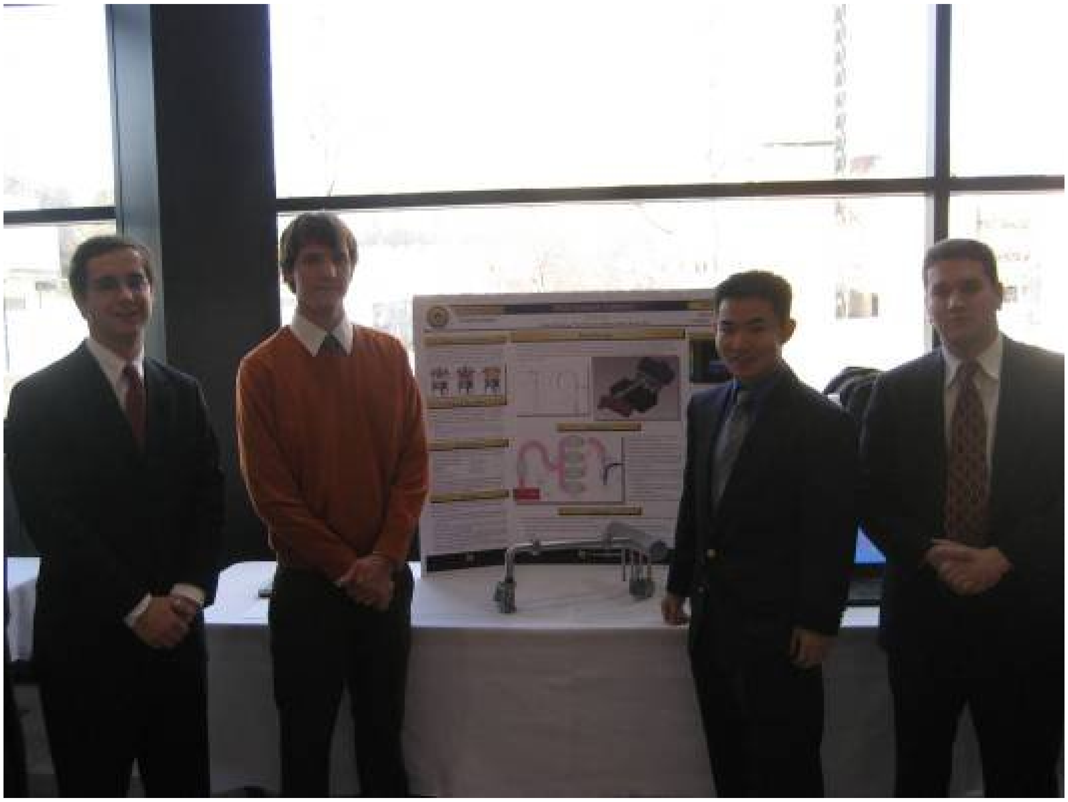
HCCI Engine Control



DESIGN REVIEW #5
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Executive Summary

The homogenous charge compression ignition (HCCI) is a promising new engine technology that combines elements of the diesel and gasoline engine operating cycles. It offers diesel-like efficiency without the high levels of NO_X and soot emissions of diesel engines. HCCI is particularly sensitive to cylinder-to-cylinder variations making uniform combustion difficult. We have been asked by the National Vehicle and Fuel Emissions Laboratory of the Environmental Protection Agency (EPA) to minimize this disparity by designing a control system to monitor and control a selected parameter.

We developed an exhaust gas recirculation (EGR) heating system which routes exhaust from the exhaust manifold directly into one of the two intake runners per cylinder. Using solenoid actuated control valves, we can then control exhaust gas injection with quick response times and high precision to control the intake temperature of each cylinder independently. Through temperature we have control over stability, power, and fuel consumption. Previously there was no means to control cylinders on an individual basis; therefore if one cylinder had surpassed its threshold of stability then all four cylinders would have to have their power reduced in order to stabilize the one cylinder. As a result, power was not being extracted as efficiently as it could have been compared to a system that could stabilize the cylinder independently of the other three. Our design successfully accomplishes this, allowing for higher power output while maintaining stability.

We had been informed that we were not going to be able to test our prototype due to time and financial constraints. As a result, the prototype is not functional however illustrates the final design. Validation had to be done to prove the functionality of our design. Computational fluid dynamic simulations were run to prove that we could accurately control temperature. A regression equation was used to show that the effects of a change in temperature outweighed the effects of a corresponding change in composition in terms of power and fuel consumption. Also further simulated combustion analysis was done to further investigate the effect of temperature on ignition timing and power. These validation methods have confirmed the feasibility of our design.

An in depth description of the manufacturing plan for our design will be outlined in the report. In addition there is a corresponding bill of materials; we found that the total cost for parts would be $1674.36. This is one drawback as this is a considerable addition to the total cost of an engine.

The concept of EGR and its application is not new; in fact it is already used on the test engine at the EPA to provide power to the turbochargers and to control composition. Our application of EGR however is a novel concept as we use the heat from the exhaust gas as a method for individual cylinder control. Since EGR is already a proven technology, it will make the implementation of our system easier. In addition, the manufacturing required is also not too extensive and can easily be adapted for other engines. We recommend that the EPA consider our design for implementation and experimentation.

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1. Introduction

The EPA has been researching Homogeneous Charge Compression Ignition (HCCI) engines. In HCCI combustion, the fuel is injected into the intake manifold (like gasoline engines) and ignites under compression in the cylinder (similar to diesel engines), giving a high efficiency and low emission burn. In multiple-cylinder engines, a noticeable difference in combustion strength has been observed between cylinders. The difficulty of an effective control method still remains, despite much research in this area. Our project will control one of the many parameters effecting intake temperature and combustion strength, thereby allowing for maximum efficiency and power without sacrificing much of the engine's energy to the device. This document will discuss all of these points in further details. This design review will also include information on our brainstorming process, our design concepts, and the method through which we will select our final design.

2. Problem Description

Increasing fuel prices have sparked a renewed interest in advanced engine technologies. One such technology is homogenous charge compression ignition (HCCI). HCCI combines elements of the diesel and gasoline engine operating cycles. It promises efficiency equal to or greater then a diesel engine without the high NO_x and soot emissions associated with diesels.

However, significant problems exist with HCCI technology. Chief among them is the issue of engine controls. To control combustion, numerous variables such as intake air temperature, intake air pressure (boost), exhaust gas recirculation (EGR) must be precisely controlled. This task is made harder by the inherent cylinder-to-cylinder variation in factors such as compression ratio, intake air turbulence and cylinder wall temperature. Our project seeks to actively control one or more variables in each cylinder to allow for simplified engine control.

This project is sponsored by the National Vehicle and Fuel Emissions Laboratory of the Environmental Protection Agency (EPA). Our desired outcome is an inexpensive and robust system which can compensate for the inherent cylinder-to-cylinder variation to improve engine stability.

3. Problem Analysis

Although HCCI is a promising technology, numerous technical challenges must be overcome before it is successfully implemented. The problem our group has been tasked to overcome is the issue of cylinder-to-cylinder variations. All multi-cylinder engines have inherent cylinder-to-cylinder variations. These variations arise from many factors, such as manufacturing tolerances, wear and the dynamic nature of gas-exchange and combustion within an engine. The consequence of these variations is cylinder-to-cylinder differences in intake mass, residual exhaust gasses, compression ratio and other factors. HCCI requires active and precise control of many variables (intake mass, intake temperature, boost pressure, air/fuel ratio, EGR etc). To effectively control these variables, one must be able to compensate for cylinder-to cylinder variations.

Several unique difficulties can be expected in the design of this system. First, the system must be designed to fit in the engine compartment. Although not an issue for our test engine (which is mounted in a dynamometer cell), commercial implementation can only occur if the system can be packaged within the engine bay. Similarly, the system must be able to cope with the temperature, vibration and electrical noise of the engine bay. Another difficulty is the speed of measurement of cylinder-to-cylinder variation. For example, for a 4-stroke engine operating at 3000 RPM, 1500 variation measurements of each variable (boost pressure, EGR etc) must be taken every minute for every cylinder. Apart from these unique difficulties, the system must be designed to be robust, inexpensive and easily manufactured.

Successful implementation of the design is contingent upon overcoming several technical problems. One such problem is the speed of actuation. To accurately control the variable, the actuators must complete their work during the intake stroke. This time period can be as little as 5 milliseconds. Fortunately, sufficiently fast actuators exist and are used on variable-valve-timing and variable intake-manifold systems. Also, there is the option of sensing and actuating over an average of several-cycles. Although this reduces the precision of the controls, the actuation time is much longer and the number of measurements is greatly reduced. Another challenge is independently controlling the variables for each cylinder. For example, numerous systems can actively vary the length of the intake manifold. However, there is no known system, which can vary the length of a single cylinder runner. To tackle this problem, we propose the use of individual actuators for each cylinder. Although doing so increases the cost and complexity of the system, it allows unparalleled individual control of a variable. Testing of the system presents another problem. Ideally, the engine would be tested on an engine dynamometer. Unfortunately, this is very unlikely. Accordingly, engine simulation software such as *GT-Power* and commercial computational fluid dynamics (CFD) software will be used to simulate engine operation. Moreover, specific tests can be completed without dynamometer or simulation software. For example, response time of our actuators can be tested using existing laboratory equipment.

To successfully complete this project, our team will need help in several areas. Testing and measurement is one such area. Currently, there are few metrics to judge the success of our design. Our goal is to simplify engine controls to improve engine stability. One metric that can be used is combustion pressure. The test engine is instrumented with pressure transducers in each cylinder. The peak pressure and pressure profiles of the cylinders can be compared to judge operational stability. Also, the output and response of the engine can be observed and measured over various loud conditions to judge stability.

4. Information Sources

Research was conducted to deepen our knowledge of the HCCI engine and to provide benchmarking concepts to aid our project. This research included an investigation into the fundamental principles of the HCCI engine. In addition benchmarking concepts were identified through an investigation into current research projects and relevant patents. This research allows us to begin on the development of original concepts to solve the problem we have been presented with.

4.1. Fundamental Principles

The HCCI engine is an internal combustion engine that achieves ignition through compression similar to a diesel engine; however fuel and air are already mixed before entering the combustion chamber similar to the spark ignition engine. This yields ignition at multiple places within the mixture providing a nearly instantaneous combustion. As a result the HCCI engine has relatively low combustion temperatures, since there is no flame propagation, resulting in minimal NO_x emissions while maintaining diesel-like efficiencies. There are inherent drawbacks such as limited operating range, troubles with cold starting, considerable hydrocarbon and carbon monoxide emissions. Perhaps the main challenge associated with HCCI is that it lacks a direct combustion trigger making ignition timing control difficult. Compounding upon this, multi-cylinder engines are even harder to control due to inherent cylinder-to-cylinder differences in combustion factors (e.g. intake temperature or compression ratio) making uniform combustion difficult. Of these two control issues, the latter is the problem we have been asked to address by designing a control system to minimize differences in cylinder-to-cylinder performance.

4.2. Research Findings on Control Methods

4.2.1. Cylinder-to-Cylinder Controls

Lawrence Livermore National Laboratory (LLNL) is currently working with the University of California Berkley on solving the problem of variations in cylinder-to-cylinder performance. They have found that cylinder-to-cylinder variations are greater at lower intake temperatures but have not yet solved the issue of controlling these variations as research is still in progress. [1] They are attempting to develop a thermal management system where a controller will manage cylinder-to-cylinder differences by adjusting the intake temperature of each cylinder. This is being achieved through control algorithms regulating the timing of the intake and exhaust valves. [2]

4.2.2. General HCCI Combustion Control Methods

Four main areas of timing control were identified in an investigation of the available literature: thermal control through exhaust gas recirculation (EGR), variable compression ratio (VCR), variable valve timing (VVT), and fuel mixtures or additives. [3]

EGR: By recycling exhaust gases and adding them to the intake air it is possible to control temperature, mixture, pressure, and composition. In comparison to the other control methods EGR is relatively simple, which is a great benefit. One drawback is that the system would have relatively low feedback response perhaps making it inadequate in transient situations. It should also be noted that the engine our project is focused upon uses EGR; however this does not mean that this is the only control system that could be implemented.

VCR: Could be achieved through a couple different methods. One method would be to place a plunger within the cylinder head that could vary the compression ratio. Another option would be to have an opposed-piston design which would include variable phase-shifting between the two crankshafts. Other possibilities exist as well but the key is to develop these in order to have excellent response time to handle transient situations.

VVT: Allows variation of the compression ratio not through geometric means but through timing of the opening and closing of the intake and exhaust valves. In addition, this system can act as a more direct method of EGR by controlling the amount of trapped residual gases thus allowing temperature and mixture control.

Fuel Mixtures & Additives: By using two fuels with varying combustion properties, combustion timing could be improved over a wide operating range. However a two fuel system brings about practical and infrastructural issues that could prevent commercialization. Experiments have shown that ozone as a fuel additive can greatly improve ignition even at very low concentrations.

4.2.3. Relevant Patents [4]

 # 7,025,042: A control system designed for a multi-cylinder HCCI engine with the goal of stabilizing combustion through speed and load transitions. This is done by adjusting some engine parameter (e.g. intake temperature or intake oxygen concentration) to control load with transitioning speeds. Further control exists by adjusting by the fuel delivered on a cylinder-to-cylinder basis. Each cylinder has its own closed loop fuel control to maintain uniform combustion.

 # 7,086,391: Provides a method to estimate the fuel/air ratio at all working points within the engine. This is done on a per cylinder basis for a multi-cylinder combustion engine allowing accurate and rapid measurements. This could be applied to a controller for adjusting fuel injection.

 # 7,066,158: An invention that utilizes EGR to expand the operational range and also includes a control method for ignition timing. Using a variable pressure supercharger and closing the exhaust valve before it reaches top dead center on the exhaust stroke, it is possible to control ignition timing.

4.3. Quantitative Benchmarking

While a number of concepts were identified to benchmark our project against, hard data is something that we could not find. We attribute this to the novelty of cylinder-to-cylinder combustion control.

5. Customer Requirements and Engineering Specifications

The customer identified the ten most important requirements for the project. These are listed in the QFD diagram in Figure 1. The requirements are ranked again taken into consideration the engineering parameters. The top requirements were response speed, reliability and increased efficiency. Table 1 shows estimated engineering targets for the project. Our engineering targets were updated as we gained a better sense of what are design will do and under what limitations. We also have updated the QFD to better incorporate these targets. However, it should be noted that these targets (with the exception of efficiency) have large uncertainties included since we have no real device to benchmark against.

Table 1: Engineering Targets

Thermal Efficiency	40%
Combustion Strength	Within 10% of each cylinder
Production Cost	$400
Number of Parts	< 30
Response Time	< 100 ms
Weight	< 40 lb

Figure 1: QFD Diagram

0	**Not Related**
1	**Weakly Related**
3	**Moderately Related**
9	**Strongly Related**

	Part Characteristics / Customer Requirements	Importance to Customer (Weight)	Relative Combustion Strength	Weight	Production cost	Size	Thermal Efficiency	Response Time	Number of Parts	TOTAL - CUSTOMER REQUIREMENTS	RANK
USER-PERCEIVED QUALITY	Low Cost	3		1	9	1			3	42	8
	More Efficient Than Currently Available	9	9		1		9	9		252	1
	Easy to Install	1		1	1	3			9	14	10
	Adaptable Over Time	6	9				9	9	3	180	4
	Light Weight	5		9		9				90	6
	Unobtrusive	2			1	3			3	14	9
	Programmable	7	3		3		9	9	1	175	5
	Able to Respond Quickly to Engine Varations	10	9		1		3	9	1	230	2
	Reliable	8	3				9	9	3	192	3
	Easy to Maintain	4	3		1		3	3	9	76	7
	Target Area		Within 10%	Under 40 pounds	$400	Able to fit under hood	40%	Under 100 ms	Under 30 Parts		

6. Concept Generation

To assist the concept generation process a FAST diagram was created (see Appendix C). The purpose of the diagram was to better understand the functions of our potential design. The task function is our primary goal: minimize cylinder-to-cylinder variations. The main two functions are sensing and controlling combustion variations. These functions were then more clearly explained by expanding the diagram. As a result we developed a group of parameters for control that we would use for the basis of our brainstorming session.

These control parameters included: compression ratio, intake/exhaust temperature, intake mass, intake air pressure, composition. We then developed as many concepts as possible that could potentially control these parameters. Care was taken to ensure no criticism was given during this process. Fourteen concepts were originally developed, two of which were quickly deemed impossible, resulting in twelve concepts that can be found in Appendix B. The ideas were organized into several major categories which are shown below with accompanying description reminders.

VCR (Variable Compression Ratio)
Changing the amount of compression for each cylinder would drastically change the engine characteristics. By incorporating a device that could change cylinder volume rapidly, individual control of each cylinder could be conceivably achieved. Many ideas using this concept were developed. An example of such a concept will be explained in the next section.

VVT (Variable Valve Timing)
Changing the timing of the intake/exhaust valve changes the amount of combustible air in the cylinder thus controlling combustion strength and timing. This could be used to change the cylinder performances individually if a good control method is found. It should be noted that typical VVT schemes run cylinders with set timing, whereas ours would need flexibility not only in timing but per cylinder.

EGR (Exhaust Gas Recirculation)
Exhaust gas recirculation is a concept originally developed to reduce emissions. The exhaust gas from an engine contains carbon dioxide and water along with some unburnt fuel. EGR can produce more power in an engine because more fuel could be pumped into the cylinder without spontaneous ignition due to the relative inertness of the emissions gas compared to air. It also could be used to control individual cylinder performance.

Fuel Mixtures
Mixing an additive into the fuel could change the combustion characteristics of a cylinder. Mixing a combustion retardant such as water would cause a delay in combustion, while mixing a combustion accelerant such as hydrogen gas would increase the speed of combustion. This could be controlled for each cylinder individually.

Temperature Adjustment
The effect of temperature on engine performance is well documented. In fact, a few degrees of difference in intake temperature can have significant effects on combustion strength. By varying intake temperatures for individual cylinders, combustion could be controlled.

