1. EXECUTIVE SUMMARY

Our Project Sponsor, Professor Dennis Assanis of the University of Michigan, has required the design and construction of a hydraulic dynamometer for use with a Ricardo Hydra single cylinder research engine. This test engine will be used to research the benefits of variable valve timing (VVT) and/or homogeneous charge compression ignition (HCCI) technologies. The design and sizing of each engine peripheral system is also part of this project. The main goal of this project is to design the dynamometer and the peripheral systems to be modular so that they may be 1) transported with ease, if need be; and 2) easily adapted to any future changes to the engine test cell.

The output specifications of the engine are not standardized and, therefore, the values have been acquired from benchmarking other single cylinder research engines in the Walter E. Lay Automotive Laboratory (AutoLab) at the University of Michigan. The expected maximum torque is 35 Nm; expected maximum engine speed of 4000 RPM; and expected maximum power output of 4 kW. The hydraulic components will have to be sized accordingly to supply a resistance to this load of the engine. In addition, the mounting brackets to attach the dynamometer to the engine will be designed and manufactured to handle the reacting load of the hydraulic motor used to apply the load to the engine.

Manufacturing of the mechanical systems of the dynamometer is one of our deliverables. This includes the fabrication of a rigid frame for the dynamometer, procurement of all required parts, and final assembly of purchased components. The dynamometer frame is complete and all components have been ordered; however, fastening of the components to the frame has not been completed. This will occur once all dynamometer parts have arrived. Arrival of all components and final assembly of the dynamometer will occur before the end of the term.

Design of the peripheral systems for the test cell is the other deliverable of our project. These designs include component selection and sizing for the following peripheral systems: intake, exhaust, oil, coolant and fuel. The sizing and selection of these components has been completed. In addition to the component selection, the budget required to purchase and assemble these systems will be included. This information will be summarized in a design manual; this design manual will expedite the assembly and installation of these systems next term.

2. INTRODUCTION

A new test cell is being designed for the AutoLab at the University of Michigan under the supervision of Professor Dennis Assanis. This test cell will house a single cylinder Ricardo Hydra research engine which will be used to research variable valve timing (VVT) and homogeneous charge compression ignition (HCCI) technologies. We were required to propose designs for the test cell layout to optimize test bed space. We were also responsible for construction of a hydraulic dynamometer and the design of the peripheral systems for the engine test cell.

Direct current (DC) dynamometers are the traditional technology used to measure engine output. Due to the near continuous power output of a multi-cylinder engine, a DC dynamometer can measure the engine power output continuously. However, due to the discrete power output of a single cylinder engine (one power stroke per two revolutions), a hydraulic dynamometer is the preferred design. The number of power strokes per revolution of a single cylinder engine is lower than that of a multi-cylinder engine. The lower frequency can lead to resonance failure of the driveshaft used in a DC dynamometer. A hydraulic dynamometer does not require a driveshaft because a hydraulic motor is coupled directly to the crankshaft, thus, alleviating this failure mode.

A hydraulic dynamometer uses pressurized fluid to apply a torque to the crankshaft. The hydraulic system first uses an electric motor to drive a hydraulic pump to pressurize the fluid. The pressurized fluid is then routed to a hydraulic motor. The fluid is depressurized through the internal mechanism (pistons, gears, or vanes) of the motor. This depressurization causes the drive axle of the motor to spin which, in turn, drives the engine. The load that can be applied to the engine is dependent on the mass flow rate and the pressure of the hydraulic system. The exact geometry and size of the hydraulic pump and motor are major design aspects of the dynamometer.

The deliverables of this project will be the mechanical system of the hydraulic dynamometer. The frame of the dynamometer provides a base to securely fasten all system components so that they do not shake during testing, reducing the chance they will break. Current designs of hydraulic dynamometers utilize a chilled water supply to cool the hydraulic oil. Because there is not a chilled water source in the test cell, we will design an air-oil heat exchanger to cool the oil. In addition to the design and construction of the hydraulic dynamometer, our group also designed the peripheral systems for the engine (oil, cooling, fuel, intake, and exhaust). The assembly of each peripheral system is not within the scope of this project.

3. ENGINEERING SPECIFICATIONS AND FUNCTIONAL DECOMPOSITION

3.1 Hydraulic Dynamometer

There are three single cylinder engines in the AutoLab that are used for other research projects. The capacities of these were used to generate the design targets for the hydraulic dynamometer. The expected capacities are 35 Nm of torque, 4 kW, and 4000 RPM. Conversations with researchers here at the University of Michigan have indicated that the hydraulic system should be run at or below 21 MPa to provide the required load. Another benefit to the hydraulic dynamometers is that they do not require the system to be on a dampened test bed because they do not have the same vibration characteristics of regular AC or DC dynamometers. This allows for more space for research engines on the test bed.

In addition to these specifications, a hydraulic accumulator will be required to reduce hydraulic shock induced into the system. Every hydraulic system requires an accumulator to dampen out pulsations in the flow; the design options are either an inline type or a tank type accumulator. Another design facet of the system is the type of hydraulic pump/motor to be used, either a piston, vane, or gear type.

Another component of the dynamometer we will design is the supporting brackets which mount the hydraulic motor to the crankshaft. These brackets will have to support the maximum torque from the engine as well as the weight of the hydraulic motor. These brackets will be mated to the bolt pattern geometry of the engine shown in Appendix 1.

