slower and allowed for the team to keep a very close eye on what was going on and allowed for the opportunity to hit the emergency stop button if something went wrong.

The shorter aluminum pieces were able to be machined successfully. The longer pieces showed the roughed edges evident of tool chatter due to the increased vibration of a long rod while spinning and also the tools being used getting duller. The machine shop did not have sufficient tools to complete the pieces that were needed to be done. The steel piece could not be contoured on the inside because the boring bar did not have a fitting that could cut steel. It was attempted but stopped immediately because the tool chipped the instant it touched the spinning stock. Another problem was there were no drills that were long enough to hollow out the longer pieces. What drills there appeared to be very old and dull and did not cut very well and overheated very easily. It would have benefited the team to have known that the machine shop did not have a well stocked supply of needed tools to accommodate their needs ahead of time so that the proper fittings could have been purchased.

Appendix H – Necessary Pipes for Manifolds

The following table contains a summary of the pipes needed to complete the intake and exhaust manifolds as designed. The aluminum pipes are for the intake manifold, and the steel pipes are for the exhaust manifold. All lengths were standardized and extended beyond what is needed so these pipes can be cut down to the exact lengths needed for the manifolds and then welded together as appropriate. A vendor could not be located for these pipes, but there may be a mandrel bender in Epping, New Hampshire that could produce these pipes. However, the cumulative cost could easily reach $2000.

Material	ID	OD	Wall Thickness	Length	Angle	Bend Radius	Quantity
Aluminum	1.375	1.625	.125	6	45	2	4
Aluminum	1.625	1.875	.125	6	30	2	2
				6	75	3	2
				6	80	1	2
				12	90	1	2
Steel	1.125	1.375	.125	6	70	2	2
				6	90	1.5	4
				6	58	2	2
				6	110	1.25	2
				6	90	1.25	2
				24	-	-	1
Steel	1.375	1.625	.125	6	90	2	3
				6	76	2	1
				6	-	-	1
Steel	1.500	1.625	.0625	6	60	2	1
				6	90	2	1
				6	35	2	1
				24	-	-	1
Steel	1.000	1.250	.125	6	30	2	1
				12	-	-	1
Steel	1.500	1.750	.125	18	-	-	1

All dimensions are in inches.
All angles are in degrees.

Table H.1 Summary of pipes needed to completed intake and exhaust manifolds.

Appendix I – Pictures of Intake Components

The following are pictures of the machined and purchased intake components.

Figure I.1 Restrictor inlet cone with extra stock, complete.

Figure I.2 Restrictor outlet cone with extra stock, incomplete.

Figure I.3 Restrictor inlet (right) and outlet (left), side by side.

Figure I.4 Plenum cone, completed.

Figure I.5 Plenum cone, alternate view.

Figure I.6 Runner cones, completed.

Figure I.7 Runner cones, side by side.

Figure I.8 Silicone hoses, 2.0 inch ID, 45° bends.

Figure I.9 Silicone hoses, 1.5 inch ID, 45° bends.

Figure I.10 Silicone hoses, 1.5 inch ID, 90° bends.

Figure I.11 Silicone hose, 1.5 inch ID, 24 inches.

Figure I.12 Clamps and joiner for 2.0 inch ID silicone hoses.

Figure I.13 Clamps and joiner for 1.5 inch ID silicone hoses.

Figure I.14 Turbocharger with silicone hoses affixed to it.

Figure I.15 Turbocharger with silicone hoses affixed to it, alternate view.

Figure I.16 Intercooler with silicone hoses affixed to it.

Appendix J – Intercooler Drawings and Pictures

The following pages contain detailed drawings of the intercooler as well as pictures of the final product manufactured by Bell Intercoolers. There is no drawing for the ducting because the design of this component was never finalized for the actual chassis. All drawings have been scaled to 75% to allow them to fit within the word processor’s formatting. All models and drawings were created by David Curran.

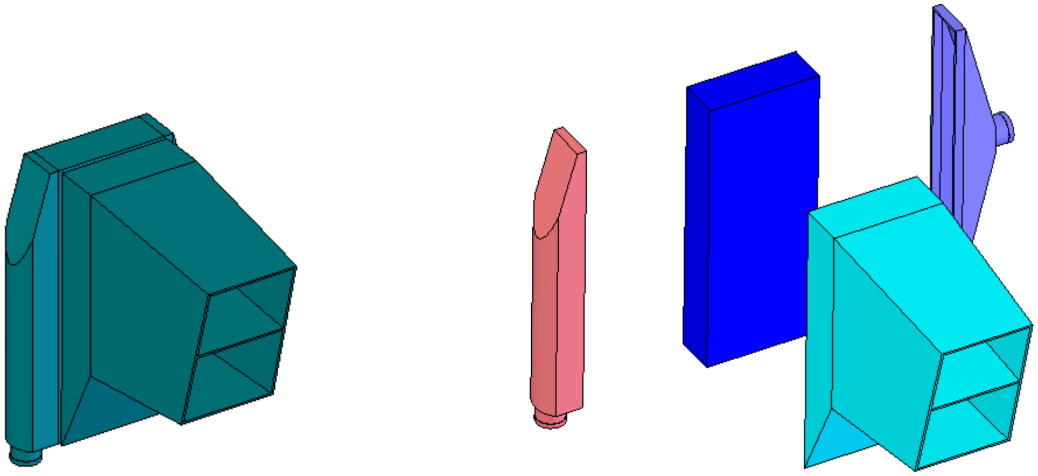


Figure J.1 Assembled intercooler (left) and exploded view showing individual components (right).