7. Concept Selection Process

Once brainstorming and concept generation was completed, we then moved on to concept selection entailing: a discussion with our sponsor, Morphological and Pugh charts, and further research. Our sponsor helped us evaluate our designs based on feasibility and expected results. He also clarified that not only are we trying to minimize cylinder-to-cylinder variations but we are also doing this while achieving every horsepower possible from each cylinder. Therefore a control method that reduces the performance of three cylinders to match the fourth is unacceptable. Combining this discussion with results from Morphological and Pugh charts allowed us to select our five most viable options.

The purpose of the Morphological charts is to take functions from the FAST diagram (e.g. combustion control parameters) and then generate and evaluate concepts on their fulfillment of these functions. We developed two of these charts for the two necessary functions of our device: combustion parameter control and parameter sensing. Since our concept brainstorming session had been based on parameter control methods we already had the concepts needed for the first chart; results are shown in Appendix C. It should be noted that our goal is to successfully control cylinder-to-cylinder variations and not necessarily the most combustion parameters. For parameter sensing we developed a number of instruments capable of performing the functions and then evaluated their function fulfillment (see Appendix D). It should be noted that our parameter sensing instrument selection is highly dependent on our control method selection.

The purpose of the Pugh chart is to evaluate our control method concepts based on our customer requirements. Customer requirements were taken directly from the QFD diagram with the exception of "feasibility" which was a separate addition based on our most recent discussion with our sponsor. Weightings were also developed using the same information. Concepts were evaluated on a "-1" "0" and "1" basis for each customer requirement. For each concept the number of positive and negative scores were tallied along with a net score. The most crucial statistic was the weighted total, which formed the basis for rating the concepts. Using these ratings we determined our most viable concepts. Of the six top ranked concepts two of which were VVT schemes which we decided to forgo since our engine is not configured for VVT. Of the remaining four, further research was conducted; results are shown below in order of rank.

Concept 11: Plunger mounted in cylinder head to vary compression ratio
One variable compression-ratio (VCR) system that is under consideration is the compression plunger concept (Figure 2). By varying the stroke of the electro-pneumatically actuated piston mounted in the cylinder-head, significant changes in

compression ratio can be realized (Table 2). For an estimated piston diameter of 20mm, a piston stroke of ±3mm results in a change in compression ratio of approximately 1.2.

Figure 2: Ignition Piston and Actuator in Cylinder Head

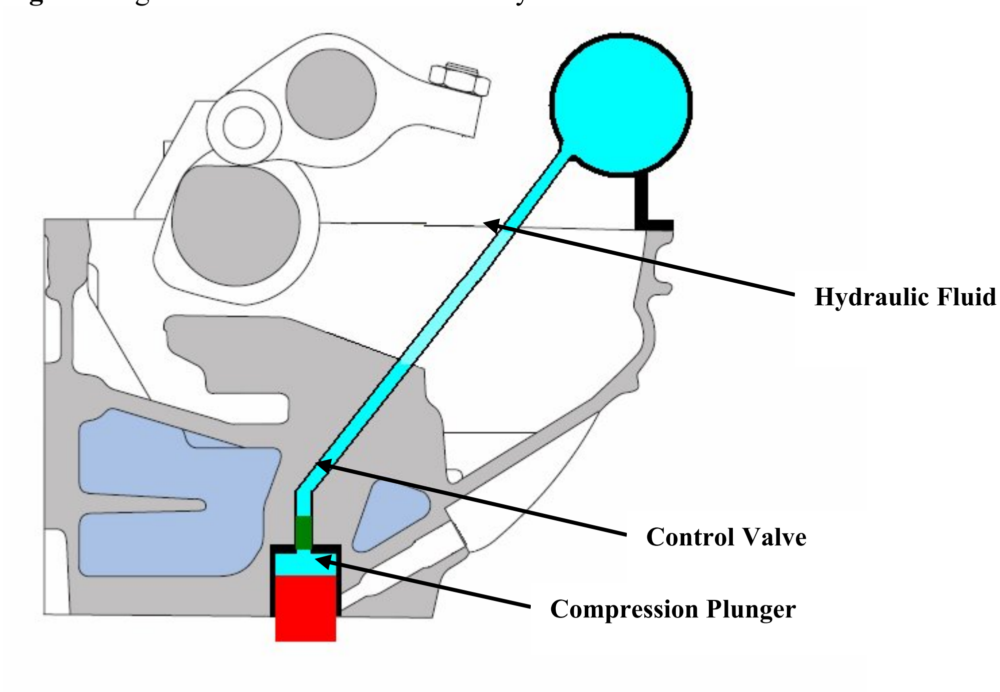


Table 2: Compression Ratio Variation

	Piston Stroke (mm)	*Compression Ratio*
Piston Diameter = 20mm	-3	17.44
	-2	17.63
	-1	17.84
Nominal Compression Ratio = 18:1	0	18.00
	1	18.19
	2	18.39
	3	18.59

The cylinder-head mounted compression plunger is a concept with many advantages. It uses mostly off-the-shelf components and is a relatively simple to implement VCR concept. However, major technical hurdles remain such as sealing and mounting issues. The advantages and disadvantages of this system will be described in detail.

Advantages

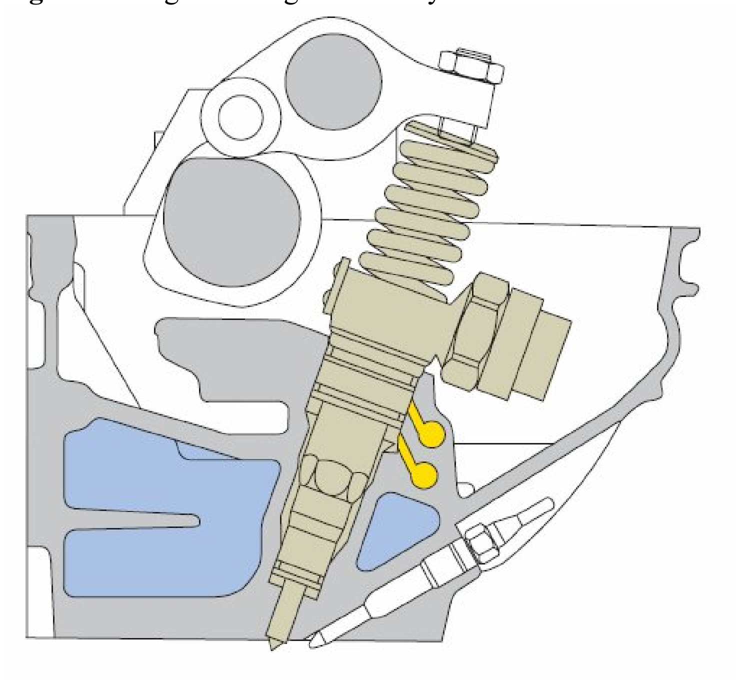
As noted before, the system has the advantage of using mostly off-the-shelf components. For example, a similar pneumatic system is used on certain race engines to help prevent valve-float at high engine speeds (15000+ RPM). Such a system would undoubtedly have a sufficiently fast response time for our compression plunger. Also, the pressure control mechanism could readily be adapted to our concept.

Another advantage is the simplicity of the system. Unlike other VCR concepts which use complex linkages to vary the piston or deck height, the compression plunger needs only a handful of components. Also, the hardware is mounted in a temperature controlled zone which does not move. Consequently, the thermal and mechanical stresses the hardware is subject to are less then other VCR concepts.

Disadvantages
A piston mounted in a cylinder-head presents several engineering challenges. First, the piston must seal combustion pressure from the cylinder-head. Previous ignition-piston designs have uses piston rings for sealing. These rings are components which would need to be custom made for out prototype.

An additional challenge is mounting the hardware. Most of the cylinder area is consumed by the engine valves and any fuel injectors or spark/glow-plugs. As a result, the area available to mount the hardware is very small. This naturally limits the piston diameter, which then reduces the compression-ratio adjustment range. In the case of the test engine, this is not a serious issue. Because our engine was originally a diesel, the space that would otherwise be occupied by a fuel-injector and glow-plug is now free (Figure 3). This area is where our compression plunger will be mounted.

Figure 3: Original Design of TDI Cylinder Head

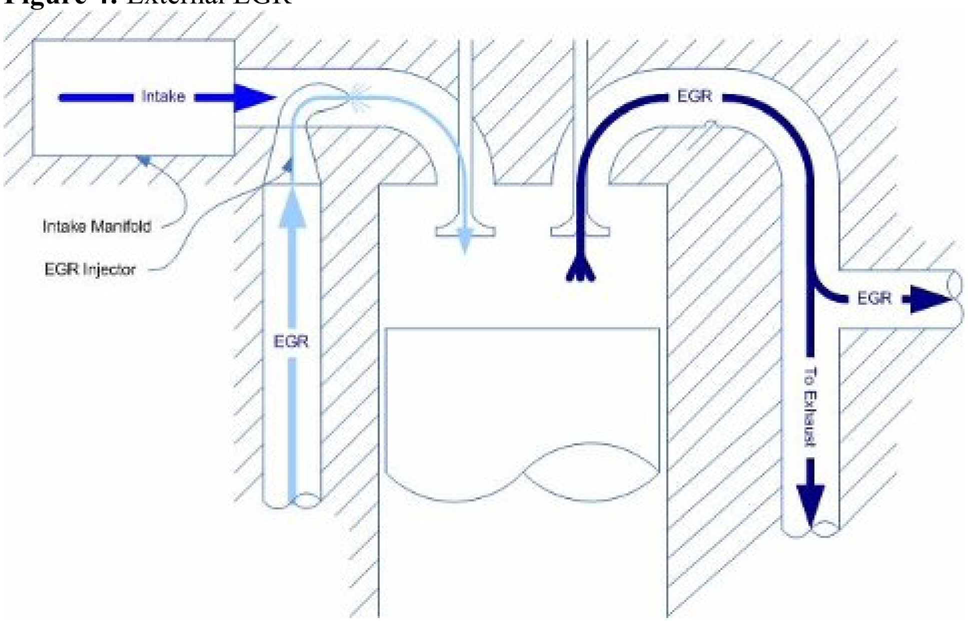


Concept 9: External EGR directed to individual runners
Although external EGR may be one of the more counterintuitive methods to control an HCCI engine, its technology could prove to be a great improvement over the current system. The way this works is by recirculating some exhaust gas back into the intake

manifold. In our system, there would be a control valve of each of the runners, which would allow for a specific amount of EGR to be re-emitted into the combustion chamber. Intuitively, one might assume that recirculating inert gases into the chamber would limit the combustion strength. However, theoretically, this inert gas will actually increase the combustion strength, by retarding the combustion speed, therefore getting power over a longer amount of time, thereby increasing the overall strength of the combustion.

There are two methods of getting EGR into the combustion chamber. The first is the method described above, or external EGR. The second is by varying the valve timing, such that the exhaust valve closes before a certain amount of EGR can escape, or internal EGR. This latter method is less effective however, because in this case the EGR is hundreds of degrees hotter than the external method. This further limits the amount of oxygen and fuel that can come in, and reduces the total amount of additional power possible. By recirculating the EGR externally, these gases are able to cool down significantly, while keeping the H_2O as steam. A diagram has been provided in Figure 4 below. Our engine is already set up for EGR which adds to the appeal of a control system utilizing EGR.

Figure 4: External EGR



The drawbacks of this system are that there is no way to control each charge that would enter the chamber. Instead, this system would have to work on a time averaged principle, meaning that if over time one cylinder is running too weak, more EGR could be introduced into the runner for that cylinder, in an effort to keep that cylinder running at the same strength as the other cylinders. Also, introducing the EGR into the runner may disrupt flow characteristics of the air-fuel mixture as it passes the injector. Further study of this problem could be investigated to better understand the effect of the injectors in the runners, and if it is indeed a problem.

Overall, the use of external EGR may be a very promising way to ultimately maintain stability for HCCI engines; either by itself or as a part of a more intensive system has the potential to make a great difference in the stability.

Concept 6: Compressor bypass valve (CPV) for each individual runner
A (CPV) or blow-off valve (BOV) is a valve used in turbocharged engines to prevent surging. This occurs when the throttle is closed and pressure build up occurs in the intake system forcing air back into the turbocharger ultimately yielding turbo lag. The valve then acts as a pressure release method to prevent surging and minimize turbo lag. A BOV is designed to release air to the ambient, whereas a CPV recirculates the pressurized air to the intake before the turbocharger; however the valve itself is essentially the same. Both are typically actuated by a spring, which is compressed when the throttle is closed and a vacuum occurs in the manifold. Since one design goal of our design is to maximize efficiency we would most likely use the CPV so that we are not wasting boost. However, the HCCI engine runs unthrottled and our purpose for the valve would be different.

Our concept would implement a CPV control system for each individual runner where the released air would then be recirculated to entry before the turbocharger. The idea is that since each cylinder has a different threshold for stability we could provide enough boost to maximize the performance for the top performing cylinder but through the use of a CPV we could limit the boost for cylinders with a lower threshold of stability. We were able to identify a bypass valve patent (#6,810,667) sharing the same fundamental concept, however we were not able to find any systems that would incorporate valves for each individual runner.

An advantage of this system is that bypass valves are already in production and would not be difficult to find. Since a four-cylinder engine would require four valves this would likely bring the price outside of our budget; however the price could be manageable for industrial application. A drawback is their implementation in individual runners has not been previously tested adding some uncertainty. For example, we foresee an issue in maintaining pressure variations between runners since the built up pressure from the manifold is going to want to remove any pressure gradients between runners; this would defeat the whole purpose of individual runner valves. Another disadvantage of this system is that the turbocharger would have to provide enough boost to compensate for the boost being recirculated by the valve system decreasing efficiency.

Concept 4: Heating element for intake air preheating
Prior EPA testing has indicated that slight intake air temperature variations yield large differences in the combustion strength of a cylinder. This design concept calls for a control mechanism to vary the temperatures in the individual intake runners, thus altering intake air temperature. The mechanism to change air temperature could be a heating coils or jacketed water cooling coil inside the intake manifold. A sensor of sort will have to be placed inside the cylinder to monitor the combustion strength. This data will be put into a direct control algorithm and control the intake temperature. This system would be adaptable to the constantly changing engine and be versatile across a wide operating range.

One of the advantages of this concept is that temperature is known to be a very effective criterion in changing the combustion strength. And data from past experiments support this idea. The intake temperature would only have to vary slightly for an effect to take place. This device would also be easy to implement, since it only requires changing the intake manifold and headers and does not change the engine significantly.

A disadvantage of this idea is that air flow inside the intake headers is not uniform, and determining the temperature of the overall air would be difficult. Changing the temperature of the air quickly and precisely could also be difficult due to the difficulty with air sensors and controlling the heating/cooling process.

8. Alpha Concept Selection Process

Of our four most promising concepts we wanted to further expand on two of them (external EGR and intake heating), while still considering compression ratio plunger. Both the EGR and intake heating concepts would utilize heat from different locations in the engine cycle to individually control and heat the intake runners. In this sense, both concepts are actually intake heating elements; however the EGR concept uses the injection of exhaust gases for heat, while the other uses hot intake air before it enters an intercooler for heat. Both systems would incorporate tubing from the desired location to the individual runners where a valve would control the injected mass flow rates.

Figure 5: Implementation would be in all runners

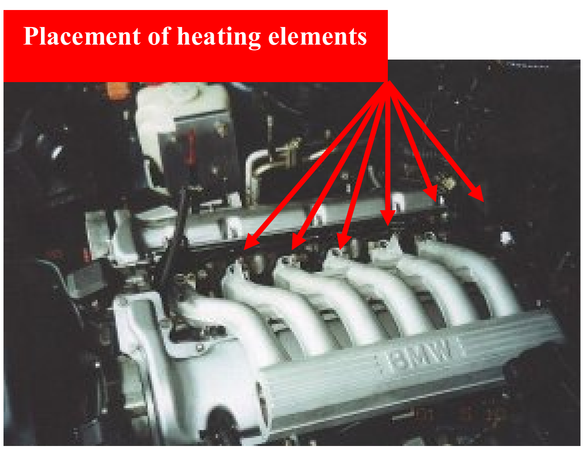
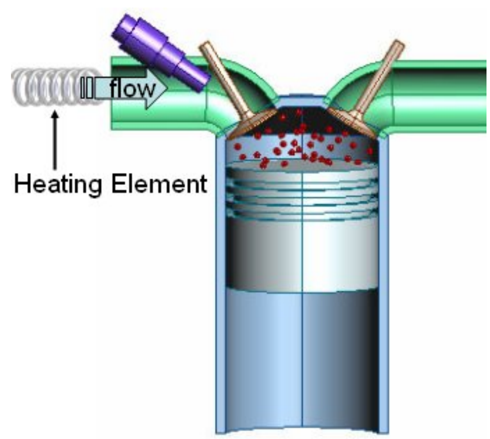


Figure 6: Heating Element in Intake



Originally it was thought that external EGR could be used for composition manipulation, however we found it would be more beneficial to use it as a heating method. This is because we can achieve desired temperature manipulation using a very small fraction of the exhaust gases compared to composition control, which would require much more exhaust to achieve comparable control. Since the energy in the exhaust gases are used to drive the turbines, which then drive the intake air compressors, we want to minimize the amount of exhaust gases used for intake runner injection.

Our other heating concept utilizes intake air after it has passed through the 2nd and final intake air compressor but before it passes through an intercooler. This location was chosen because it is the where the highest temperature occurs within the intake system and has a slightly higher pressure than in the intake runners providing the necessary pressure gradient for flow.

When comparing the two concepts a couple of concerns were identified: turbine power loss as a result of lost exhaust gas, mixed effects of the exhaust gas on temperature and composition, and an insufficient temperature gradient between compressed intake air and intake runner air. Analysis was done to compare the two concepts, which can be found in Appendix F on page 42. From our analysis it was determined that using the exhaust gases would be more promising since the exhaust gas is considerably hotter than the uncooled compressed intake air (~ 325°C) the mass needed to increase individual runner temperatures would be 4-5 times less. Also because only a minimal amount of exhaust gas would be needed for heating, power losses and effects on composition are negligible. An unforeseen issue with using the compressed intake air was the effect on mass flow rates into the cylinders not requiring heating thus complicating control. Another advantage of exhaust gases is that it has a pressure of ~100 kPa more than the intake manifold pressure, compared to only a ~ 3 kPa gradient for compressed intake air. This extra pressure could be used to facilitate the injection process.

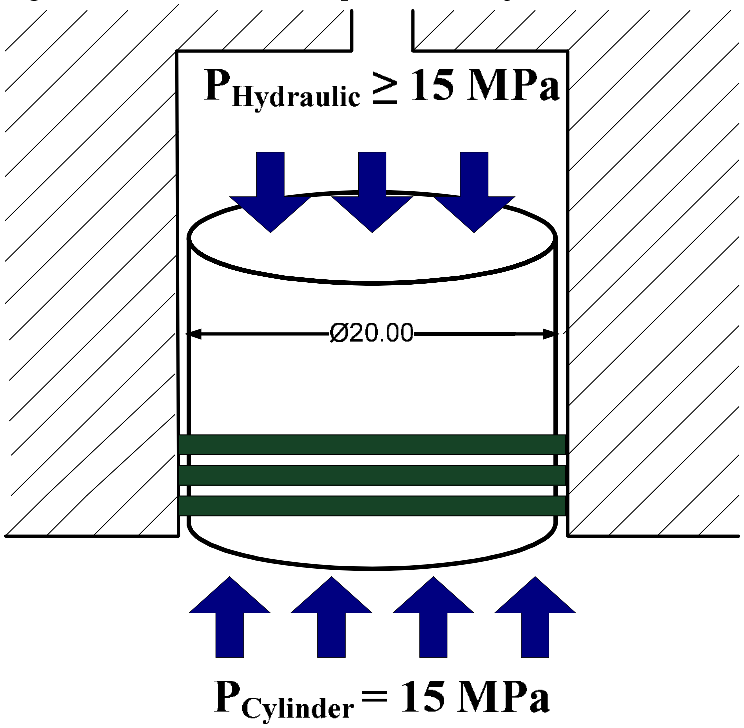
Once we had determined that using the exhaust gases for heating would be better than compressed intake air, we then compared this idea to our plunger concept (see Figure 7 on page 15) to decide on a final design. For a piston diameter of 20mm, a stroke of ±3mm results in a change in compression ratio of approximately 1.2. This concept has the advantage of using many off-the-shelf components and is, at first glance, a relatively simple VCR concept. However, several technical challenges have led us to disqualify this concept.

A main challenge would be sealing the plunger. The plunger must seal combustion gasses from the cylinder head as well as sealing pressurized hydraulic fluid from the combustion chamber. In our conceptual design we proposed using piston rings to seal the plunger. This is a problematic solution from several standpoints. First, some combustion gasses will inevitably slip past the piston rings. Consequently, a system would have to be developed to purge these combustion gasses from the hydraulic system and/or cylinder head. Conversely, some hydraulic fluid could also slip past the rings into the combustion chamber. This would lead to loss of hydraulic pressure and therefore affect plunger actuation. Lastly, a system would have to be devised to lubricate the piston rings.

An additional challenge is the development and mounting of the hydraulic system. This system would need to be designed from scratch and mounted around the existing cylinder head components and the area available for mounting is limited. Furthermore, this area experiences considerable temperature gradients and the system would need to be designed to cope with the thermally induced stresses. In addition to the thermal stresses, mechanical stresses would be induced by the pressurized hydraulic fluid.

Although the compression plunger idea has merit, the decision was made to disqualify this concept. The work necessary to develop the hydraulic system, build the components, and validate the concept is simply too large for our semester project. In light of the time and financial constraints of this project, the concept was determined to be unfeasible. Therefore, EGR intake heating is our chosen design.

Figure 7: Schematic of Compression Plunger



9. Alpha Concept: EGR Intake Heating System

With a final design selected, we then had to determine exactly how it would be implemented, manufactured, and validated. Analysis was done to determine design parameters such as material selection and dimensions. A description of our design and our prototype was then developed, followed by an initial manufacturing plan and validation approach.

9.1. Engineering Design Parameter Analysis

To determine design parameters of all the necessary components, analysis had to be done to determine the injection flow rates that will be required to produce the desired temperature change. We were supplied with data from the EPA, running the engine at 1600 rpm and 11.3 bar brake mean effective pressure (BMEP). Since there are too many load and speed combinations to possibly analyze, this running point was chosen because it is a good representative of ideal system operation. Based on this data, required mass ratios were determined for intake runner temperature increases of 1°C and 5 °C. An assumption here is that, while solving for only one cylinder, we can adequately account for heating up to three cylinders (since at no point would all four require heat). This means that, for example, a 6°C increase in one runner is equivalent to a 2°C increase in three cylinders. This is due to the linear relation between the effect of adding mass and the increase in temperature as shown in Appendix F. Results are shown below:

For one degree increase: $m_e = 5.9\text{x}10^{-4} m_a$
For five degree increase: $m_e = 0.0030 m_a$

where m_e is the mass of the exhaust used for injection, and m_a is the intake air. Since stability, fuel consumption, and power are highly dependent on intake temperature, we think an equivalent five degree increase should be sufficient for most situations; however a worst case scenario would be 10°C, still only requiring an injected mass that's 0.6% of the intake mass.

Since the previous results are based on steady state values, we had to do analysis to determine the values on a rate basis. We were provided with flow rate data of the incoming air, oxygen, and EGR. Analysis can be found in Appendix G. The required mass flow rate would be 0.127 g/s and the corresponding volume flow rate would be 0.126 L/s. It should be noted that analysis was done assuming complete combustion and stoichiometric conditions which we determined was reasonable since the exhaust oxygen percentage was 0.67%. These flow rates will allow us to determine the valve characteristic when we choose to buy our selected valves.

For determining the types of components needed, we knew that we needed tubing to transport the exhaust to the individual intake runners. We decided that even though there are two runners per cylinder, injecting into only one will be sufficient and more importantly will lower cost and parts. Our main concern with tubing was that it could withstand the high temperatures of the exhaust gases. Also the tubing used to connect to the intake runners had to be small enough to fit onto the intake runners which have a 0.9

in outer diameter. We had hoped to use aluminum because of the ease of manufacturability and welding associated with its material properties. We determined that we would have to use 304 stainless steel in order to withstand the exhaust temperatures of close to 750 K. To minimize heat transfer from the recirculation system, our final design calls for a ceramic thermal coating, this was also used on our prototype.

In discussion with Automotive Research Center, it was determined that a reservoir would be required in order to have a relatively constant and temperature and pressure source for injection. By including a reservoir, the control aspect is simplified because it doesn't have to deal with constant changes in pressure and temperature. We decided on a cylindrical volume with a length of 1 ft and a diameter of 2 in.

In addition to piping, the other main components are the valves which will control flow. We will be using a total of five valves, one inlet valve on the exhaust manifold used to control the pressure in the reservoir. From the reservoir there will be four EGR runners each with its own valve controlling the injection process. Our main concerns with the four injection valves were their ability to withstand high temperatures and have very fast response times. In order to get response times needed to inject only when the intake valve was open, we had a maximum response time of 0.075 s. There are solenoid actuated valves that can accomplish this response time and withstand high temperatures. For the pressure regulating valve, our main concerns were ability to monitor pressure effectively and also to withstand high temperatures. There are already EGR valves on the market that can accomplish these goals.

9.1.1. DFMA

To aid our design analysis, we used design for manufacturing and assembly (DFMA) guidelines. Our five selected guidelines are shown below

DFAS – 1: Minimize part counts
DFS – 1: Eliminate fasteners
DFMC – 2: Use standard stocks
DFMC – 3: Use standard dimensions
DFMC – 9: Place holes away from corners

For the first guideline, we had originally planned on injecting exhaust gases into each runner for each cylinder; we realized that we could still achieve adequate control by just injecting into one runner per cylinder. For the second guideline we had originally planned on using clamps for piping attachment, however we have decided to use welds instead. For the third and fourth guidelines we decided to use standard stocks and dimensions to eliminate any custom needs. For our final guideline we are designing our valve locations to be sufficiently far enough away from our 90° piping turns.

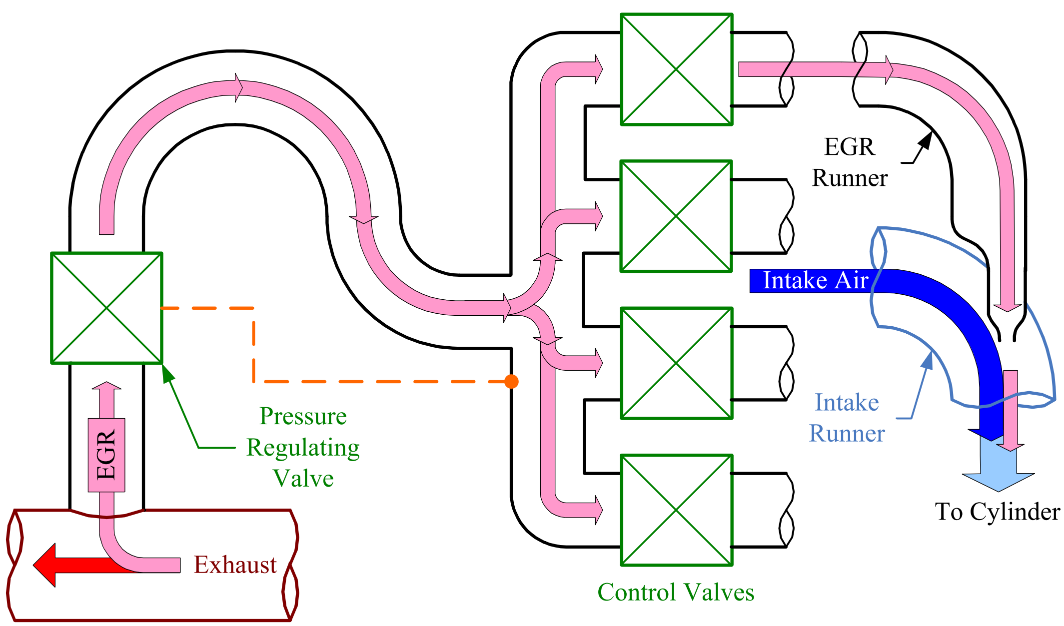
9.1.2. FMEA

A Failure Mode Effects and Analysis study was conducted on the system. The chart can be found in Appendix H on page 35. Although no problems had an RPN of over 100, the rolling top 25% included problems that mainly related to pipe connections. For instance, with a potential failure mode as an EGR leak, the highest RPN comes from the potential for breaks in the welds and other applied attachment methods. These will be further review and studied, to ensure that the best possible attachment methods are used in an effort to lower the RPN numbers in these areas. Additionally, the other larger RPN's come from failures in the valves. This is hard for us to control, considering that we will buy these from a manufacturer and not fabricate them ourselves. We will focus mainly on ensure the weld qualities needed to ensure no pipe failure modes.

9.2. Concept Description

The stability problem inherent with HCCI engines is to be alleviated by supplying a variable amount of EGR to each intake runner. By choosing the amount of EGR per intake runner, it will be possible to maximize the amount of power and stability per cylinder, thereby increasing the overall power and stability of the engine as a whole. The system that has been designed to utilize this principle can be seen in Figure 9.

Figure 8: Schematic of the Overall EGR System



There are two main parts to this design: heat addition and EGR recirculation. The EGR in this system is to be taken directly from the exhaust manifold. This provides our system with the necessary exhaust gases and heat differential. It will be attached to the exhaust manifold by drilling a hole in the exhaust manifold and welding a tube to protrude at a right angle from the manifold. The piping of the system will be connected to a pressure control valve, which will run off of a sensor in the splitter further downstream. When the pressure exceeds the desired pressure in the splitter, this pressure control valve will shut off until the pressure drops below the desired pressure. At that time, the valve will re-open, and operate on a closed-loop control to maintain pressure downstream. After the splitter, each runner will have a one-way valve, which will reduce the amount of backflow in the system. Each runner from the intake manifold will have a stream of EGR, which will be re-emitted into the system by an open nozzle. The nozzle will be directed downstream as to reduce the amount of EGR free to travel backwards through the intake system. Finally, in each runner of the EGR system, there will be a proportioning valve designed to only let a certain amount of EGR to be recirculated.

It has been empirically shown that there is a direct correlation to the amount of EGR reintroduced into the cylinder and the power provided by the cylinder; the more EGR recirculated into the system, the more power the engine can produce. However, if too much EGR is recirculated there exists a point at which the system will fail. This point is different for each cylinder. In the proposed system, this max power point could be achieved for each cylinder, thereby maximizing the power availability from the entire engine.

Further, tests have been run that show that the addition of heat into a system directly increases the stability of the engine. Unfortunately, this comes at the price of lowering the overall power of the engine. By adding heat in the form of hot EGR, we would be able to raise the stability of the engine, while making up for the power losses associated with the heat addition. It should be made clear at this time that this proposed system is an auxiliary system to the current dynamometer setup at the EPA's test lab.

9.3. Prototype Description

We anticipate manufacturing our project as described in the design description; however we see a couple of potential issues. Our analysis shows that parts procurement may be delayed, but we must discuss this further with our sponsor. It is also unsure at this time where we would be able to do the welding of our project, as well as an uncertainty involving the quality of the welds themselves. It is most likely that we would have to purchase some aluminum piping that we could tap for valve insertion. Another way to eliminate cost but still portray our concept would be to only build a prototype for a single cylinder control, therefore we would not be requiring a split into four pipes with four corresponding valves. CFD simulation will also be used to show the capabilities of our injection system since actually installation on the test engine at the EPA is unlikely. In summary, it is unclear at this time how the final prototype will look like. It is quite possible that it will solely be a representative mock-up of out system, and may lack the needed temperature resistance and functionality the real system would need.

9.4. Final Design Description

The concept of our design remains intact since our last design review; however since then we have taken this concept and finalized it by choosing the appropriate parts and materials and we have laid out a plan for assembly.

9.4.1. Bill of Materials

The bill of materials can be found below in Table 3. The total cost of parts was found to be $1674.36, which shows why it would have been impossible to create a functional prototype with a budget of $400. We have utilized standard sizes to ease the manufacturing and assembly process.

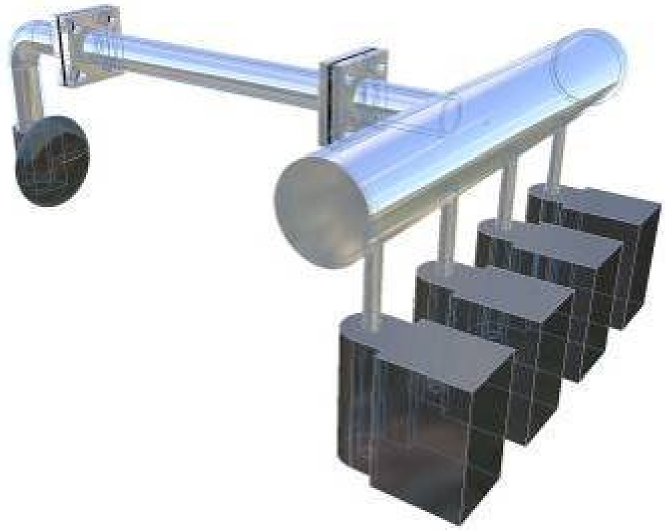
Each part has been chosen for a reason, either motivated by cost or quality. The bolts, nuts, and washers were chosen to ensure a tight fit of the flanges even at high pressures and temperatures. All the tubing, end cap, and flange material has been selected as stainless steel-304 to ensure high strength even at the high operation temperatures. The Gaskets were chosen because of their high resistance to heat transfer. It was important to limit the amount of heat transmitted through the metal of the pipes, and these gaskets provide a high level of thermal insulation. The Ceramic-Metallic Thermal Barrier Coating helps reduce the amount of heat transfer lost to the ambient, and is resistant to temperatures up to 1700°F. The EGR Valve kit from Mopar was selected, because it includes the vacuum line, which most EGR Valves do not include. Since this is used in its intended application, there is no question that it will work in these conditions. The Burkert 255 2/2-Way High Temperature Solenoid Valves were selected because they were the least expensive valve that would respond in 10-20 milliseconds as well as withstand the temperatures of the medium traveling through it. Lastly, the stainless steel compression fittings were chosen because of their high integrity of strength especially at high temperatures.

Table 3: Bill of Materials

Quantity	Part Description	Purchased From	Part Number	Price (Each)	Price (Total)
4	1" SAE Grade-8 3/8" Hex Bolts	BoltDepot.com	613	$0.28	$1.12
4	SAE Grade-8 3/8" Washers	BoltDepot.com	3049	$0.16	$0.64
4	Stainless Steel 18-8 3/8-16 Flanged Nuts	BoltDepot.com	7405	$0.35	$1.40
1	3" to 3" 90° Elbow 1" Stainless Steel-304	Woolf Aircraft Products	100-065-100-090-304	$17.00	$17.00
1	2" Straight 1" Stainless Steel-304 Pipe	Woolf Aircraft Products	100-065-304	$7.50	$7.50
1	2.5" Straight 3/8" Stainless Steel-304 Pipe	Woolf Aircraft Products	038-065-304	$7.50	$7.50
1	12"x2"x3/16" Stainless Steel-304 Plate	ALRO Metals Plus	17501380	$7.68	$7.68
1	12"x3"x1/8" Stainless Steel-304 Plate	ALRO Metals Plus	17500440	$7.93	$7.93
1	Exhaust Gasket Sheet (500x500mm)	GasketSheet.com	WLT-EGR-02	$20.00	$20.00
1	Ceramic-Metallic Thermal Barrier Coating	ThemalCoatings.com	--	$100.00	$100.00
1	EGR Valve Kit	Mopar	B7010-148269	$53.57	$53.57
4	Burkert 255 2/2-Way High Temp Solenoid Valve	Valin.com	452909F	$223.77	$895.08
3	Stainless Steel Compression Fittings 1"	AdaptersPlus.com	6504-100-100-SS	$41.98	$125.94
12	Stainless Steel Compression Fittings 1/4"	AdaptersPlus.com	6504-025-025-SS	$35.75	$429.00
				Total:	$1,674.36

9.4.2. Manufacturing Plan and Engineering Changes Notice
A detailed manufacturing plan can be found in Appendix I and our design changes since Design Review 3 are presented in Appendix J. We have included a final CAD drawing in Figure 10 below. Part by part engineering drawings can be found in Appendix K.