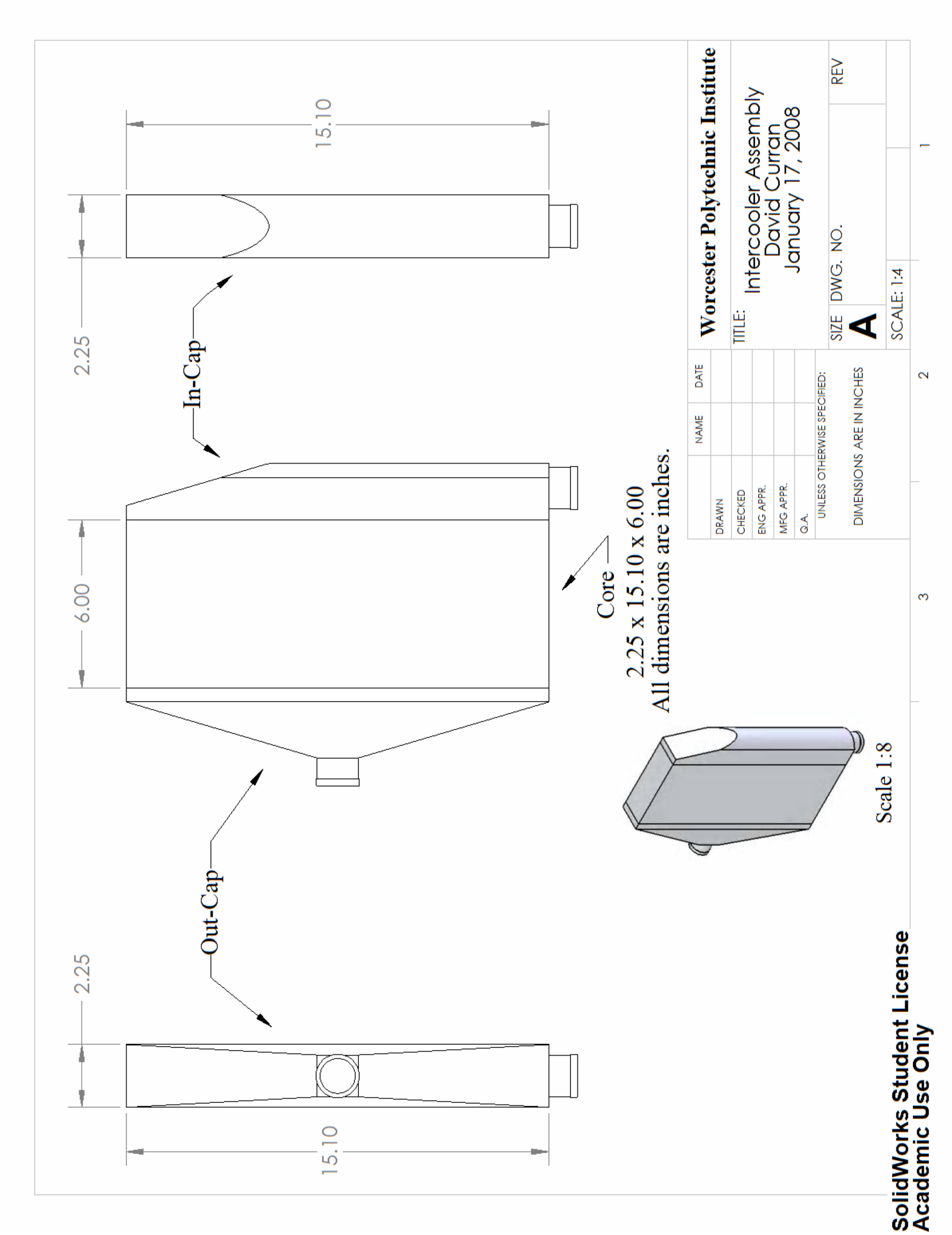


Figure J.2 Intercooler core and end tanks drawing.

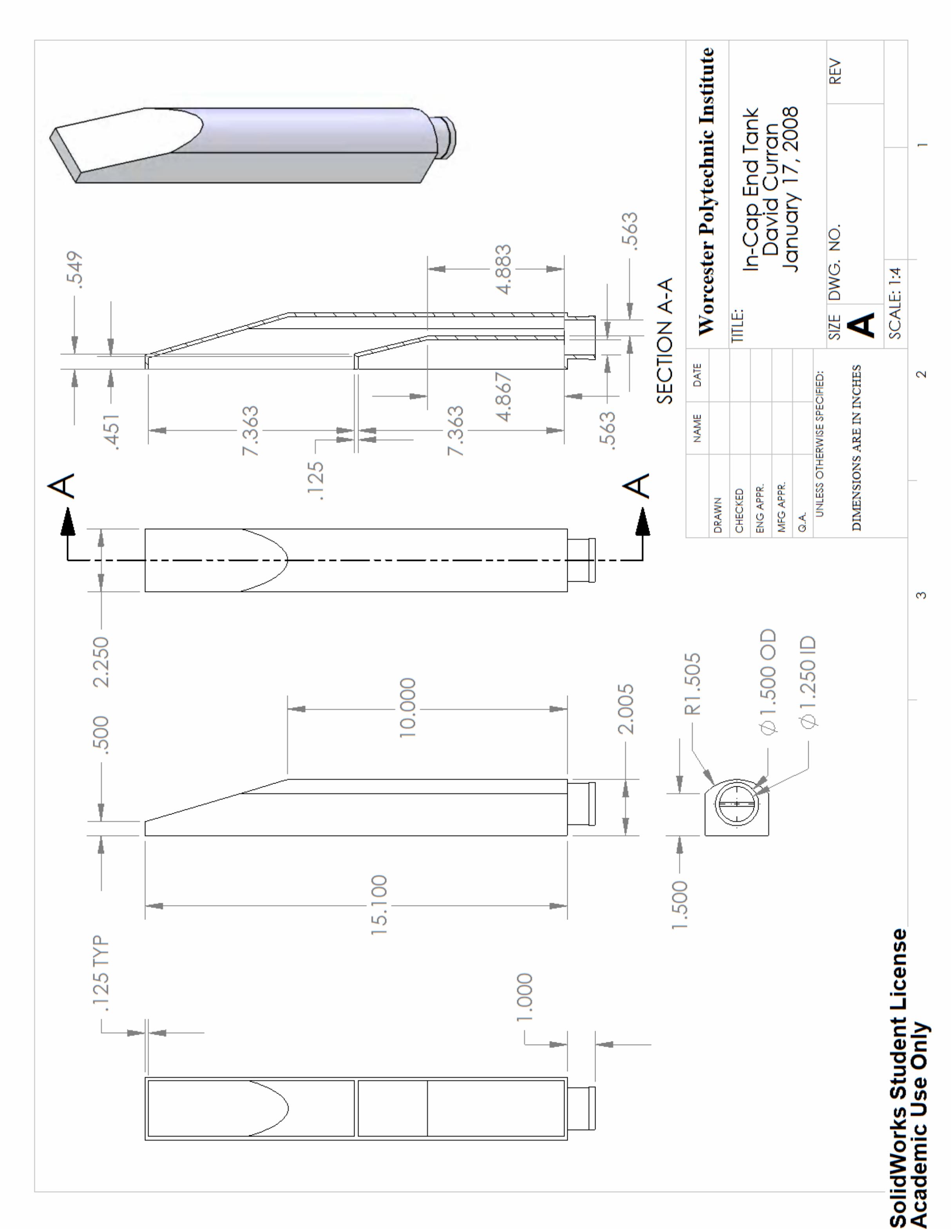


Figure J.3 Inlet side end tank drawing.