Figure 9: CAD Drawing of EGR System



10. Validation Results

Ideally, we would have liked to implement our design on the engine at the EPA; however this was not possible due to the time, money, and alterations that this would require. Therefore, we decided that to adequately validate our design we needed to show that (1) injecting exhaust gases would result in the desired in-cylinder temperature, (2) that the effects of change in temperature would outweigh the effects of change in composition, and (3) that temperature is a sufficient method for controlling power. We used Fluent®, a computational fluid dynamics simulation, to model the injection of exhaust gas into the intake runners and the effects on temperature. To compare effects of the temperature and composition on output power we used a regression equation provided to us by the EPA. Finally, Chemkin, an analytical combustion simulator, was used to gather more in depth information on the effects of temperature on output power.

10.1. Computational Fluid Dynamic Simulation

A 2-d model was set up using Fluent to model the injection of exhaust into an individual intake runner and how this impacted the intake runner temperature. Based on previous calculations we had theoretically determined that using a mass flow rate of 0.1267 g/s would achieve an in-cylinder temperature of 5° C, based on exhaust and intake temperatures provided to us by the EPA at an operating point of 1600 rpm and 11.3 bar brake mean effective pressure (BMEP). Our results from Fluent, using the proposed mass flow rate, showed that the exit temperature of the individual runner would result in an average increase of 9° C; however since there are two runners per cylinder this would effectively result in a 4.5° C increase of the in-cylinder temperature, assuming sufficient mixing within the cylinder. This therefore validated our prediction of a 5° C increase within 10%. A temperature contours plot and a graph of the average exit temperature with respect to the exhaust injection are shown below.

Figure 10: Temperature contour plot shows location of injected exhaust gas at the end of the injection period (temperature in degrees Kelvin).

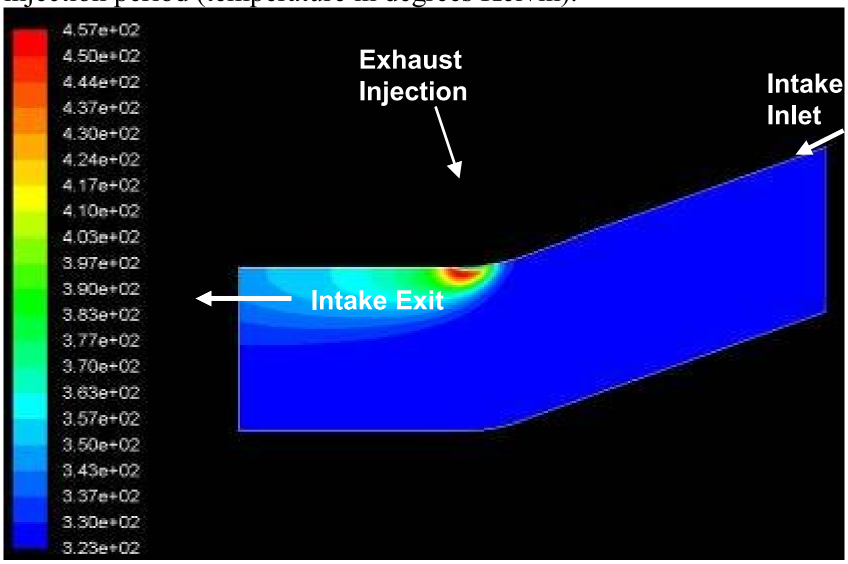


Figure 11: Injected exhaust flow rates and average intake exit temperature as a function of time. One full cycle takes 0.075 seconds corresponding to ~1600 rpm**.**

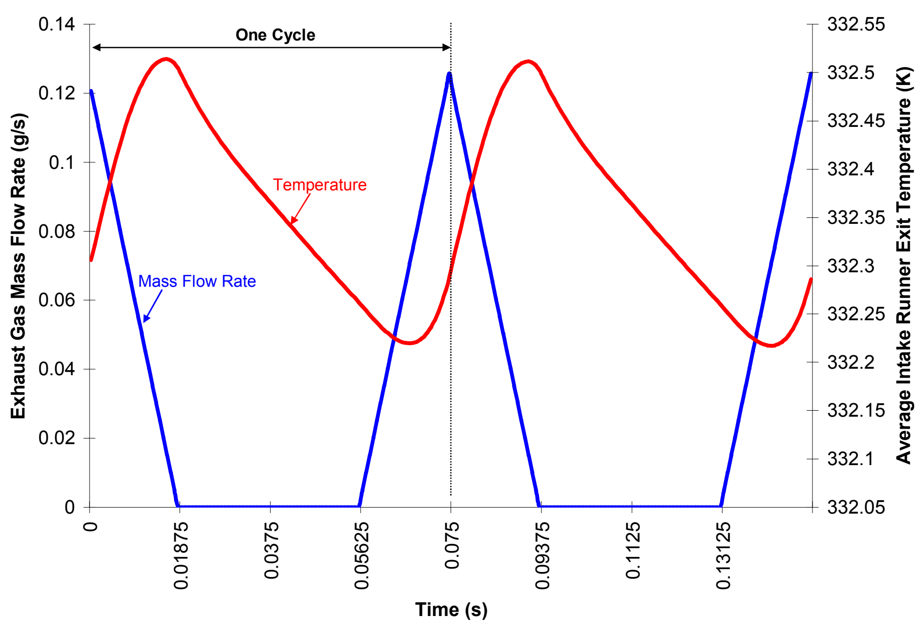
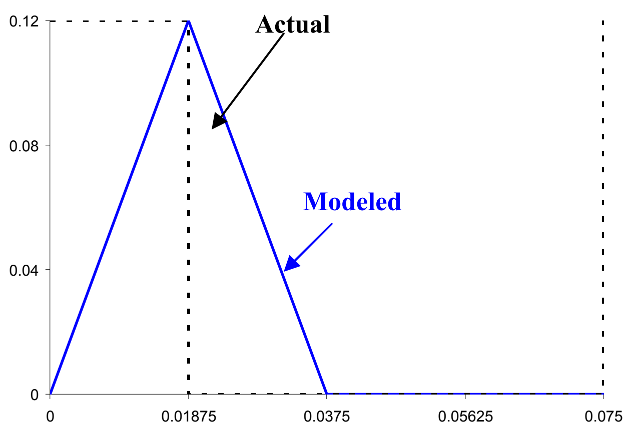


Figure 10 shows the extent of the temperature mixing. The injection diameter was set to 0.375 in; as evident in the temperature contours plot, the flow of the intake air overwhelms the exhaust injection which doesn't allow for ideal convection heat transfer.

Decreasing diameter size could further improve the mixing process; however with a diameter of 0.375 in we had achieved our average exit temperature within 10% of the desired.

Figure 11 shows the cyclic variation of temperature at the intake runner exit with respect to the varying injected mass flow rate of exhaust gas. Since Design Review 3 it was brought to our attention that injected continuously could result in backflow into the intake manifold while the intake valve was closed, which would reduce the effectiveness of our temperature control. To deal with this problem we determined that we would achieve better control by injecting only when the intake valve was open. For the operating speed of 1600 rpm, we determined the time of one cycle: 0.075 s. This corresponds to the intake valve being open for 0.01875 s. A user-defined profile was used to achieve the effect of injecting only when the intake valve was open. However, due to our limitations in our Fluent experience we could not model instantaneous injection which is why it appears that injection takes place for twice as long. This is better shown in Figure 12 where the actual injection process would immediately drop off at 0.01875 s, and the modeled has a triangular shape. However, the code was written to ensure that if the curves were integrated over one cycle, the same total mass injected would be the same. As a result, this could overestimate the temperature when the intake valve, but since the variation in temperature is so minute, ~ 0.3° C, this overestimation would be insignificant.

Figure 12: Difference between actual and modeled injection processes.



A number of assumptions were made in order to run the simulation. We assumed air as the working fluid, which we determined a good representation based on the composition of the intake air and the exhaust gas; however it does not take into account the presence of injected fuel. Unfortunately, to model this would have been beyond our knowledge of Fluent. Dimensions of the intake runner and intake air mass flow rates were supplied by the EPA. A realizable k-ε turbulent model was used with an unsteady solver to sufficiently model the turbulence and transience. Relaxation parameters were left as default. We also have neglected the heat transfer to the ambient through our EGR system, which will overestimate the injection temperature and thus the amount of heat added to the intake runner. Ideally a 3-d model would have been used but due to time constraints and lack of familiarity with Fluent we were unable to achieve this; however for symmetric flows, such as ours, a 2-d model gives accurate results.

10.2. Effects of Temperature and Composition on Power and Fuel Consumption

The EPA developed a regression equation to model the effects of temperature and composition on BMEP and fuel consumption. An increase in intake air temperature results in higher stability and a decrease in power, using a higher percentage of inert exhaust gases will result in less stability and an increase in power. Therefore, when adding exhaust gas the effects of composition counteract the desired effects of adding heat. Using the regression equation, we determined that the effect of temperature on BMEP and fuel is 10.9 times greater than the effect of the corresponding change in composition. Since the effect of temperature is significantly greater than composition we can conclude that our concept is still viable, however we will have to take the counteracting effect of composition when determining desired flow rates.

The regression equation tells us that for a change of 5° C of the in-cylinder temperature will result in a change of the BMEP and fuel consumption by 2.5%. This corresponds to our exhaust injection flow rate of 0.126 g/s. Taking into account a counteracting 0.23% change in BMEP and fuel consumption by the change in composition we can deduce that the overall change is ~2.3%. From discussion with our sponsor, we don't foresee requiring more than a 5% change in BMEP and fuel consumption, corresponding to a maximum flow rate of 0.278 g/s. This is the critical value used when we determined appropriate parameters and dimensions of our system.

10.3. Combustion Analysis

Apart from the CFD simulations to model mixing of the gasses, we needed to quantify the effect of increasing intake temperature on combustion strength. Initially, our group sought to use GT-POWER® to simulate engine operation. However, because GT-POWER is a comprehensive engine simulation program, the amount of input data it requires was not available to the group. Consequently, the combustion simulation program CHEMKIN® was used to model HCCI during the compression and expansion stroke. CHEMKIN is a zero-dimensional, thermokinetic combustion modeling program. It uses a detailed thermodynamic code which incorporates 897 chemical species and approximately 3600 chemical mechanisms for isooctane combustion. The simulation assumes a closed cycle which begins at intake valve closing and ends at exhaust valve opening. Heat transfer is accounted for through an empirical correlation based on engine geometry, cylinder wall temperature and cylinder gas velocity.

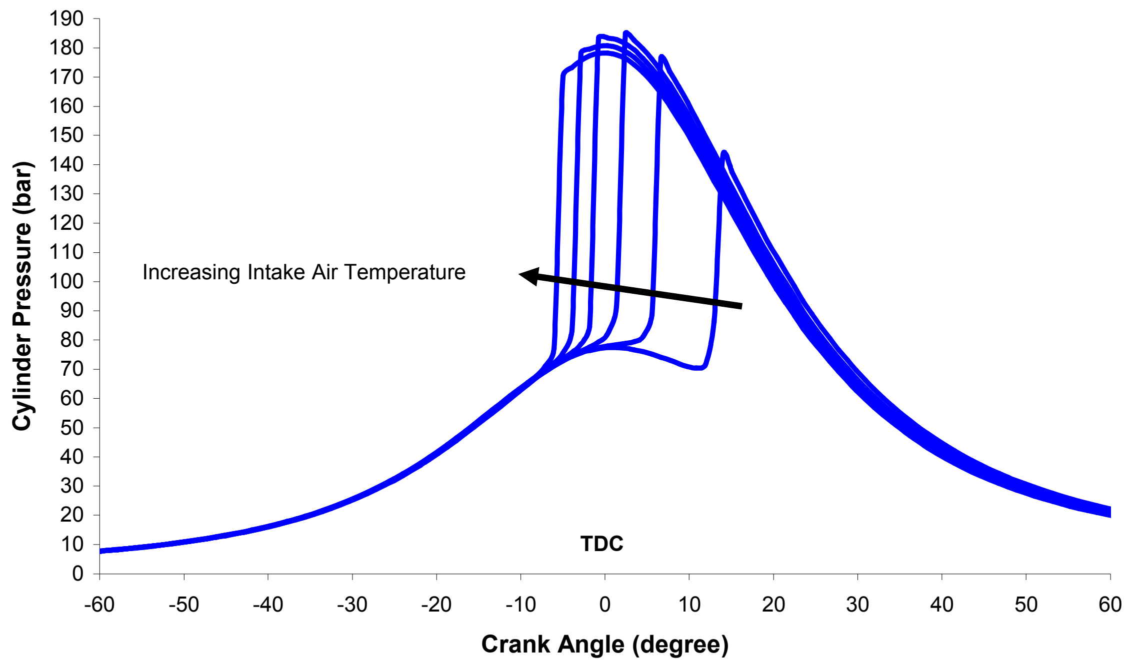
The simulation was tailored to our engine by specifying the engine geometry, engine speed, compression ratio, valve timing (intake valve closing and exhaust valve opening), intake mixture composition and inlet pressure/temperature conditions. These inputs are listed in Table 4. The numerical results of the simulation were visualized as a cylinder pressure trace over the duration of the compression and expansion strokes. This pressure trace was then numerically integrated and normalized by the cylinder displacement to determine the gross indicated mean effective pressure.

Table 4: Engine Information

Engine Speed:	1600 RPM
Cylinder Bore:	79.5mm
Cylinder Stroke:	95 mm
Compression Ratio:	16.5
Crank Radius/Connecting Rod Ratio:	3.2
Intake Valve Closing:	156° BTDC
Exhaust Valve Opening:	152° ATDC
Intake Pressure:	2.05 bar (absolute)

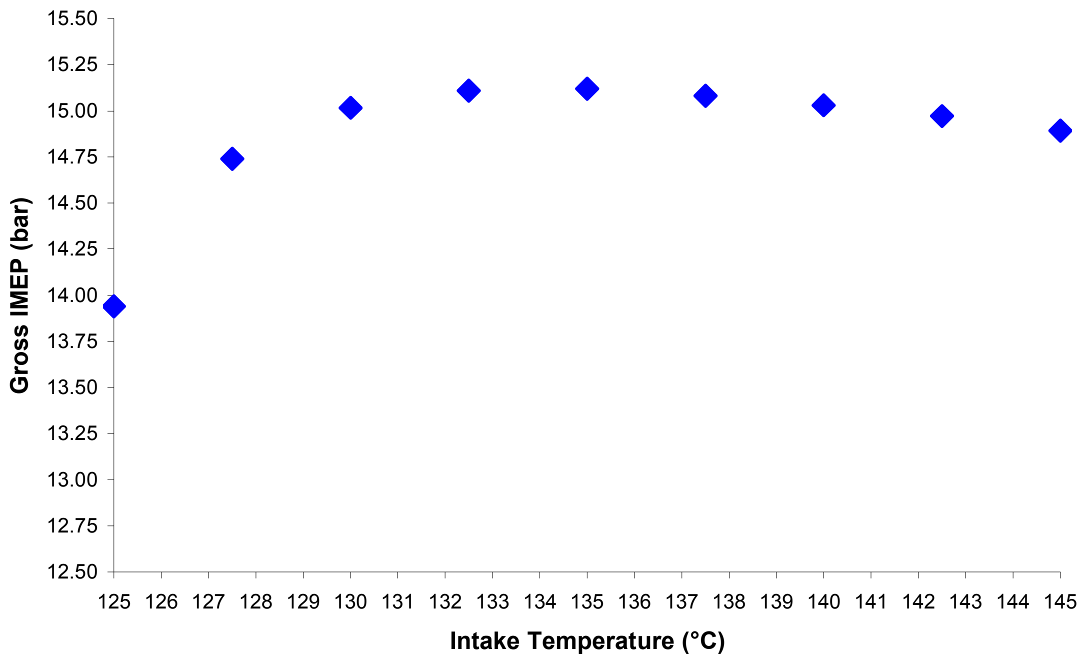
The effect of increasing intake air temperature on ignition timing is shown in Figure 13. It is clear that increasing intake air temperature causes ignition to be advanced. Whether this is beneficial or detrimental depends on the actual ignition timing. The CHEMKIN model predicted that the ideal ignition timing corresponds to the piston being just before top-dead-center (TDC). Advancing ignition timing before TDC causes combustion to occur as the piston is still traveling upward. Consequently, ignition occurs before the compression stroke has finished, reducing the maximum compression of the working fluid prior to combustion. This causes a minor reduction in the power generated by the expansion stroke. Conversely, if ignition occurs after TDC, the duration of the power stroke is reduced because the working fluid is allowed to expand before ignition. This again causes a reduction in the power generated by the expansion stroke.

Figure 13: Cylinder pressure traces show the advance of ignition timing with increased intake air temperatures. The peak pressure trace corresponds to 135°C intake air temperature.



In Figure 14, we see the effect of increasing intake air temperature from 125 to 145°C on gross indicated mean effective pressure. Gross indicated mean effective pressure (IMEP) is defined as the work delivered to the piston over one compression and expansion stroke. The negative work associated with pumping the working fluid into and out of the cylinder is not accounted for. One can easily see that peak IMEP occurs at an intermediate temperature. In this case, peak IMEP occurs when the intake air temperature is 135°C which corresponds to ignition at approximately 1° before top-dead-center. Although it is desired to have ignition occur just before TDC, this is not always possible due to the dynamic nature of gas exchange. If ignition must occur at another time, it is better to heat the intake and cause ignition to occur before TDC. We see that the reduction in gross IMEP by slightly advanced ignition timing is considerably less then the loss in gross IMEP with slightly retarded ignition timing.

Figure 14: The effect of increasing intake air temperature on gross indicated mean effective pressure at 1600RPM, 11.3 bar BMEP operating condition. Peak IMEP corresponds to combustion at approximately 1° BTDC.



Several important modeling assumptions should be mentioned. First, isooctane was assumed as the fuel. Because gasoline is a mixture of hydrocarbons of slightly varying formula, direct modeling can be difficult. Isooctane is sufficiently similar in composition and chemical properties to be used as a close approximation. A second key assumption is that combustion is clean and complete. Accordingly, no carbon monoxide is formed, no dissociation of nitrogen occurs and no unburned hydrocarbons are recycled into the intake. This allows for vastly simplified intake conditions. The existence of small amounts of the aforementioned compounds does not appreciably affect the simulation of combustion. These compounds do however affect engine emissions, which is a topic beyond the scope of our project. A final assumption is our use of a zero-dimensional model. This assumes that combustion occurs in a single homogenous zone and that fluid dynamics does not affect the combustion event. Fluid dynamics is only taken into account in our heat transfer model. The details of our heat transfer correlation are explained in Appendix I.

11. Project Plan

This project has a number of milestones that are set forth. The most important deliverables are the Design Reviews #1-4, the presentation at the Design Expo, and the Final Report. These items are outlined in the Gantt Chart in Appendix A. These items will also be updated as more details are determined.

For the first Design review, we have researched the topic of HCCI engines and what is currently being done to enhance this concept. As of now, our group is focusing its efforts on figuring out which characteristics of the engine can be modified, and ultimately controlled in real-time. Once these topics are decided on, we will develop different concepts that will address these possibilities. From there we will determine the most feasible design and try to design it to maximize performance. For this project, our budget is $400, although our sponsor is willing to provide more, assuming an adequate business case be presented for additional funding.

Since the first design review, there has been an update to the Gantt Chart. This update includes some more specific logistics of our project. We have determined that we will have our final design chosen by Friday, October 13. After this we can focus on developing the design, both mathematically and through engineering drawings. After this is set, we will start the construction, in hopes to finish that by November 21. After the product is created, we will spend time testing the design and verifying it. This will be done along with the Design Fair and the final report. This timeline is subject to change, but seems to be reasonable to everyone in our group. The updated Gantt chart is located in Appendix A.

The prototype, validation, and final design have all been successfully completed.

12. Discussion

Our system is a novel integration of two powerful methods of HCCI Engine control. It has many improvements off of other systems, yet has some inherent problems associated with it. This section will discuss the benefits and drawbacks with our system.

12.1. Benefits

Our system uses the old concept of EGR in a new way. The benefit of this is that our design is simple in nature and will be very easy to implement, without the need for extensive machining on the original test setup. This system also allows cylinders to be tuned on an individual basis, which was always required, but never available. Tuning each cylinder individually can be very useful in realizing the potential of HCCI.

Additionally, this system does not require an external power source, which reduces the overall loss it incurs on the system.

12.2. Drawbacks

Naturally, no system is perfect. Some drawbacks with this system include that there will be significant thermal losses through the metal piping. We have addressed this problem in our design, by including a thermal barrier coating on the metal, but there is no way to completely eliminate this loss from our system. Similarly, there is a problem with the conduction of heat through the metal of the piping. Although we want to increase the temperature of the intake gas, if heat is conducted through the metal, this could have an undesirable effect on the intake temperature, and is therefore a drawback of our system. We have attempted to limit this effect by including two thermal barrier gaskets to serve as thermal breaks in the system.

Since our system works off of exhaust gas bled from the exhaust manifold, there will be a loss associated with this that can be seen in the turbine of the turbochargers. This will effectively lower the boost and therefore power of the engine. With the very small amounts of exhaust gas that we will be removing from the exhaust system, this loss should be very small, but it is still noteworthy.

Lastly, the overall cost of the system totaling over $1600 will be quite expensive, coupled with the lab time losses associated with the installation and testing period, this system will be expensive to implement. Another cost associated with this system would be the hardware and software needed to control the system. Such a control system was outside the scope of this project and subsequently was not developed by our team. The benefits that this system could have on the development of HCCI are worth all of these expenses.

13. Conclusions and Recommendations

In conclusion, HCCI is an engine technology that is currently being heavily researched due to its benefits in efficiencies and emissions. Before production can take place, the issue of control must be addressed. The EPA is focusing on cylinder-to-cylinder variations which make uniform combustion difficult. We were asked to minimize these variations by designing a control system to monitor and control a selected parameter.

A number of concepts were considered that could potentially control these variations. They main concepts included: variable compression ratio, an intake heating element, EGR, fuel mixtures, and variable valve timing. We then had variations on essentially each one of these main concepts. They were evaluated using Pugh and Morphological charts. Eventually we chose EGR as the concept we wanted to continue with.

The EGR heating system routes exhaust from the exhaust manifold directly into one of the two intake runners per cylinder. Using solenoid actuated control valves, we can then control exhaust gas injection with quick response times and high precision to control the intake temperature of each cylinder independently. Through temperature we have control over stability, power, and fuel consumption. Previously there was no means to control cylinders on an individual basis; therefore if one cylinder had surpassed its threshold of stability then all four cylinders would have to have their power reduced in order to stabilize the one cylinder. As a result, power was not being extracted as efficiently as it could have been compared to a system that could stabilize the cylinder independently of the other three. Our design successfully accomplishes this, allowing for higher power output while maintaining stability.

We were informed that we were not going to be able to test our prototype due to time and financial constraints. As a result, the prototype is not functional however illustrates the final design. Three validation approaches were done to prove the functionality of our design. The results of these validation methods confirm the feasibility of our design.

Computational fluid dynamic simulations were run, using Fluent, to prove that we could accurately control temperature. From Design Review 3, we had determined that a mass flow rate of 0.127 g/s would give us an increase in the in-cylinder temperature of 5°C. With Fluent, these theoretical calculations were validated within 10%.

A regression equation was used to show that the effects of a change in temperature outweighed the effects of a corresponding change in composition in terms of power and fuel consumption. More specifically it was determined that, to achieve a 5% change in BMEP and fuel consumption, which we consider the maximum or critical level per cylinder, an 11°C increase in temperature would be needed corresponding to a injected mass flow rate of 0.280g/s.

Also further simulated combustion analysis was done, using Chemkin, to further investigate the effect of temperature on ignition timing and power. It was determined that temperature has a strong effect on the ignition timing which then has a big impact on output power per cylinder.

We developed an in depth manufacturing plan for the final design which would allow the EPA to easily manufacture our design. This includes in depth engineering drawings, list of parts, and an assembly guide. In addition there is a corresponding bill of materials; we found that the total cost for parts would be $1674.36.

We recommend that the EPA consider our design for implementation and testing. If testing were done and results were not as expected we recommend that the EPA consider other methods to eliminate heat transfer from the EGR system such as an insulation cover. Adapting the system to inject into each intake runner, instead of one per cylinder, could enhance the control of in-cylinder temperature eliminating the need for mixing within the cylinder. Also location of EGR injection could have an effect on the mixing within the intake runner. From Fluent analysis we know that decreasing the diameter of the EGR injection could also further aid the mixing within the intake runner as the injection velocity would be greater. If power losses from the turbochargers were large than expected, the volume of the reservoir in our system could be minimized to decrease the amount of exhaust bled from the exhaust manifold.

14. Acknowledgements

We wish to thank Mike Prince and Andrew Moskalik of the Environmental Protection agency (EPA) for their sponsorship of this project. We also wish to thank Professor Kazuhiro Saitou for leading the automotive section of ME450. We further wish to thank Bob Coury and Marv Cressey of the Mechanical Engineering machine shop for their help in manufacturing the prototype.

Hakan Uras wishes to thank Aris Babajimopoulos, Zoran Fillipi, George Lavoie and Kyoungjoon Chang of the Automotive Research Center for their guidance and assistance in simulating HCCI combustion. He also wishes to thank Doug Yonish of Static Controls Corporation for generously offering to powder-coat the prototype.

Evan Fleming wishes to thank Prof. Rayhaneh Akhavan for her assistance with the computational fluid dynamics.

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16. Biographies

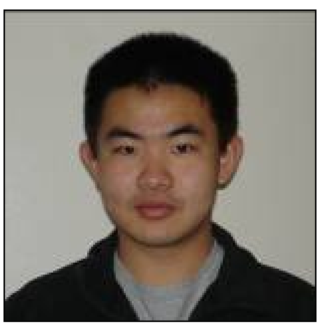
Evan Fleming:

Evan Fleming was born and raised in Ann Arbor and is in his final semester as an undergraduate here at the University of Michigan. As early as middle school, it was his dream to attend the College of Engineering at his hometown university. Influenced by the strong presence of the auto industry in southeastern Michigan, Evan has had an interest in cars since a young age.

While taking classes at U of M he realized a strong interest in the thermal and fluid sciences. In addition he has a strong interest in the environment and means to reduce our impact on it through alternative fuels and technologies.

Future plans hopefully include graduate studies in combustion or alternative energy technologies en route to a career in related fields.

(Kevin) Quan Zhu:



Kevin Zhu is a senior in Mechanical Engineering and minoring in German. Having spent half of his life in Beijing, China, and the other half in the United States, he came to realize the enormous opportunities that the University provides. After majoring in Biomedical Engineering, Chemical Engineering and Premed, he finally found his passion in the ME department. Despite warnings from his mechanical engineer parents not to major in ME, he decided to pursue his passion and love of cars which brought him to this present day.

With experience in medical research, chemical industry, and electronics industry, he hopes someday to use his abilities in this world of interdisciplinary global engineering. He is also fluent in Chinese, English and German. In his free time he plays tennis, snowboards and is a marathon runner. He is also addicted to fishing.

Paul Russell:

Paul Russell is a senior in the University of Michigan College of Engineering. He is from a small town in southwest Connecticut. Engineering has been a passion in his life, and has lead him to the UofM. He is planning on graduating in April 2007 with a Degree in Mechanical Engineering and a Minor in German.

Paul has interned at EX-CELL-O Gmbh. in Eislingen, Germany, and last summer at the DaimlerChrysler headquarters in Auburn Hills, MI. At these internships, he has been exposed to manufacturing, development, Side Airbags, among various other technical areas. On Campus, Paul is the corporate relations Chair for the UofM Chapters of the Society of Automotive Engineers.

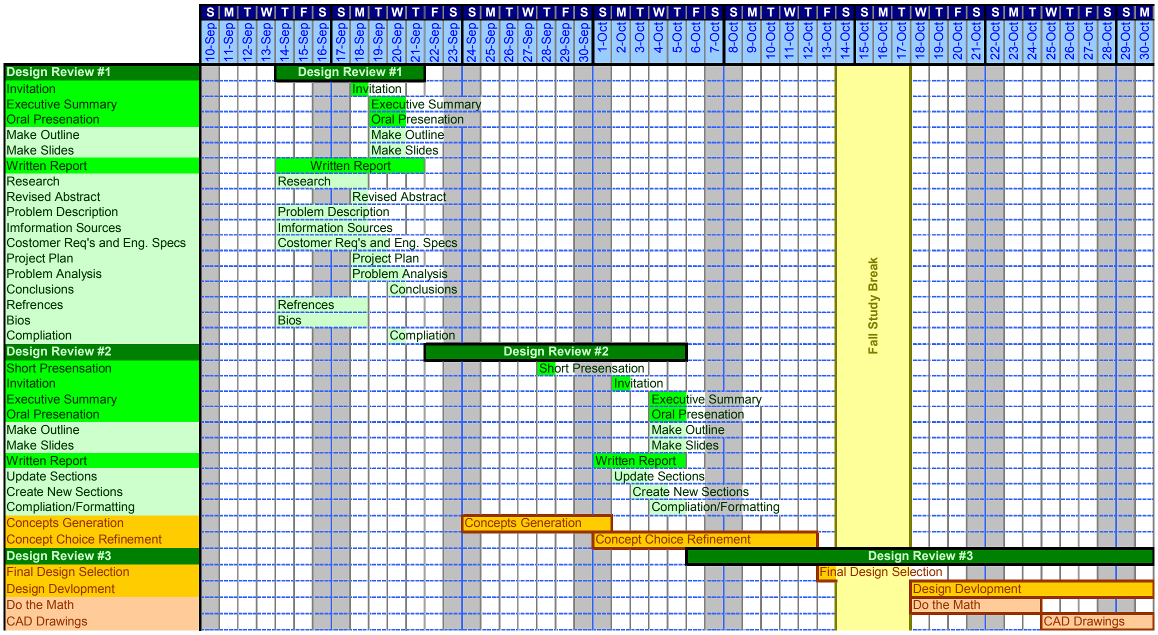
Hakan Uras:

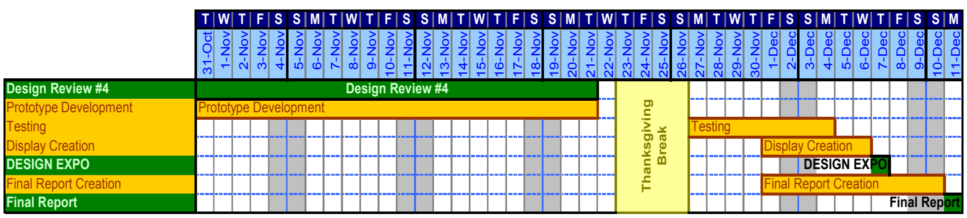
Hakan Uras is a senior in the Department of Mechanical Engineering at the University of Michigan. Born and raised in Ann Arbor, he is a second-generation Michigan engineer. His interest in mechanical engineering stems from a love of automobiles. An interest in math further served to bring him to engineering.

After graduating in December of 2006, he plans to work in the field of automobile design and development. Of particularly interest are advanced internal-combustion engines which combine high output with exceptional efficiency and minimal environmental impact.

In addition to his technical interest in cars, he is a keen competitor in amateur motorsports. He is a member of the SCCA and competes in BMWCCA sanctioned events. Away from the track, he enjoys music and good food.

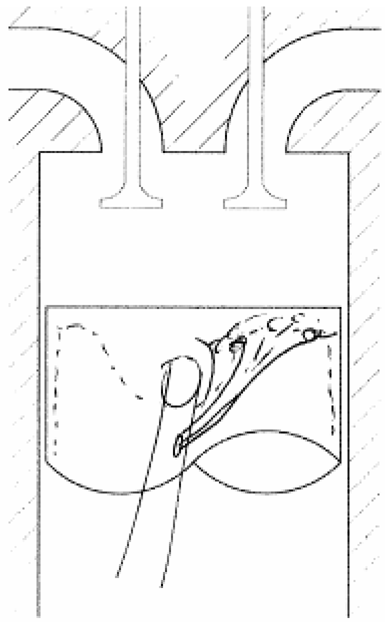
Appendix A: Gantt Chart



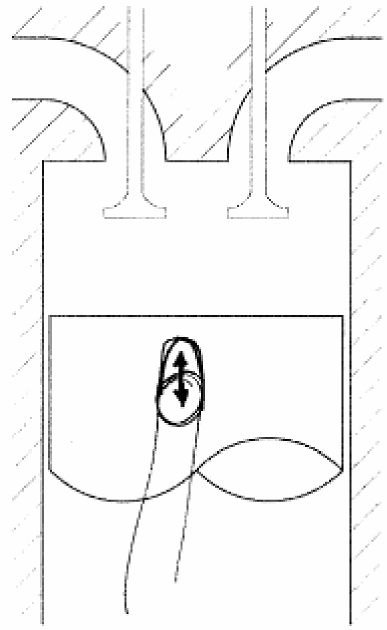


Appendix B: Design Concepts

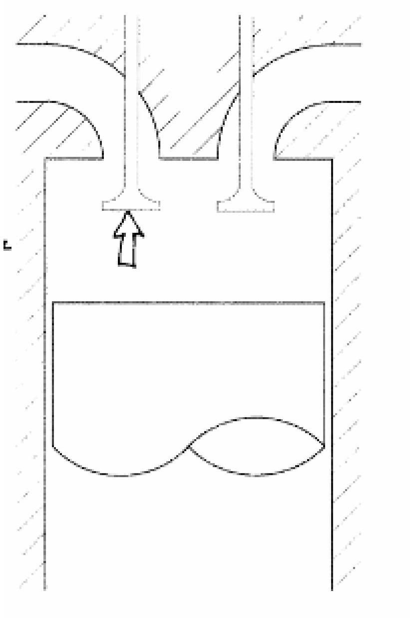
Design 1: Cool Piston Head in a controlled fashion, per cylinder



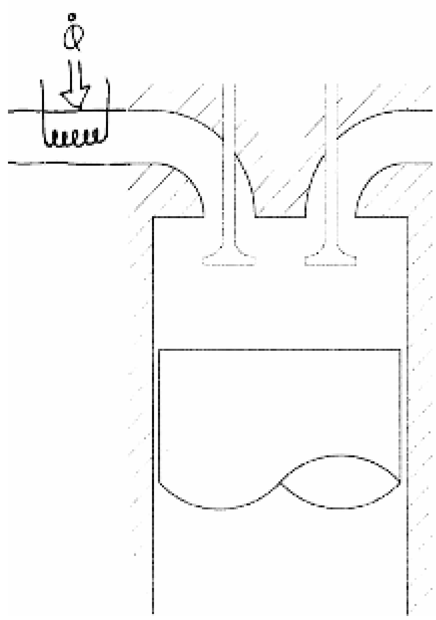
Design 2: Control CR by adjusting cylinder height at knuckle



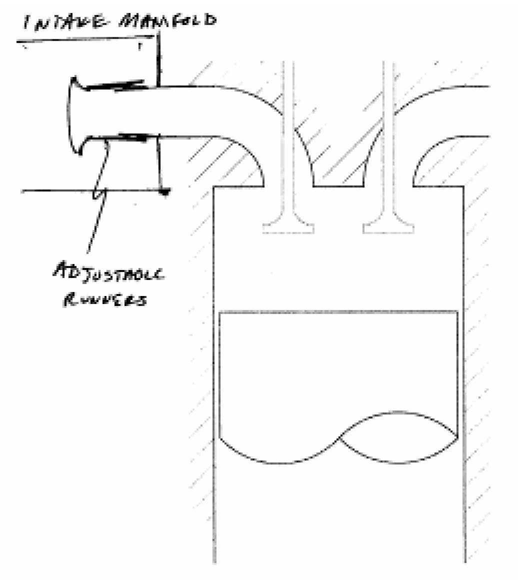
Design 3: Control CR by the use of VVT for Intake Valve



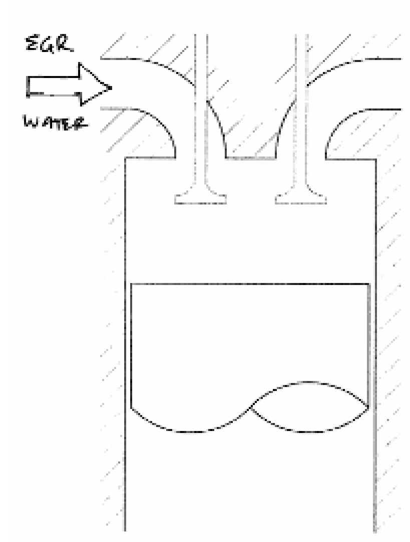
Design 4: Preheat intake air/ fuel mixture