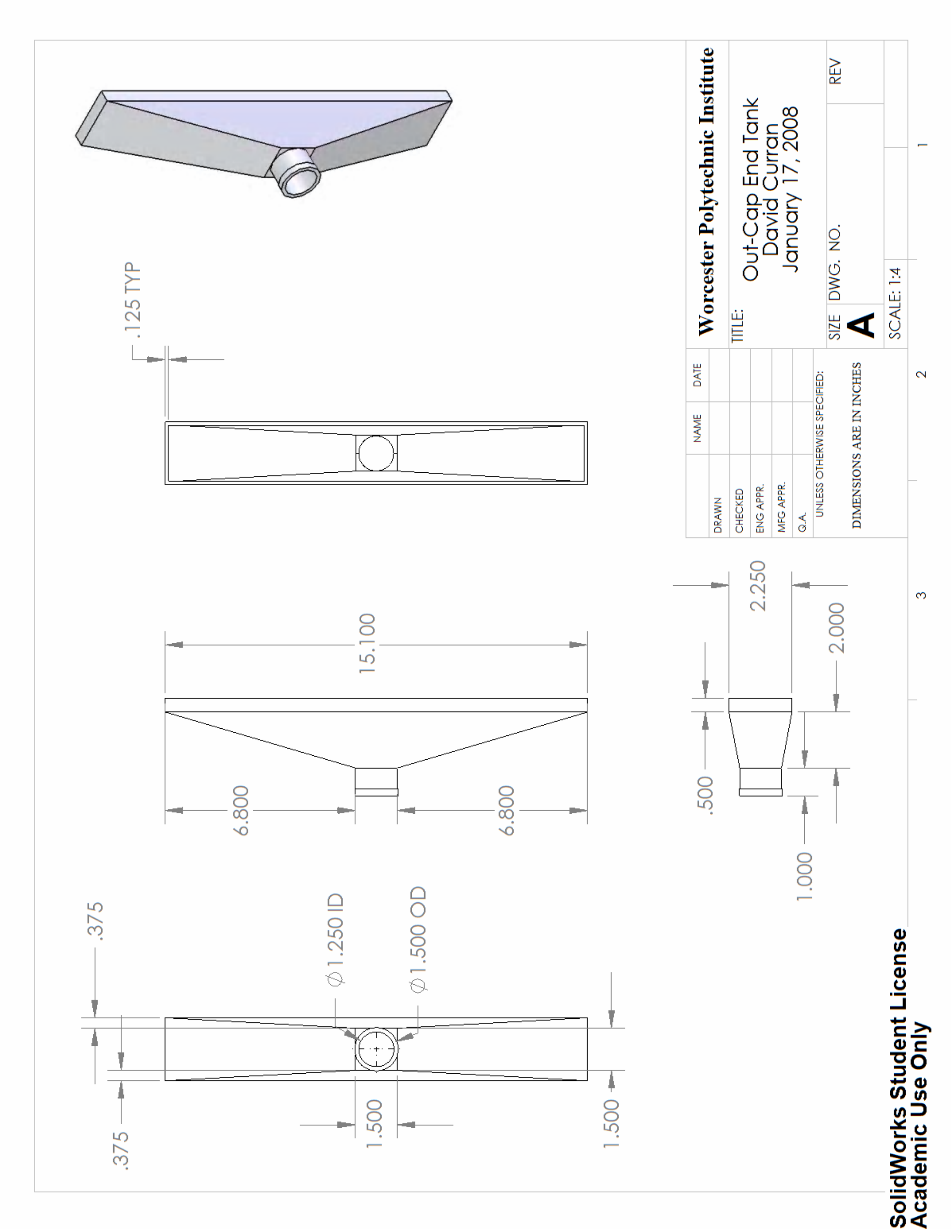


Figure J.4 Outlet side end tank.

Figure J.5 Intercooler manufactured by Bell Intercoolers.

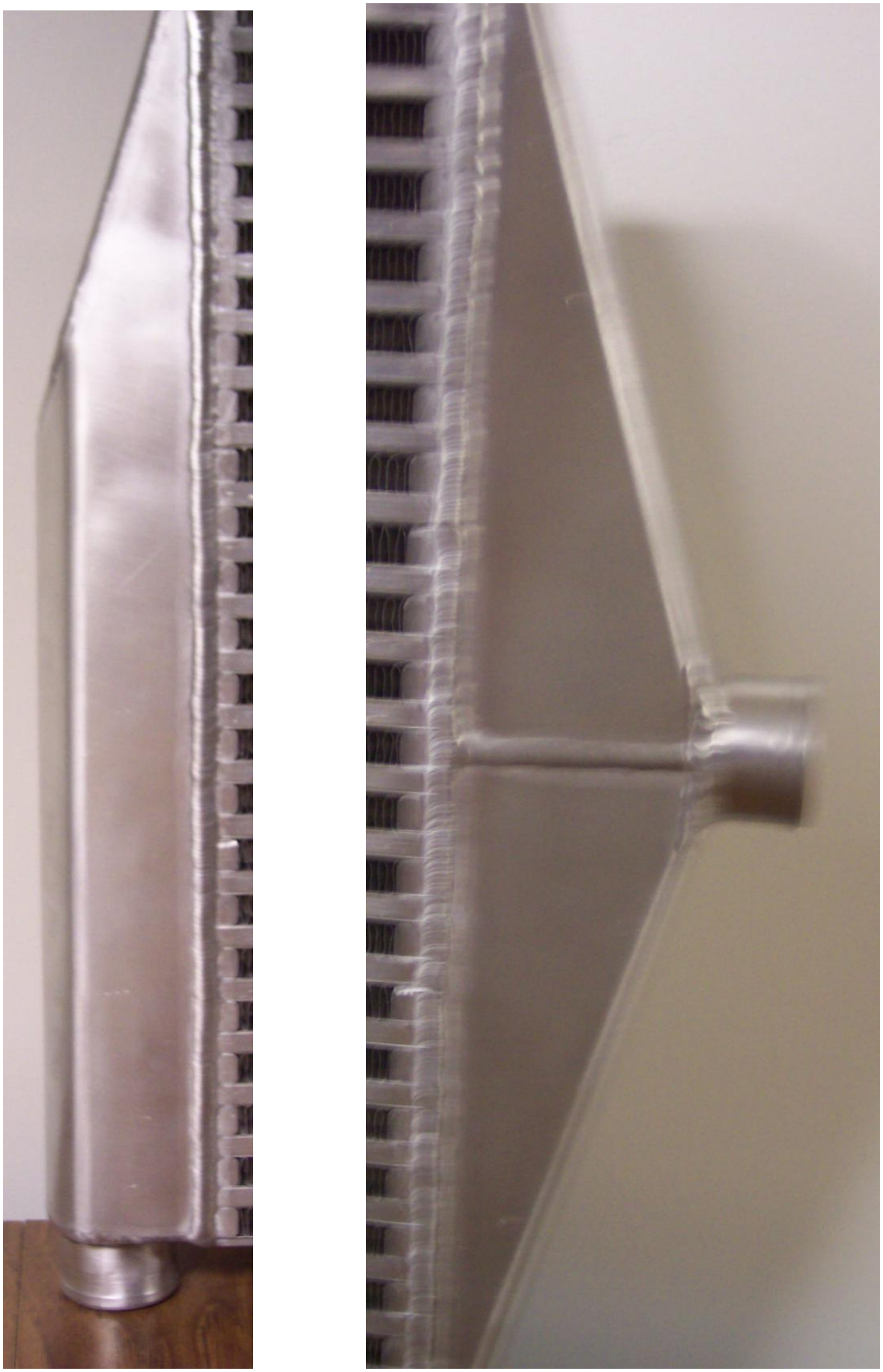


Figure J.6 Close up of end tanks, inlet side (left) and outlet side (right).

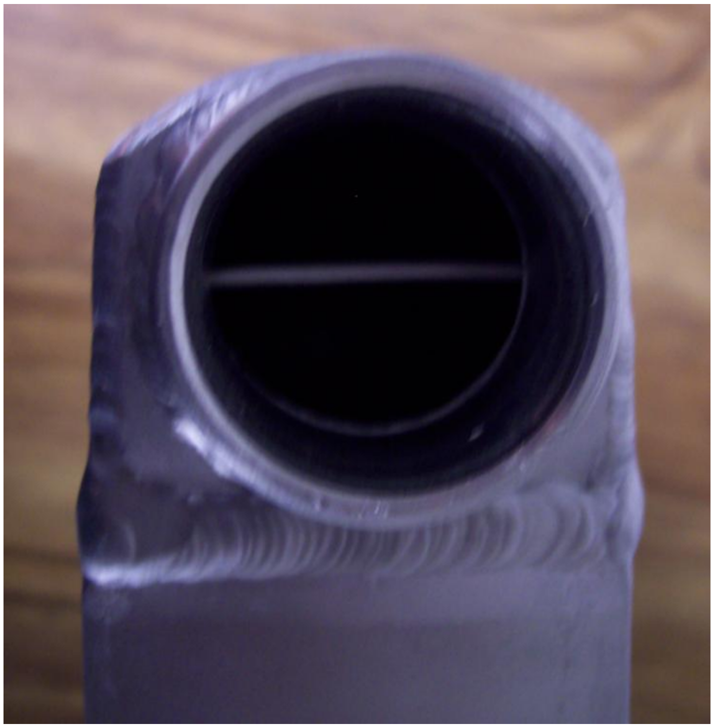


Figure J.7 Internal baffle of inlet side end tank.

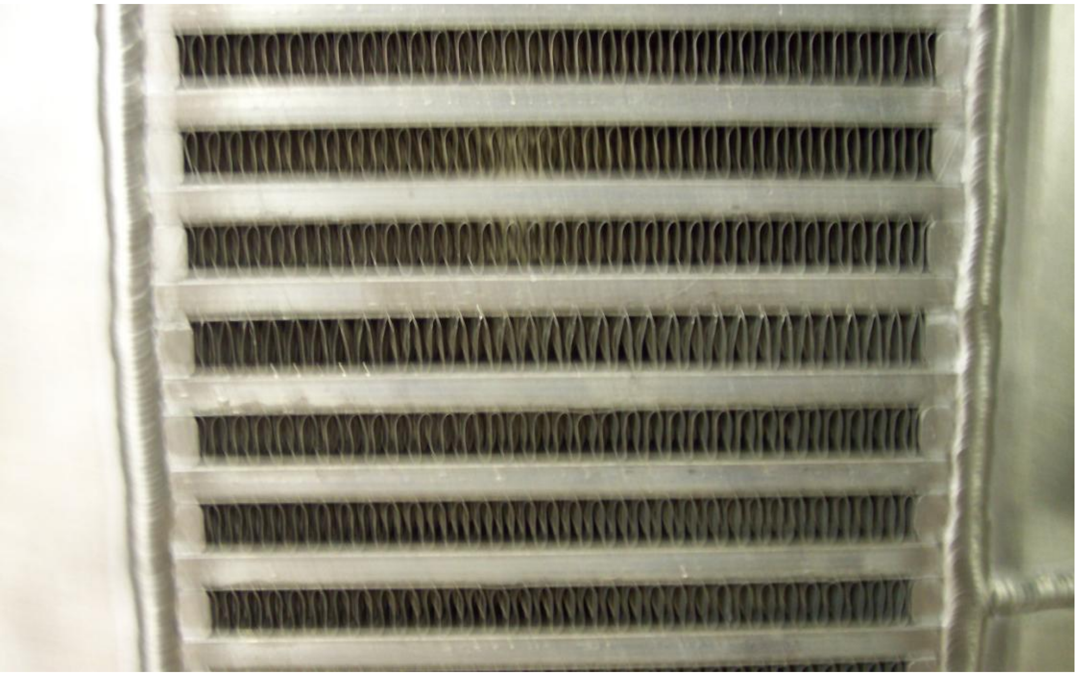


Figure J.8 Close up of intercooler core.