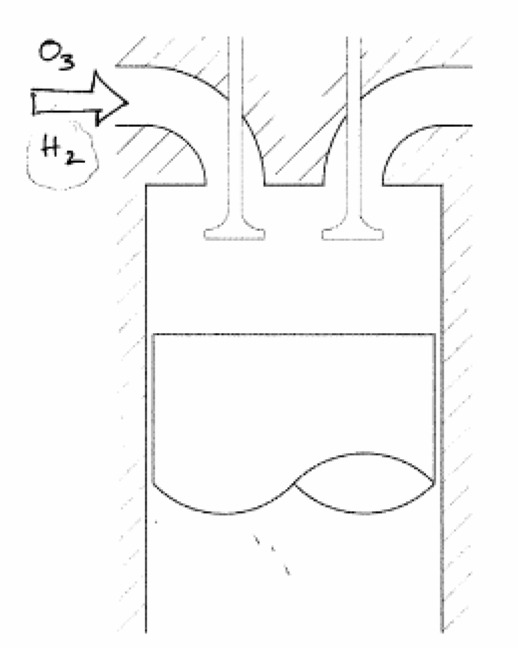
Design 5: Variable runners in intake manifold



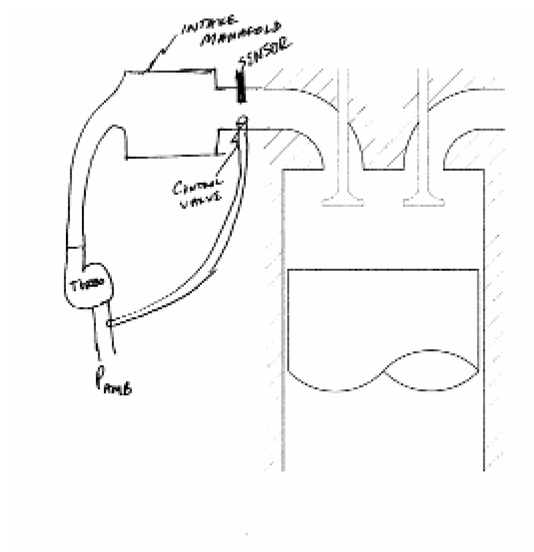
Design 6: Add combustion retardants



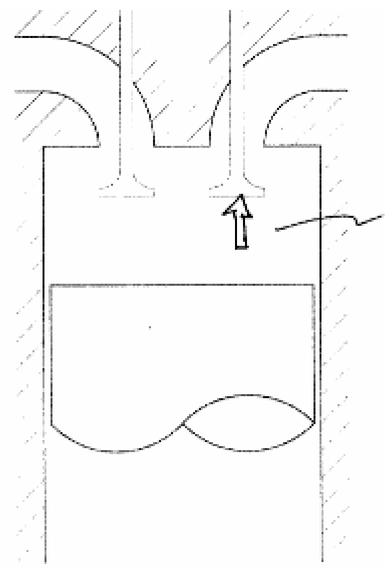
Design 7: Add Combustion Accelerants (O_3, H_2)



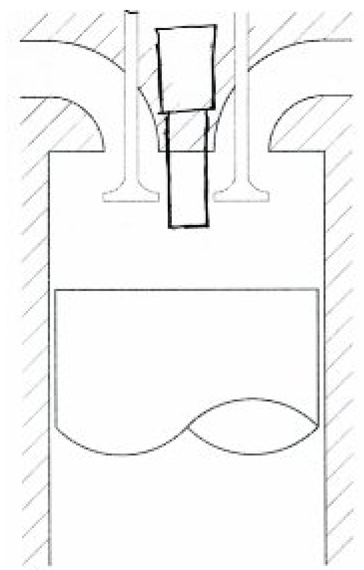
Design 8: Blow-off valve to ambient pressure before turbo



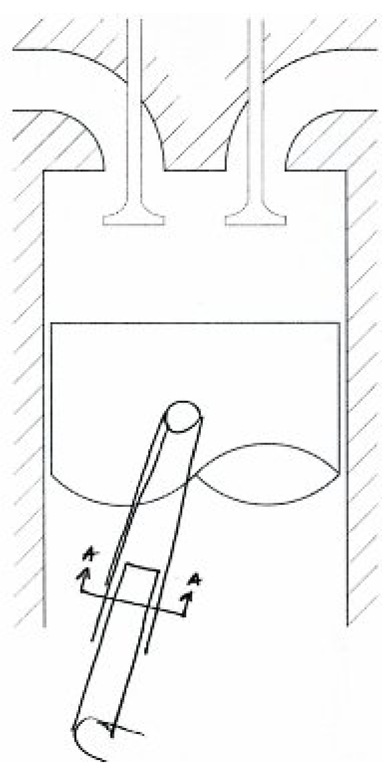
Design 9: Internal EGR control through VVT



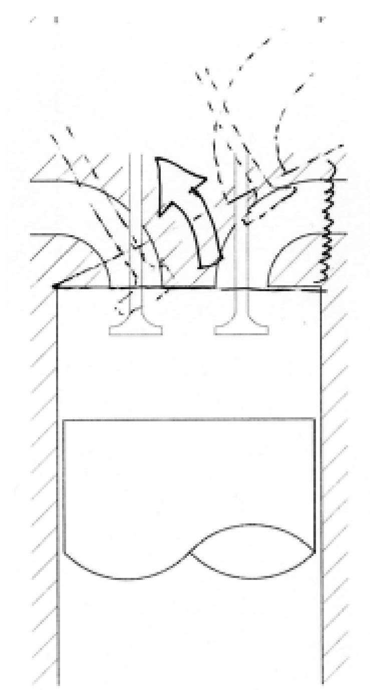
Design 11: Variable compression ratio piston



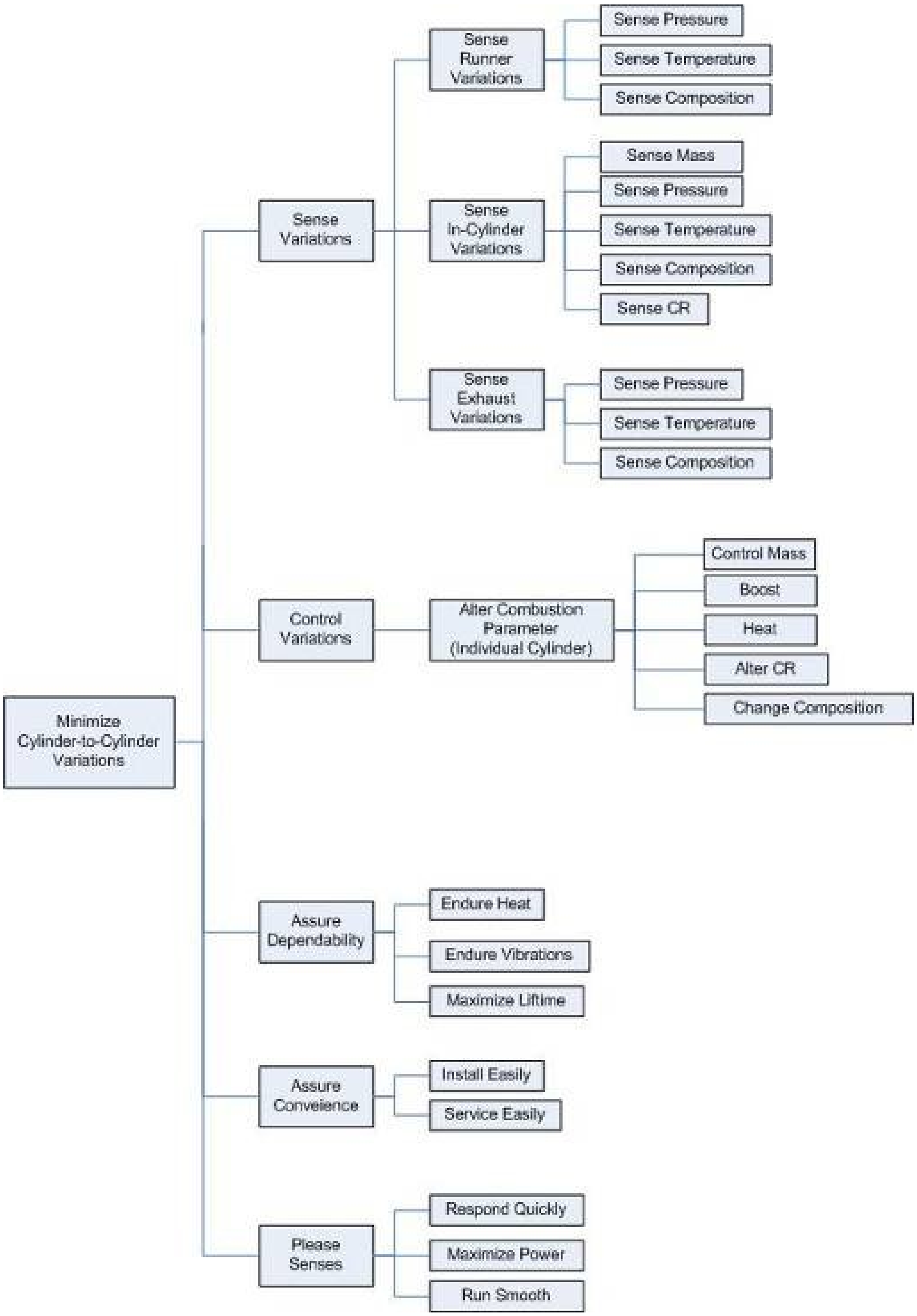
Design 10: Variable length connecting rod



Design 12: Hinged cylinder head to change CR



Appendix C: FAST Diagram



Appendix D: Morphological Charts

Combustion Parameter Control Methods

Concept / Function	1	2	3	4	5	6	7	8	9	10	11	12
Boost			1	1	1			1				
CR		1	1							1	1	1
Intake Temp				1		1			1			
Coolant Temp	1											
Composition						1	1		1			
Intake Mass					1			1				
Totals:	1	1	2	2	2	2	1	2	2	1	1	1

Parameter Variation Sensing Instruments

Concept / Function	Transducer	Thermocouple	Lambda Sensor	Linear Potentiometer	Infrared Sensor
Pressure	1				
Intake Temp		1			1
Cylinder Temp		1			1
Composition			1		
Mass	1				
CR	1			1	
Totals:	3	2	1	1	2

Appendix E: Pugh Chart

	Concept	1	2	3	4	5	6	7	8	9	10	11	12
Customer Requirements	**Weight (1-4)**												
Low Cost	*1*	1	-1	-1	1	-1	-1	-1	-1	1	-1	-1	-1
Improve Efficiency	*3*	0	1	1	1	1	0	1	1	1	1	1	1
Ease of Installation	*1*	0	-1	0	1	0	0	1	0	1	-1	0	-1
Adaptable Over Time	*3*	0	-1	0	0	0	0	0	0	0	-1	0	-1
Light Weight	*2*	1	0	1	0	1	1	-1	1	1	0	1	0
Unobtrusive	*1*	1	-1	1	0	1	1	-1	1	0	-1	0	-1
Programmable	*2*	1	0	1	1	0	1	0	1	0	0	1	0
Responsiveness	*4*	-1	0	0	-1	0	1	1	0	0	0	1	0
Reliable	*4*	-1	0	1	1	1	1	0	1	1	-1	1	0
Easy to Maintain	*2*	0	0	0	1	0	0	1	0	1	0	0	0
Feasibility	*3*	1	-1	1	1	0	1	0	1	1	-1	1	0
	Σ+	5	1	6	7	4	6	4	6	7	1	6	1
	Σ-	2	5	1	1	1	1	3	1	0	6	1	4
	Σ	3	-4	5	6	3	5	1	5	7	-5	5	-3
	Weighted	1	-6	14	12	9	15	6	14	16	-10	17	-3
	Rank	9	11	**4**	6	7	**3**	8	**4**	**2**	12	**1**	10

Appendix F: External EGR vs. Compressed Air Heating Analysis

Using exhaust to heat one cylinder: assuming air is the working fluid, ideal gas behavior, ideal processes, and steady state. Data supplied by the EPA at a working condition of 1600 rpm and 11.3 bar BMEP.

Required mass

m_a: mass of air	T_a: Temperature of air	T_{ae}: Temp. after mixing
m_e: mass of exhaust	T_e: Temp. of exhaust	

$$0.25 m_a T_a + m_e T_e = (0.25 m_a + m_e) T_{ae}$$

$$\frac{m_e}{m_a} = \frac{T_{ae} - T_a}{4(T_e - T_{ae})}$$

For one degree increase: $m_e = 5.9\text{x}10^{-4} m_a$
For five degree increase: $m_e = 0.0030 m_a$

Power loss

W: Work	C_P: Constant-pressure specific heat	T_e: Exit temp.
T_i: Inlet temp.		

$$W = m_e C_P (T_e - T_i)$$

Therefore work and mass are directly proportional so that the percent decrease in work as a result of a decreased percent of gas, is identical magnitude. For worst case scenario (equivalent to a ten degree increase for a single cylinder) the power loss would therefore be 0.6 %.

Using uncooled compressed air to heat one cylinder (same assumptions apply).

m_b: mass of injected air	T_b: Temp. of injected air	T_{ab}: Temp. after mixing

$$0.25(m_a - m_b) T_a + m_b T_b = 0.25(m_a + 3m_b) T_{ab}$$

$$\frac{m_b}{m_a} = \frac{T_a - T_{ab}}{3T_{ab} - 4T_b + T_a}$$

For one degree increase: $m_e = 0.00265 m_a$
For five degree increase: $m_e = 0.0132 m_a$

Using uncooled compressed air requires 4.5 times more injected mass for equivalent change in temperature.

Appendix G: Flow Rate Analysis

We were provided with the values for intake fresh air mass and volume flow rates along with the percentage of oxygen in the mixed air and EGR. It should be noted that the EPA has asked us to not reveal the specific information they have provided us with. We then used this information to determine the percentage of EGR being used in the cycle which then allowed us to determine the mass and volume flow rates for the EGR and for the total intake air flow. To determine the composition of the air it was assumed that fresh air consisted of 21% O_2 and 79% N_2. Then using the provided percentage of O_2 in the fresh air mixture, we determined the percentage of N_2 assuming the same ratio. The remainder was therefore the percentage of EGR mixed with the fresh air.

Once the composition of the air was determined, we then were able to determine the total volume flow rate and the EGR flow rates using the equations below:

$$\dot{V}_T = \frac{\dot{V}_{fa}}{(\%FA)_V}$$

$$\dot{V}_{EGR} = \dot{V}_T (\%EGR)_V$$

where $\dot{V}_T$ is the total volume flow rate, $\dot{V}_{fa}$ is the fresh air flow rate, $\dot{V}_{EGR}$ is the EGR flow rate, and $(\%FA)_V$ and $(\%EGR)_V$ are the percentages of fresh air and EGR on a volume basis respectively.

We then had to determine the corresponding mass flow rates using the ideal gas law. For the exhaust, the specific gas constant was determined by averaging the constants of the exhaust products. A stoichiometric combustion balance was done to determine the composition of the gas. This is shown below with exhaust composition as well (fuel assumed to be isooctane):

$C_8H_{18} + 13.5O_2 + 50.8N_2 \rightarrow 8CO_2 + 9H_2O + 50.8N_2$

CO_2: 11.8%, H_2O: 13.3%, N_2: 74.9%

The averaged gas constant was determined to be 0.285 kJ/kg-K, which is essentially the same as air (0.287 kJ/kg-K). Since intake temperature and pressure were supplied we were able to determine the mass flow rate of EGR, which also allowed us to determine the total mass flow rates. Using our results from Appendix F, we then determined the required mass and volume flow rates that would be needed to achieve a 5°C equivalent increase. This was simply done by multiplying the total mass flow rate by the 0.3% from Appendix F. Results are shown below:

Desired mass flow rate: 0.127 g/s
Desired volume flow rate: 0.126 L/s

Appendix H: Regression Equation Calculations

The EPA supplied us with the following information allowing us to create a sort of regression equation. Also BMEP is directly correlated with fuel consumption so that a 5% change in one is equivalent to a 5% change in the other.

For every 21 degree C change in Intake Temp, with fixed intake O_2, the BMEP changes by 1 bar (15.09 N-m of Torque).

For every 1.77 % Change in Intake O_2, with fixed intake temp, the BMEP changes by 1 bar (15.09N-m of Torque).

This data was taken at an operating point of 1600 RPM and 9.5 bar BMEP; the following conclusions can be made:

Δ 5°C → Δ 2.5% BMEP/Fuel
Δ1.77% O_2 → Δ10.5% BMEP/Fuel

We know from previous calculations that 0.127 g/s or 0.126 L/s are needed for an increase in the in-cylinder temperature by 5°C (also verified with Fluent). Adding this amount of exhaust will have a slight effect on composition. Analysis is shown below:

Intake Composition
EGR: 17.52 L/s
O_2: 4.166 L/s
N_2: 15.67 L/s

Exhaust Composition
0.67 % of exhaust is O_2 (volume basis)

Therefore adding 0.126 L/s will add an additional amount of 0.00085 L/s of O_2. Taking this into account with the rest of the exhaust gas that is not oxygen the overall effect on composition is: a decrease in O_2 by 0.04%. Using the regression equation it can then be found that by adding 0.127 g/s will have the following effects:

Δ 2.5% BMEP/Fuel (due to change in temperature)
Δ 0.24% BMEP/Fuel (due to change in oxygen)

This two effects mitigate each other resulting in a net effect of:

Δ 2.26% BMEP/Fuel

A 5% change in BMEP would require an 11°C change in intake temperature which corresponds to a mass flow rate of 0.280 g/s; this is considered to be the maximum needed flow rate for a given runner.