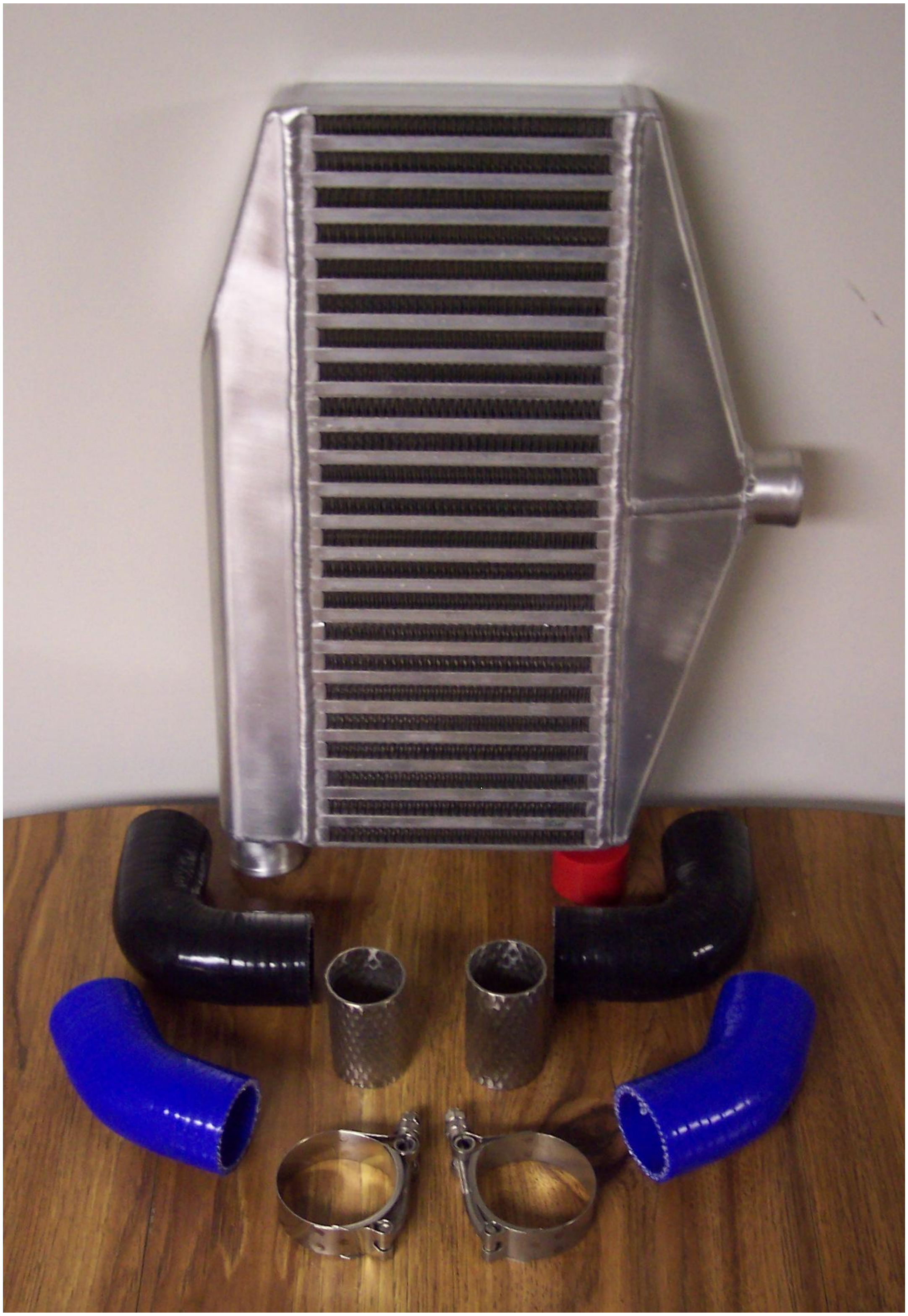


Figure J.9 Intercooler shown with silicone hoses, joiners and clamps.

Appendix K – Exhaust Drawings

All drawings scaled to 75%. All models and drawings by David Curran.

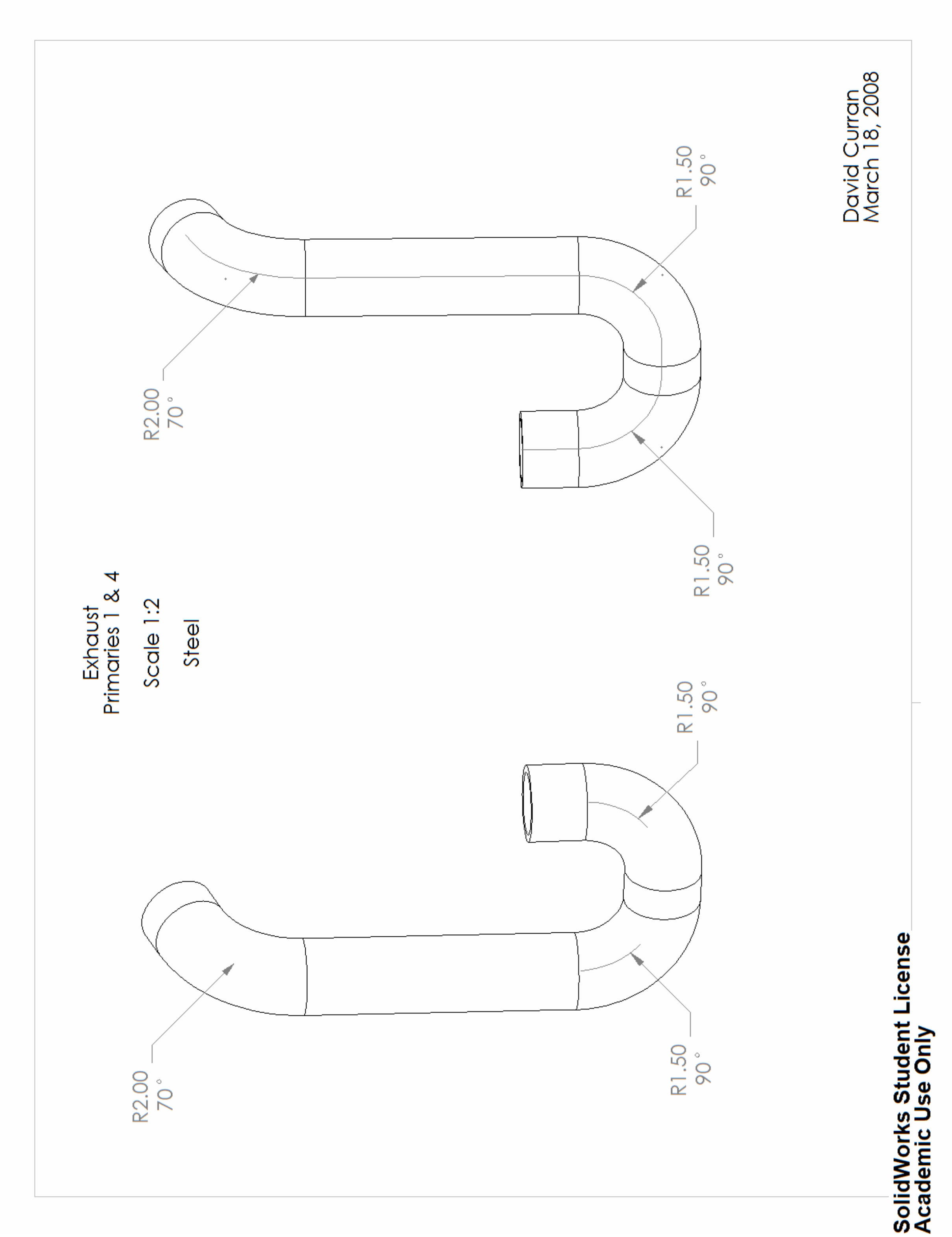


Figure K.1 Drawings for primaries 1 and 4.

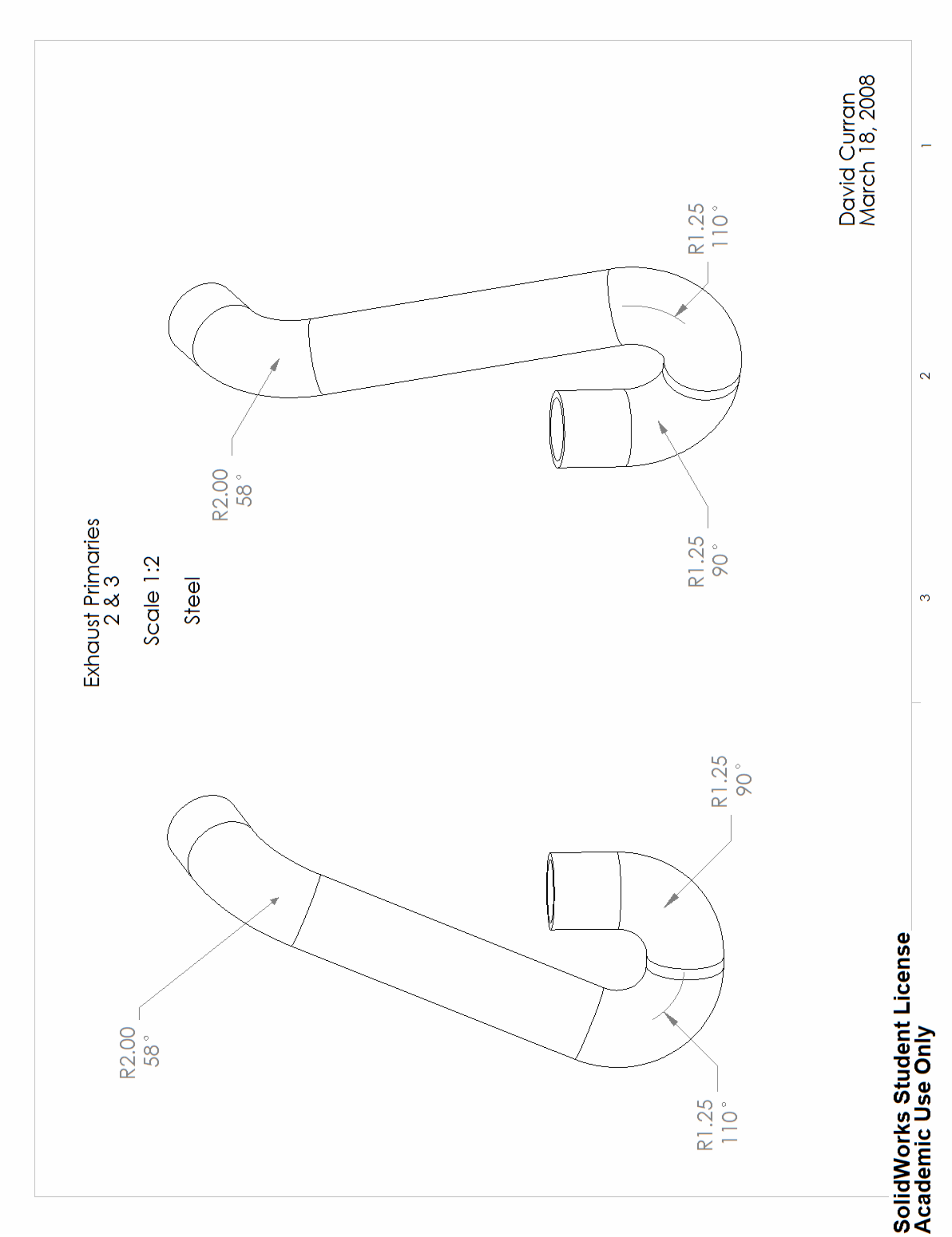


Figure K.2 Drawings for primaries 2 and 3.

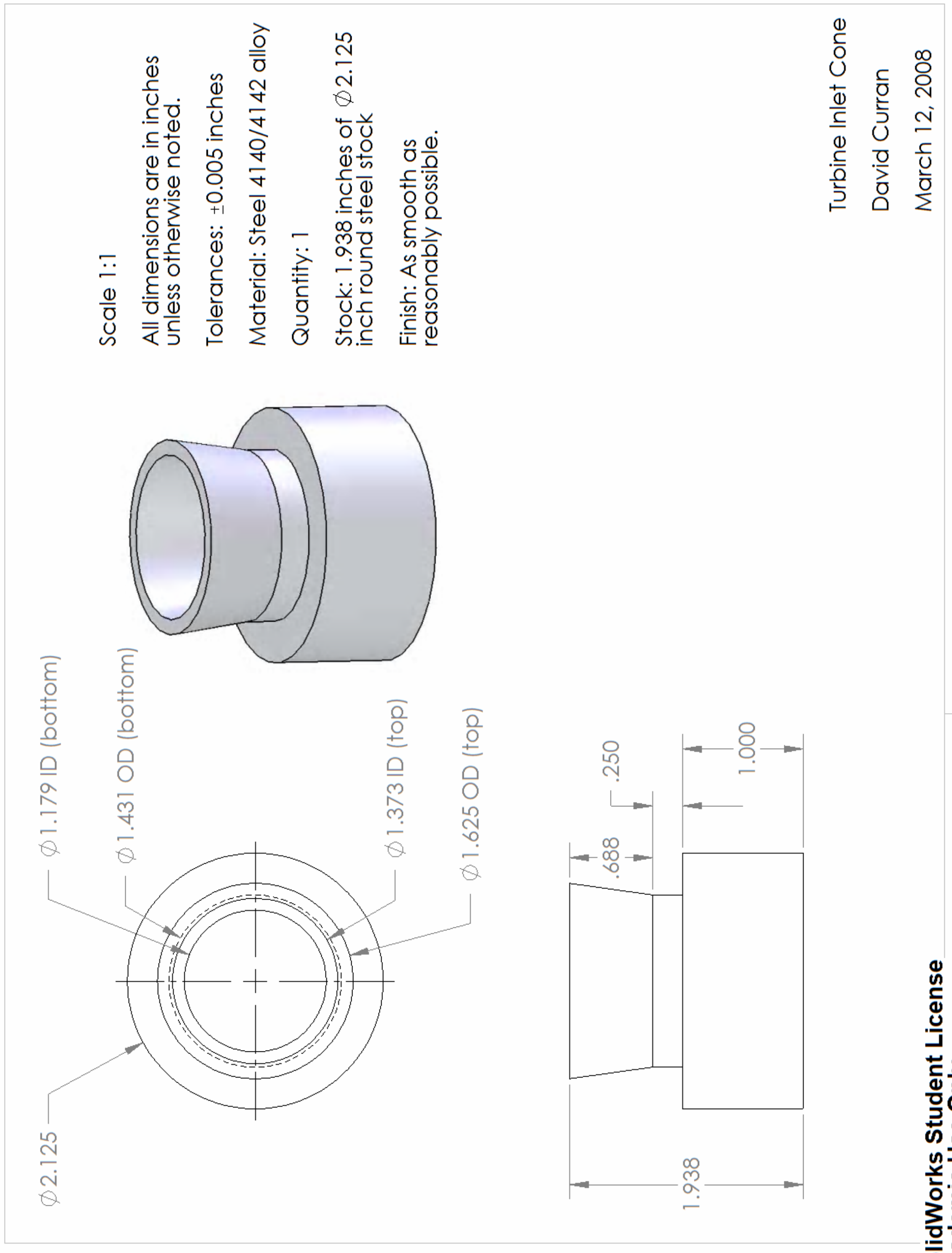


Figure K.3 Drawing for turbine inlet cone.