Appendix I: Combustion Analysis

Gross indicated mean effective pressure (IMEP) is defined as the work done on the piston during the compression and expansion strokes divided by the cylinder displacement. Defining the compression/expansion strokes in terms of crank angle, the equation for gross IMEP becomes:

$$IMEP_{gross} = \left(\frac{1}{V_D}\right)\int_{-180^\circ}^{180^\circ} PdV$$

Because the CHEMKIN models duration is defined by intake valve closing and exhaust valve opening, our calculated gross IMEP is slightly less then the actual gross IMEP:

$$IMEP_{gross} = \left(\frac{1}{V_D}\right)\int_{-156^\circ}^{152^\circ} PdV$$

CHEMKIN models heat transfer by calculating an instantaneous heat transfer coefficient at each point in time. Heat flux into the cylinder wall is then calculated by the following:

$$Q_{wall} = hA(T - T_{wall})$$

The instantaneous heat transfer coefficient is calculated through the Nusselt number and Reynolds number. For both dimensionless parameters, the characteristic length is the cylinder bore:

$$Nu_h = 0.035\,\mathrm{Re}^{0.71} = \frac{hD}{\lambda}$$

The Reynolds number is modified to account for the average gas cylinder speed by the Woschni Correlation:

$$w = \left[2.28 + 0.308\left(\frac{v_{swirl}}{\overline{S}_p}\right)\right]\overline{S}_p + 0.054\frac{V_D T_i}{P_i V_i}\left(P - P_{motored}\right)$$

$$\mathrm{Re} = \frac{Dw\rho}{\mu}$$

Appendix J: System Failure Mode Effects and Analysis

Item Name / Function	Potential Failure Mode	Potential Effects of Failure	SEV	Potential Causes/Mechanisms of Failure	OCC	Current Design Controls	DET	RPN	Recommended Actions	Responsibility and Target Completion Date
1 - Valves										
•Control flowrate of EGR in system •Control pressure of EGR in system	Valve seizes	•Unrestricted Flow of EGR •No Flow of EGR	8	Poor maintenance of valve	3	None	2	48	Periodic inspection of valves	Team 12/1/06
			8	Valve not designed for temperature	3	•Finite element analysis of design •Proper material selection for valve body	2	48	Thermal stress testing of prototype valves	Team 12/1/06
	Erratic pressure control	Incorrect EGR metering	7	Leak in pressure signal line	3	High temperature tubing for signal line	4	84	Periodic inspection of signal line	Team 12/1/06
			7	Poor control system design	2	Simulation of control algorithm	4	56	MATLAB simulation of control algorithm	Team 12/1/06
	Valve leaks when shut	Valve seal failure	6	Seal melting	2	FEA on valve body	5	60	Thermal stress testing of prototype valves	Team 12/1/06
			6	Carbon deposits on seal	3	Engine Calibration	5	90	Emissions testing of exhaust at multiple operating conditions	Team 12/1/06
2 - System Piping										
•Transport EGR from exhaust manifold to intake runner •Prevent temperature drop of EGR •Seal EGR from ambient	EGR Leak	•Increased vehicle emissions •Partial function of auxillary EGR system	10	Weld failure	2	Finite element analysis on piping	4	80	Magnaflux inspection of welds	Team 12/1/06
			10	Gasket failure	2	Proper gasket selection	4	80	Periodic visual inspection of gaskets	Team 12/1/06
			10	Clamp failure due to NVH	2	Vibration isolation of piping	1	20	Use of bellows and flex-pipe on piping	Team 12/1/06
	Unacceptable Temperature Drop in EGR	Reduced effectiveness of auxillary EGR system	6	Poor thermal insulation of piping	2	CFD on EGR flow	1	12	None	Team 12/1/06
			6	Excessive pipe length	2	CAD optimization	1	12	None	Team 12/1/06
3 - EGR Runner										
•Evenly distribute EGR to intake runners •Prevent temperature drop of EGR •Prevent reversion of intake into EGR runner	EGR Leak	•Increased vehicle emissions •Partial function of auxillary EGR system	10	Weld failure	2	Finite element analysis on piping	4	80	None	Team 12/1/06
			10	Gasket failure	2	Proper gasket selection	4	80	None	Team 12/1/06
			10	Failure due to NVH	2	Vibration isolation of piping	1	20	None	Team 12/1/06
	Unacceptable Temperature Drop in EGR	Reduced effectiveness of auxillary EGR system	6	Poor thermal insulation of piping	2	CFD on EGR flow	1	12	None	Team 12/1/06
			6	Excessive pipe length	2	CAD optimization	1	12	None	Team 12/1/06
	Reversion of Intake Charge into EGR Runner	Erratic Engine Operation	7	Poor EGR runner placement	2	CAD optimization	1	14		

Appendix K: Manufacturing Plan

To manufacture this EGR system, there is a need to create three parts, assemble them, and install it onto the engine. This section outlines these three stages in detail, such that this system could be made the same by anyone.

Part Manufacturing:

Plenum: (See Engineering Drawings in Appendix K)

1. Prepare the Plenum Pipe:
   a. Using a band saw[1] at 50 rpm, cut one (1) 12" section of a 2" diameter round cross-sectional stainless steel-304 pipe.
   b. Using a mill[1] at 450 rpm, center drill four (4) holes in the middle of the 2” diameter round pipe at 1.5", 4.5”, 7.5” and 10.5" from the starting end.
   c. Using the same setup, fully drill the four (4) 15/64" holes that were started.
   d. Using the same setup, ream the 15/64" holes to 1/4".
   e. Turn the 2” diameter round pipe 90° clockwise about its axis, and re-clamp the pipe, and re-zero the workpiece.
   f. Using the mill[1] at 450 rpm, center drill one (1) hole in the middle of the d" diameter round pipe 6" from the same edge as was used before.
   g. Using the same setup, fully drill the 15/16" hole that was started.
   h. Using the same setup, ream the 15/16" hole to 1".

2. Prepare the Plenum End Caps:
   a. Using a mill[1] at 450 rpm, cut two (2) 2.5" square pieces of 1/4" thick stainless steel-304.
   b. Using a compass, scratch in a 2" circle onto the surface of the 2.5" square pieces.
   c. Using a band saw[1] at 50 rpm, cut the two 2.5" pieces as close to the circular shape as possible.
   d. Using the lathe[1] at 1000 rpm, mount a piece of 1" circular aluminum bar-stock using a collet and surface the stock using a surfacing tool, to ensure the perpendicularity of its face to the cutting surface.
   e. Put an open-ended clamp piece into the tailstock quill, and clamp the near-circular plates (one at a time) between the tailstock and the bar-stock.
   f. Using a solid carbide cutting tool, start to round the plates removing 0.02" per pass at a speed of 100 rpm. Remove material until the plate is 1.9" in diameter.

[1] CAUTION! Be careful handling parts right after they are removed from machines, they will be very hot especially when cutting stainless steel-304.

3. Prepare the EGR Pipe:
    a. Using a band saw[1] at 50 rpm, cut one (1) 3" section of a 1" diameter round cross-sectional stainless steel-304 pipe.
4. Prepare the EGR Runners:
    a. Using a band saw[1] at 50 rpm, cut four (4) 5" section of a 1/4" diameter round cross-sectional stainless steel-304 pipe.
5. Prepare the Flange:
    a. Using a mill[1] at 450 rpm, cut one (1) 2" square piece of 1/4" thick stainless steel-304.
    b. Using the same setup, center drill holes in the exact center of the plate, and in each corner 0.325" from the top and side edges.
    c. Fully drill the center drill hole in the middle of the plate to 15/16".
    d. Ream this same hole to 1".
    e. Fully drill the four (4) corner center drill holes to 23/64".
    f. Ream these four (4) holes to 3/8".
6. Attach the End Caps to the Plenum Pipe:
    a. Using a clamp, position the round plates on the end of the Plenum Pipe, such that the ends are completely covered.
    b. Using a TIG Welder[2], first spot weld each side, and then make any adjustments to the position as needed, then run a continuous weld to seal the ends. Ensure that no holes, gaps, or defects are present at the seam.
7. Attach the Flange to the EGR Pipe:
    a. Clamp these pieces together such that the end of the pipe is flush with the Flange.
    b. Using a TIG Welder[2], first make a spot weld, and then make any adjustments to the position as needed, then run a continuous weld to seal the Flange to the EGR Pipe. Ensure that no holes, gaps, or defects are present at the seam.
8. Attach the EGR Pipe to the Plenum:
    a. Place the EGR Pipe (now with a Flange) in the 1" hole in the Plenum, and clamp into place, making sure that the Plenum and EGR Pipe are perpendicular to each other.
    b. Using a TIG Welder[2], first make a spot weld, and then make any adjustments to the position as needed, then run a continuous weld to seal the EGR Pipe to the Plenum. Ensure that no holes, gaps, or defects are present at the seam.

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[*] CAUTION! Be careful handling parts right after they are removed from machines, they will be very hot especially when cutting stainless steel-304.

[2] WARNING! TIG Welding will raise the temperature of the metal at the spot of the weld to its melting point. Allow for plenty of cooling time before handling!

9. Attach the Four (4) EGR Runners to the Plenum:
    a. Place one EGR Runner into each of the four (4) 1/4" holes in the Plenum, and clamp into place, making sure that the Plenum and EGR Runners are perpendicular to each other.
    b. Using a TIG Welder[2], first make a spot weld, and then make any adjustments to the position as needed, then run a continuous weld to seal the EGR Runners to the Plenum. Ensure that no holes, gaps, or defects are present at the seam.

<u>Connecting Pipe: (See Engineering Drawings in Appendix K)</u>

1. Prepare the EGR Pipe:
    a. Using a band saw[1] at 50 rpm, cut one (1) 9" section of a 1" diameter round cross-sectional stainless steel-304 pipe.
2. Prepare the Flanges:
    a. Using a mill[1] at 450 rpm, cut one (1) 2" square piece of 1/4" thick stainless steel-304.
    b. Using the same setup, center drill holes in the exact center of the plate, and in each corner 0.325" from the top and side edges.
    c. Fully drill the center drill hole in the middle of the plate to 15/16".
    d. Ream this same hole to 1".
    e. Fully drill the four (4) corner center drill holes to 23/64".
    f. Ream these four (4) holes to 3/8".
    g. Repeat to make second Flange.[3]
3. Connect the Flanges to the EGR Pipe:
    a. Clamp these pieces together such that the each end of the pipe is flush with a Flange, making sure that the Flanges are perpendicular to the length of the pipe.
    b. Using a TIG Welder[2], first make a spot weld, and then make any adjustments to the position as needed, then run a continuous weld to seal the Flange to the EGR Pipe. Ensure that no holes, gaps, or defects are present at the seam.

<u>Elbow: (See Engineering Drawings in Appendix K)</u>

1. Prepare Elbow Link:
    a. Using a band saw at 50 rpm, cut a 90° pipe bend such that it is 3" long from the centerline of the opposite leg to the end of the length for both legs.

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[*] CAUTION! Be careful handling parts right after they are removed from machines, they will be very hot especially when cutting stainless steel-304.

[2] WARNING! TIG Welding will raise the temperature of the metal at the spot of the weld to its melting point. Allow for plenty of cooling time before handling!

2. Prepare Flange:
    a. Using a mill at 450 rpm, cut one (1) 2" square piece of 1/4" thick stainless steel-304.
    b. Using the same setup, center drill holes in the exact center of the plate, and in each corner 0.325" from the top and side edges.
    c. Fully drill the center drill hole in the middle of the plate to 15/16".
    d. Ream this same hole to 1".
    e. Fully drill the four (4) corner center drill holes to 23/64".
    f. Ream these four (4) holes to 3/8".
3. Connect Flange to Elbow Link:
    a. Clamp these pieces together such that the end of the pipe is flush with a Flange, making sure that the Flange is perpendicular to the leg of the pipe it is attached to, and parallel to the leg that it is not attached to.
    b. Using a TIG Welder[2], first make a spot weld, and then make any adjustments to the position as needed, then run a continuous weld to seal the Flange to the Elbow Link. Ensure that no holes, gaps, or defects are present at the seam.

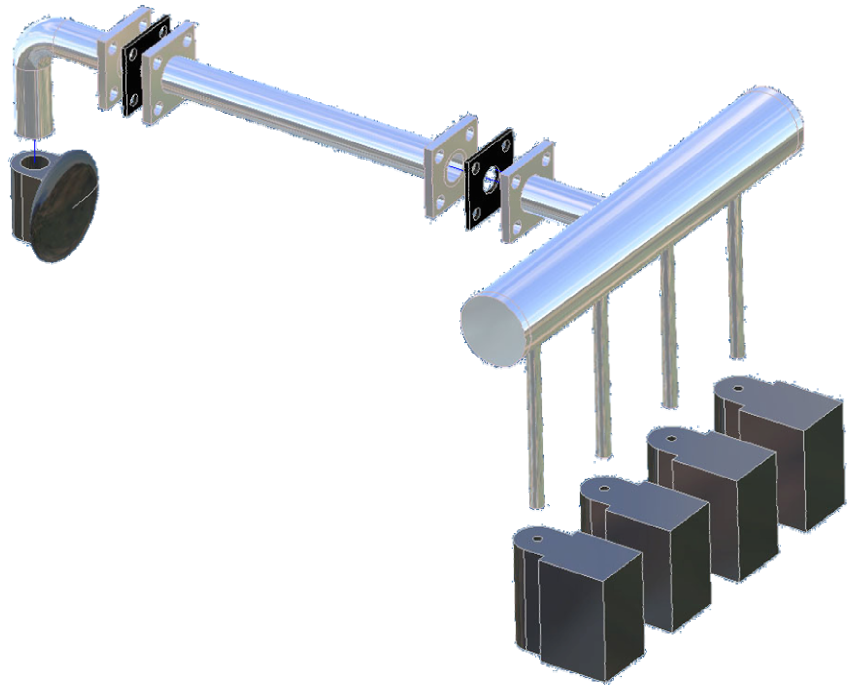
<u>Part Finishing:</u>

1. Polishing:
    a. To prepare the parts for the coating, clean everything with acetone[4], and polish with 200-grit sandpaper.
2. Coating:
    a. These cleaned parts will then be sent away to be coated in Cermakrome® ceramic-metallic thermal barrier coating.

[4]CAUTION! Only use acetone in a well-ventilated or open area. The inhalation of acetone can be dangerous to your heath!

[*] CAUTION! Be careful handling parts right after they are removed from machines, they will be very hot especially when cutting stainless steel-304.

[2] WARNING! TIG Welding will raise the temperature of the metal at the spot of the weld to its melting point. Allow for plenty of cooling time before handling!



Assembly of Parts (see exploded view above for visualization):

<u>EGR Valve Installation:</u>

1. Screw in a press-fit fixture into both sides of the EGR Valve.
2. Insert the Elbow Link onto the downstream fitting of the EGR Valve (Note: this should be difficult to warp the stainless steel-304 so use appropriate wrenches).
3. Using a power drill* cut a 1/4" hole into the Plenum and install the vacuum line.

<u>Solenoid Controlled Ball Valve Installation (x4):</u>

1. Screw in a press-fit fixture into both sides of the Solenoid Controlled Ball (SCB) Valve.
2. Insert the 1/4" pipe from the Plenum onto the upstream fitting of the SCB Valve (Note: this should be difficult to warp the stainless steel-304 so use appropriate wrenches).
3. Insert the Runner Extensions into the downstream fitting of the SCB Valve.

<u>Attach the Elbow Link to the Connecting Pipe:</u>

1. Cut out a piece of WLT-EGS-02 Exhaust Gasket with a laser cutter. This should be cut out with the footprint of the Flanges.

2. Align the two flanges with the Gasket and screw together using four (4) 1" SAE Grade-8 3/8" Hex Bolts, four (4) SAE Grade-8 3/8" washers, and four (4) Stainless Steel 18-8 3/8-16 Flanged Nuts.
3. Tighten in a star formation.

<u>Attach the Plenum to the Assembly:</u>

1. Cut out a piece of WLT-EGS-02 Exhaust Gasket with a laser cutter. This should be cut out with the footprint of the Flanges.
2. Align the two flanges with the Gasket and screw together using four (4) 1" SAE Grade-8 3/8" Hex Bolts, four (4) SAE Grade-8 3/8" washers, and four (4) Stainless Steel 18-8 3/8-16 Flanged Nuts.
3. Tighten in a star formation.

Installation of the EGR System:

<u>Installation of the System to the Exhaust Manifold:</u>

1. Preparation of Exhaust Manifold:
    a. Make sure that the Engine is not running and that the Exhaust Manifold is cool.
    b. Lay the EGR system onto the Exhaust Manifold and make a mark where the Hole should be made to tap into the Manifold. Make sure that this is at the point right when the Exhaust Manifold starts to funnel into the rest of the Exhaust System.
    c. Using a Power Drill make a 1" hole in the Exhaust Manifold.
    d. Fit a Fixture into this hole and use a TIG Welder to hold it in place. First with a spot weld and then with a continuous weld. Ensure that no holes, gaps, or defects are present at the seam.
2. Installation of EGR System:
    a. Fit the System into the Fitting and press fit the connection. (Note: this should be difficult to warp the stainless steel-304 so use appropriate wrenches).
    b. Make sure it is snug, and does not move in its connections.

Installation of the System to the Intake Runners:

1. Preparation of Intake Runners:
    a. Make sure that the Engine is not running and that the Runners are cool.
    b. Make a mark where the Holes should be made to tap into the Manifold. Make sure that this is at the point right after the bend in the Runner as it runs into the Head.
    c. Using a Power Drill make a 1/4" hole in the Intake Runners.
    d. Fit a Fixture into this hole and use a TIG Welder to hold it in place. First with a spot weld and then with a continuous weld. Ensure that no holes, gaps, or defects are present at the seam.
2. Installation/Completion of the EGR System:
    a. Fit the EGR Runners into the Fittings and press fit the connections. (Note: this should be difficult to warp the stainless steel-304 so use appropriate wrenches).
    b. Make sure it is snug, and does not move in its connections.

Appendix L: Engineering Changes Notice

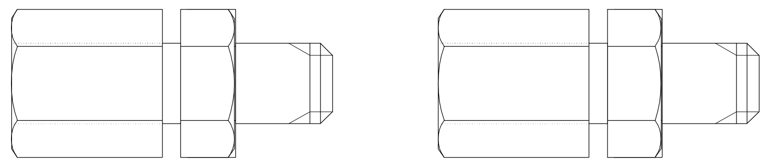
The following changes have been made since Design Review 3.