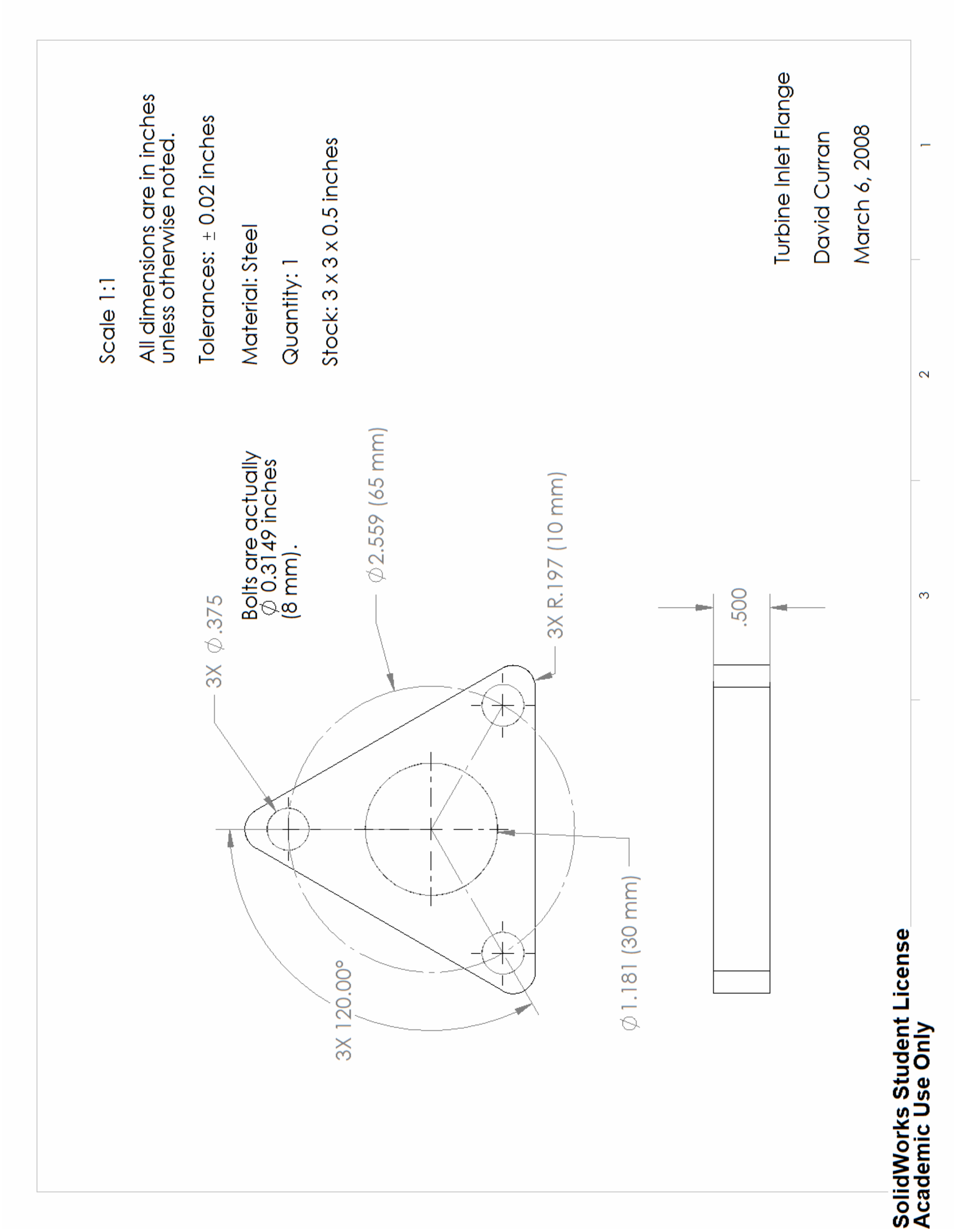
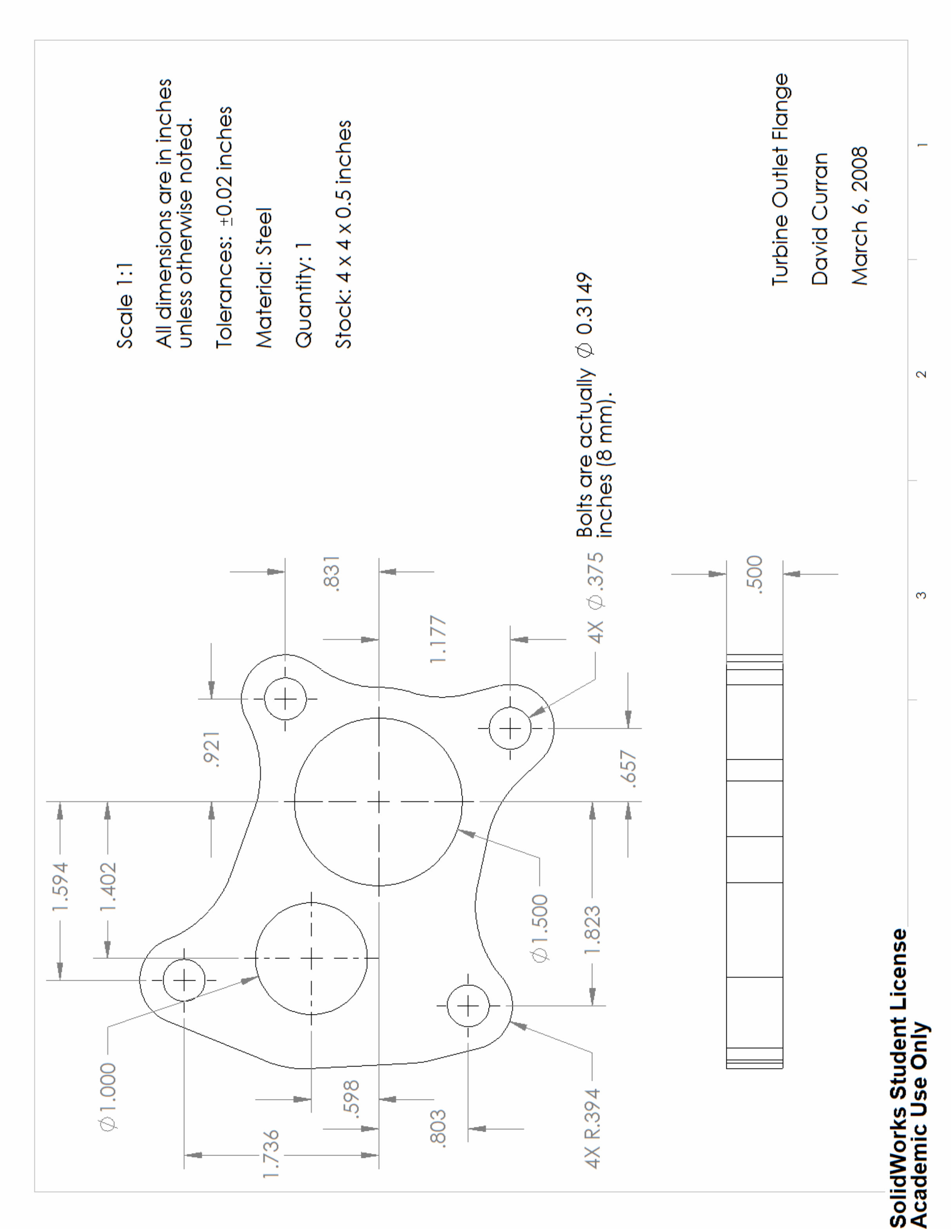


Figure K.4 Drawing of turbine inlet flange.



Scale 1:1

All dimensions are in inches unless otherwise noted.

Tolerances: ±0.02 inches

Material: Steel

Quantity: 1

Stock: 4 x 4 x 0.5 inches

Bolts are actually ⌀ 0.3149 inches (8 mm).

Turbine Outlet Flange
David Curran
March 6, 2008

Figure K.5 Drawing of turbine outlet flange.