WAS

Stainless Steel ¼ ” compression fitting

IS

Brass 3/8 ” compression fitting



Notes: Replaced stainless steel fitting for brass fitting for conceptual prototype because cheaper, but would not hold up under operating conditions. Also changed diameter to 3/8 to fit onto existing pipes

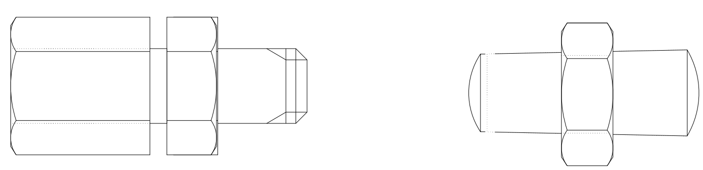
Project: HCCI Engine Control	
Ref Drawing: Brass Fitting	
Engineer: K. Zhu	12/7/06
Mgmt./Sponsor: EPA	12/7/06

WAS

Stainless Steel 1” compression fitting

IS

PVC 1” tapered fitting

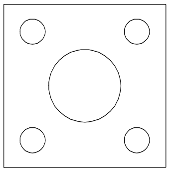


Notes: Replaced stainless steel fitting for PVC fitting for conceptual prototype because cheaper, but would not hold up under operating conditions. Also chose tapered fitting to fit onto existing pipes

Project: HCCI Engine Control	
Ref Drawing: PVC Fitting	
Engineer: K. Zhu	12/7/06
Mgmt./Sponsor: EPA	12/7/06

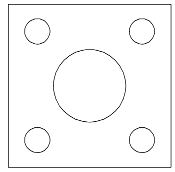
WAS

High Temperature Exhaust Gasket



IS

Cork Gasket



Notes: Replaced high temperature exhaust gasket for cork gasket for conceptual prototype because cheaper, but would not hold up under operating conditions.

Project: HCCI Engine Control	
Ref Drawing: Exhaust Gasket	
Engineer: K. Zhu	12/7/06
Mgmt./Sponsor: EPA	12/7/06

WAS

EGR Valve

IS

Gas Pressure Regulator Valve

Notes: Replaced EGR Valve for Gas pressure regulator valve in conceptual prototype because cheaper, but would not hold up under operating conditions.

Project: HCCI Engine Control	
Ref Drawing: EGR Valve	
Engineer: K. Zhu	12/7/06
Mgmt./Sponsor: EPA	12/7/06

WAS

Burkert 255 2/2-Way High Temp Solenoid Valve

IS

Solenoid Valve,1/4 In

Notes: Replaced High Temp Solenoid Valve for lower rating solenoid valve in conceptual prototype because cheaper, but would not hold up under operating conditions.

Project: HCCI Engine Control	
Ref Drawing: Solenoid Valve	
Engineer: K. Zhu	12/7/06
Mgmt./Sponsor: EPA	12/7/06

WAS

Ceramic-Metallic Thermal Barrier Coating

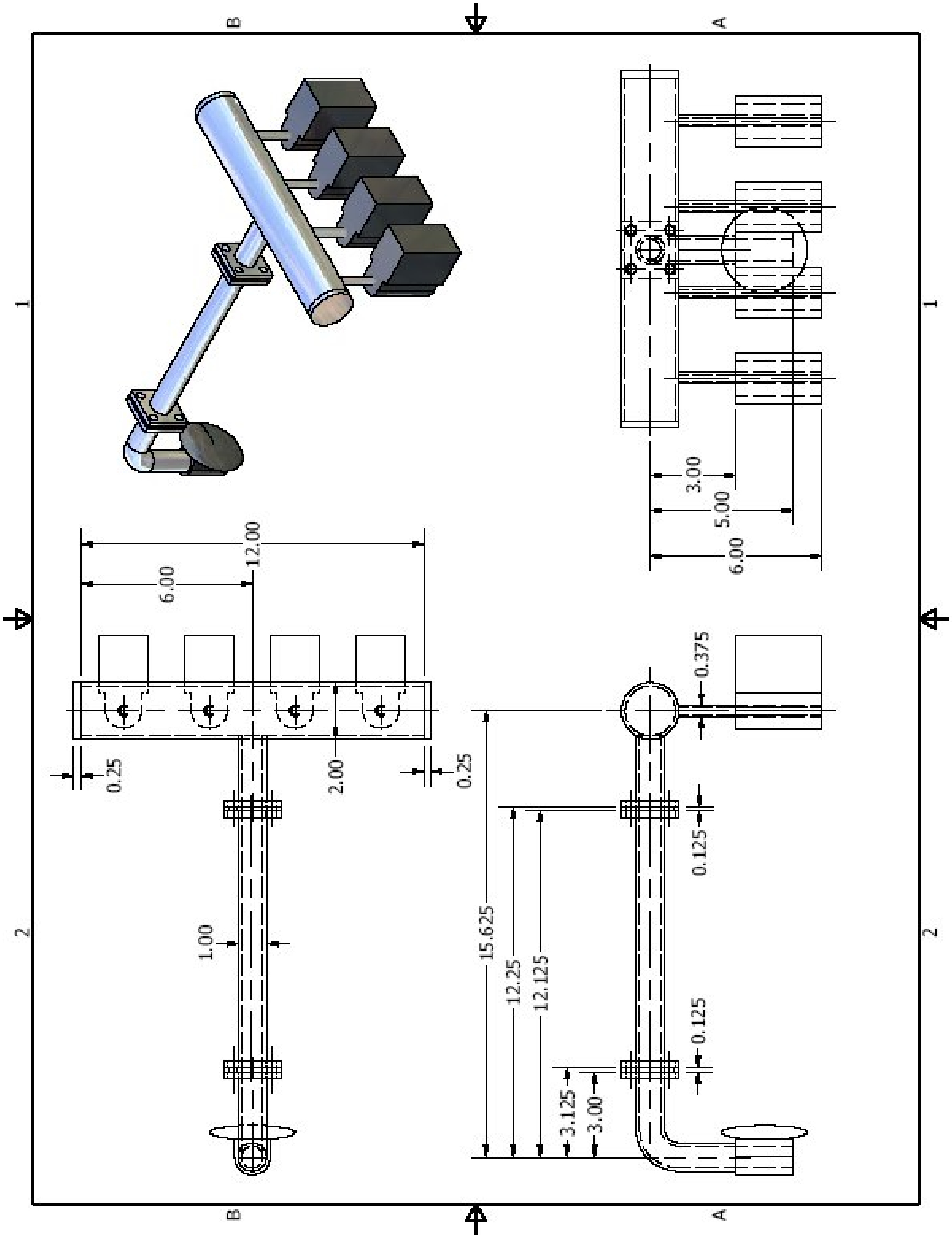
IS

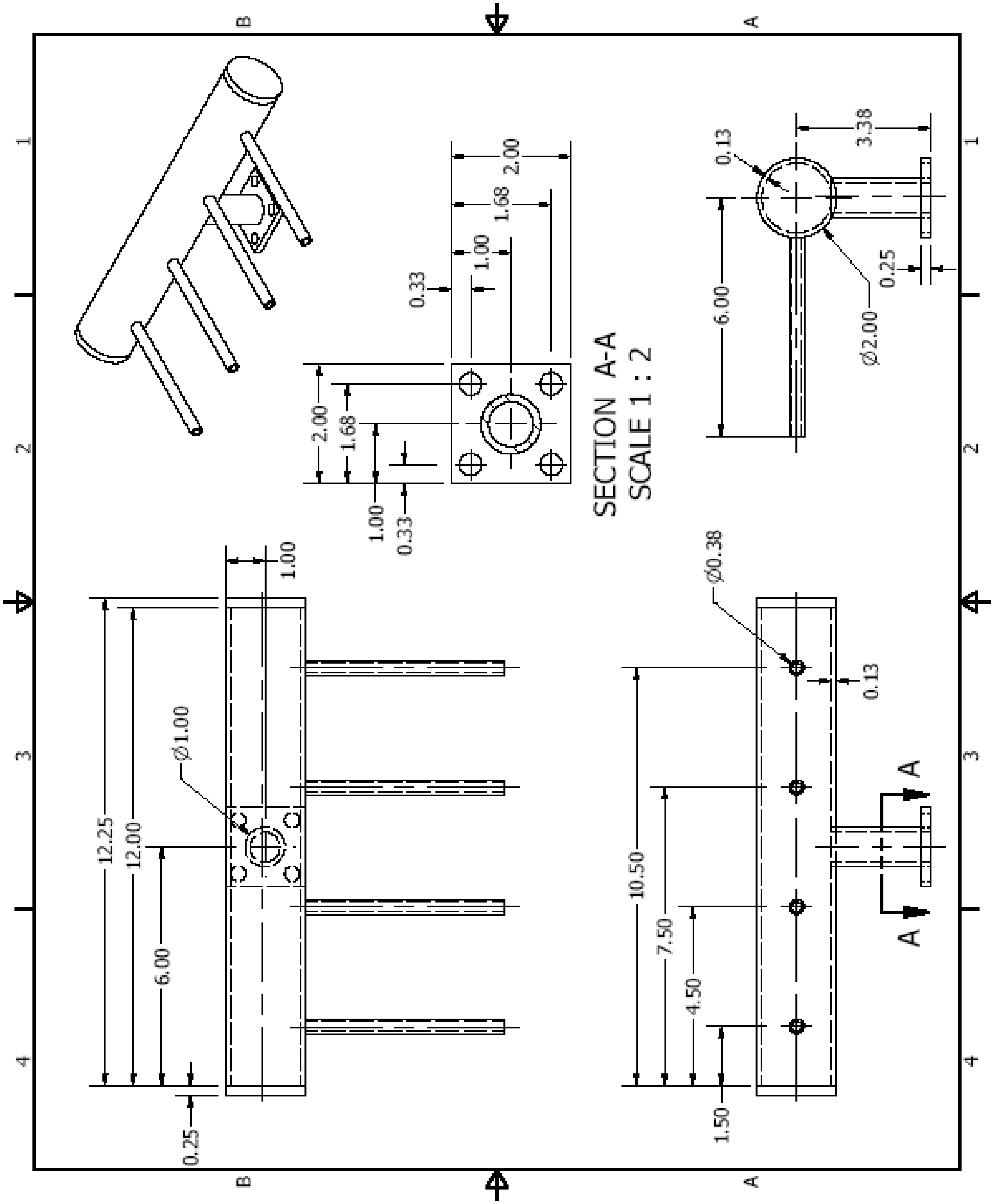
Aluminum Powder Coating

Notes: Replaced Ceramic-Metallic Thermal Barrier Coating for aluminum powder coating in conceptual prototype because cheaper, but would not be ideal for thermal isolation purposes.

Project: HCCI Engine Control	
Ref Drawing: Ceramic-Metallic Thermal Coating	
Engineer: K. Zhu	12/7/06
Mgmt./Sponsor: EPA	12/7/06

Appendix M: Engineering Drawings

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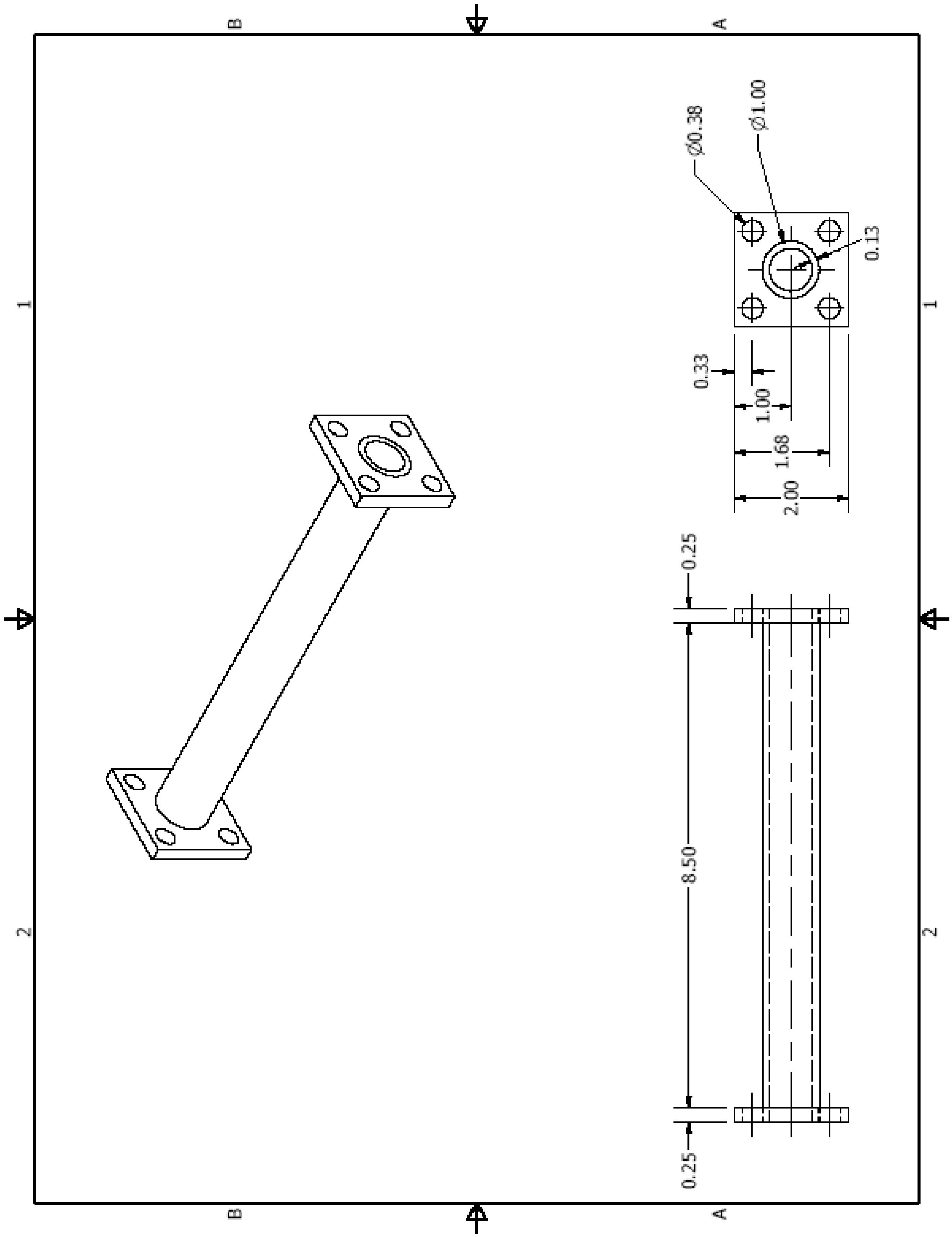

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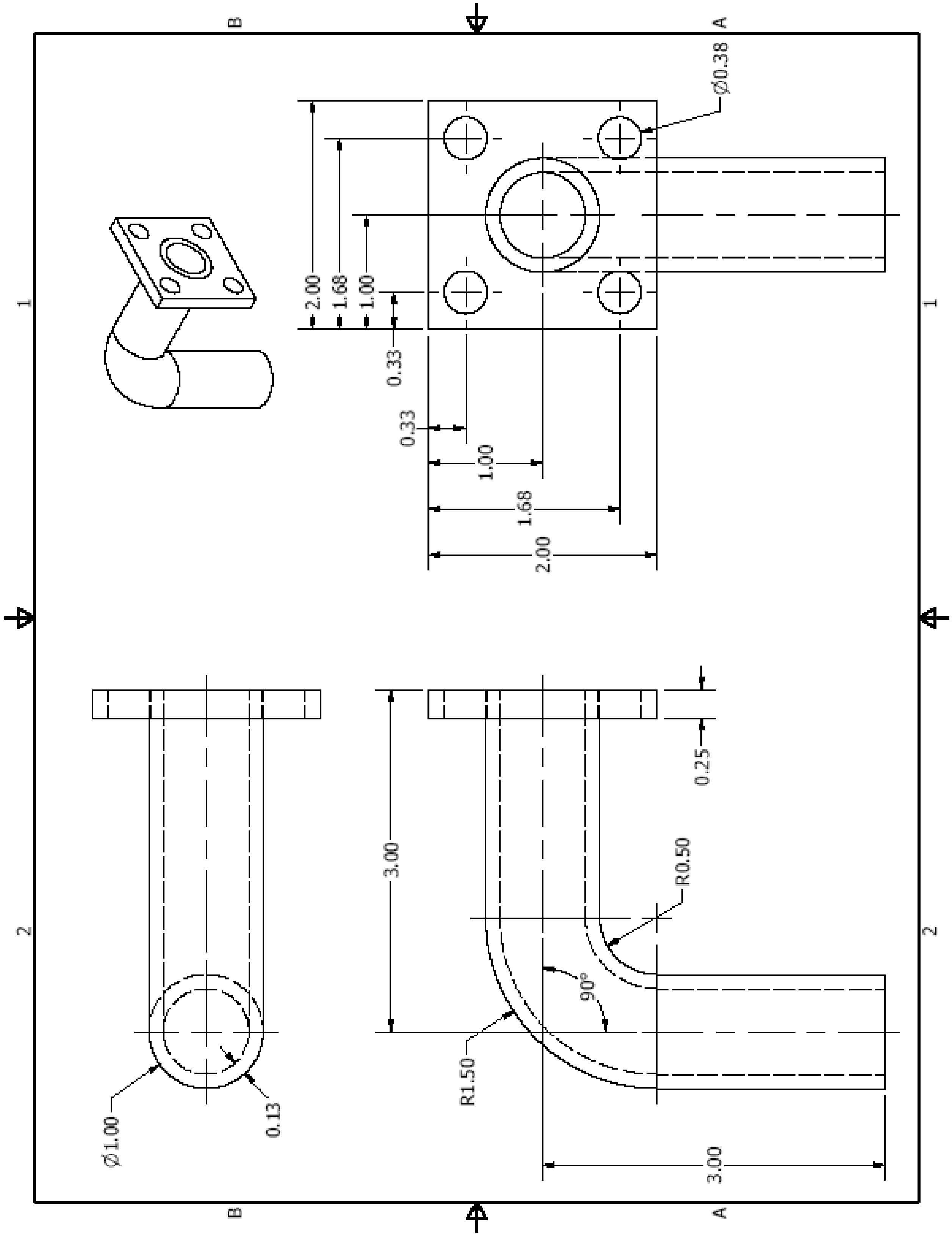
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