Appendix L – Oil Flange Drawing

Drawing shown at 75%.

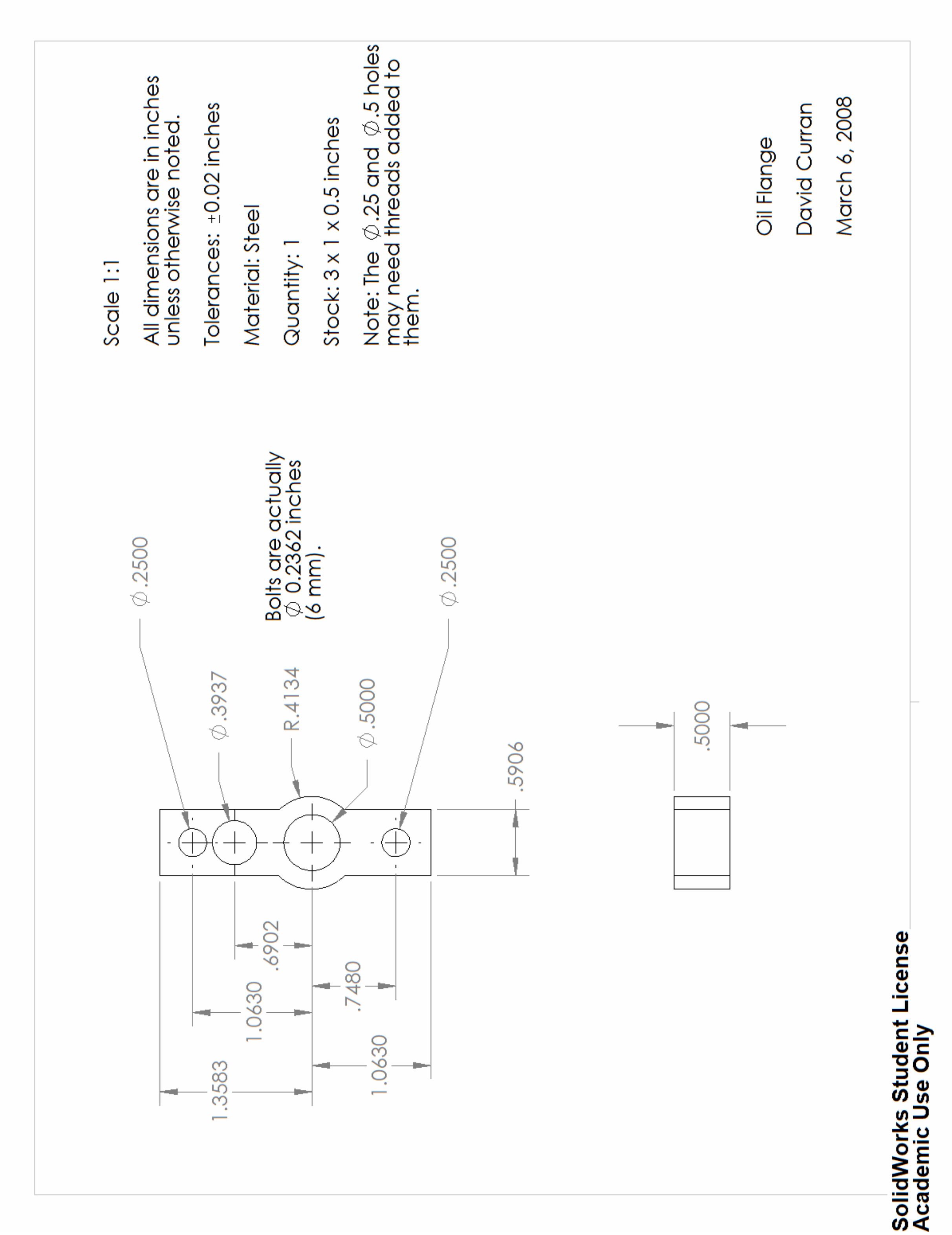


Figure L.1 Oil flange drawing.

Appendix M – Inventory

There were seven boxes as well as the engine that the team turned in at the conclusion of the project. This section contains the packing lists of those seven boxes of equipment. In addition to its packing list, each box was labeled with the following:

FSAE TURBO MQP

2007-08

Prof. Cobb

Packing Date

Box #

Item Number	Item Description	Quantity
1	’01 Honda 600 cc F4i Air Box	1
2	’01 Honda CBR 600cc Fuel Pump with Gas Tank, Banjo Bolt, and fuel/air hoses (all assembled)	1 (assembled)

Table M.1 Box #1 packing list.

Item Number	Item Description	Quantity
1	Radiator	1

Table M.2 Box #2 packing list.

Item Number	Item Description	Quantity
1	Garrett GT12 Turbocharger	1
2	Air/Air Intercooler, 15.1x6x2.25 inches	1
3	Air/Air Intercooler, 6x6x3 inches	1
4	TiAL F46 Remote Wastegate with signal ports, flanges, gaskets and bolts	1
5	4-1 Merge Collector	1
6	2.9psi Spring for TiAL F46	1
7	18psi Integral Wastegate Actuator	1
8	Punch Ports	2
9	Tee Splitter, 10mm	1
10	Threaded Adapter, 10mm	3
11	Threaded Adapter, 0.5 inch	1
12	Oil Pressure Sender, Honda CBR600 F4i	1

Table M.3 Box #3 packing list.

Item Number	Item Description	Quantity
1	Joiner, 1.5 inch silicone hose	8
2	Joiner, 2 inch silicone hose	1
3	Clamp, 1.5 inch silicone hose	18
4	Clamp, 2 inch silicone hose	4
5	45° elbow, 2 inch silicone hose	2
6	45° elbow, 1.5 inch silicone hose	2
7	90° elbow, 1.5 inch silicone hose	3
8	24 inches of 1.5 inch silicone hose	1

Table M.4 Box #4 packing list.

Item Number	Item Description	Quantity
1	Turbine outlet flange	1
2	Turbine inlet flange	1
3	Oil flange	1
4	Restrictor outlet	1
5	Restrictor inlet	1
6	Plenum cone	1
7	Runner cone	4
8	6mm bolts	2
9	6mm washers	4
10	8mm nuts	8
11	8mm washers	8
12	Turbine inlet cone	1

Table M.5 Box #5 packing list.

Item Number	Item Description	Quantity
1	Aluminum 6061, 3 inches of 4.5 inch round	1
2	Aluminum 6061, 12 inches of 3.5 inch round	1
3	Aluminum 6061, 7 inches of 2 inch round	1
4	Steel 4140, 9 inches of 2.125 inch round	1
5	Stainless steel pipe, 15.5 inches of 1.5 inch OD	1

Table M.6 Box #6 packing list.

Item Number	Item Description	Quantity
1	PE-ECU-1 aftermarket ECU	1
2	3-bar MAP sensor w/ connector	1
3	IAT sensor w/ connector	1
4	Spool of wire	1
5	User manual CD	1

Table M.7 Box #7 packing list.