The production of the dynamometer frame is part of the manufacturing component of this project. The frame is a rigid structure to which the hydraulic components will be fastened. The frame should also act as a drip pan for the hydraulic system. This will catch any minor leaks; it is not intended to contain a high volume spray. The frame will be required to fit though a standard 80 cm door frame and to maneuver around the test cell with ease. The floor plan of the test cell can be seen in Appendix 2. A customer requirement of the dynamometer is the ability to be transported without heavy machinery. Therefore, it will have to have a manageable weight.

A hydraulic dynamometer is a total loss energy system. This means that all energy that is used to pressurize the hydraulic fluid is eventually dissipated as heat. The cooling of the hydraulic fluid is an important design aspect of the dynamometer. If excess heat accumulates, hoses, fittings and valves could fail. Product research has shown that a 19 kW motor is required to pressurize the hydraulic fluid to 21 MPa. In addition to the expected 4 kW output of the engine, the coolant system should have a maximum capacity of 23 kW.

3.2 Intake System

The intake system draws air from ambient conditions and delivers it to the engine at the required temperature. The engineering details of this peripheral system are calculated from the mass of air inducted into the cylinder. The system will measure the mass flow rate of the air. The mass of inducted air is calculated from the engine bore diameter (96 mm), piston stroke (92 mm), maximum attainable speed (4000 RPM) and density of air. Assuming an idealized 100% volumetric efficiency, the intake system should have a maximum capacity of 1175 LPM. The system is also required to dampen pulsations of the air flow induced by the intake stroke. The intake system is also required to operate without creating a large pressure drop before the engine. A design concept schematic of the intake system can be seen in Appendix 3.

3.3 Oil System

The oil system will support the function of lubricating, cooling, cleaning and protecting the engine during operation. The oil system will require a heating element to warm up oil during engine start up. During start up, the crankshaft is spun without initiating combustion; this mode is referred to as motoring. The oil system will also require a cooling element to cool the oil during normal operation. During normal operation, the crankshaft is spinning and combustion is occurring in the cylinder; this mode is referred to as firing. This thermal management system must maintain an operating temperature between 340-400 K. This temperature range maintains a relatively constant oil viscosity [2]. Benchmarking studies of similar oil systems in the AutoLab have indicated that the required oil pressure to maintain proper lubrication is 410 kPa. A design concept schematic of the oil system can be seen in Appendix 4.

3.4 Coolant System
The coolant system's primary function is to maintain the engine's operating temperature during testing. The coolant system will require a heating element to warm the engine during motoring, as well as a cooling element to cool the engine during firing. This is required for a single cylinder engine because combustion alone will not warm up the engine in a reasonable amount of time (15 min). To achieve control of this system, a 3-way control valve will be necessary to alternate between cooling and heating. An automatic shut off valve will also be required to prevent overheating and failure. A design concept schematic of the coolant system can be seen in Appendix 5.

3.5 Exhaust System
The exhaust system transports the reactants of combustion out of the engine. As mentioned previously, the maximum intake flow is approximately 1175 LPM. Evaluating the ideal gas law for a constant pressure and average temperature difference between the intake and exhaust flows (500 K), we calculated the exhaust volume flow rate to be 0.06 m^3/s. A surge tank will be used to dampen pulsations of the air flow produced by the engine exhaust. The system must also have extra space allotted in the ducting for future research components to be added to the exhaust system. The exhaust system needs to be able to operate at temperatures of approximately 800 K [2]. A design concept schematic of the exhaust system can be seen in Appendix 6.

3.6 Fuel System
The customer required the fuel delivery system be designed to utilize direct fuel injection. This requires a high pressure fuel pump in addition to the standard boost pump. In addition to these pumps, the system will be required to filter the fuel, measure the mass flow rate of the flow, and separate any vapor from the liquid flow. Once the fuel is pressurized, a pressure regulator is used to maintain the required pressure for fuel injection. Any fuel that is regulated back to a lower pressure will be required to be cooled, due to the conversion of the energy in the form of pressure to heat. This will require a device to cool the heated fuel back to the supply temperature before the high pressure pump. Benchmarking of similar sized fuel systems in the AutoLab have indicated that the fuel system should be able to provide 0.5 grams per second (g/s) of fuel. A design concept schematic of the fuel system can be seen in Appendix 7.

3.7 Test Cell Layout
The useable floor space inside the test cell is 7.3 m by 4.6 m. The engines in the test cell must be placed on the test bed (1.5 m by 3.7 m). Since test cell space is at a premium, the test bed layout should be optimized as to allow enough space to work on any engine fastened to the test bed. All peripheral systems will be designed such that they may be contained in a cabinet not fastened to

the test bed and simply connected via hoses to the engine. This will allow maximum space utilization of the test cell.

4. CONCEPT GENERATION AND SELECTION PROCESS

4.1 Hydraulic Dynamometer Components

The hydraulic logic of the dynamometer is shown schematically in Appendix 8. The main design aspects of the dynamometer are the hydraulic motor, dynamometer frame, and accumulator. These items are discussed in detail below. The hydraulic logic was designed by Electro-Mechanical Associates (EMA); however, the component procurement is our responsibility.

4.1.1 Hydraulic Motor Options

There exist three types of hydraulic pumps; they are piston, vane, or gear. Piston type pumps are used for heavy duty applications and can typically handle pressures up to 35 MPa. Vane type pumps use an impeller to pressurize the hydraulic fluid. Gear type pumps use two mating gears to pressurize the fluid; however, only one gear is powered. Both of these types of pumps can handle moderate pressures (7 MPa to 20 MPa). Market research has indicated gear type pumps are more durable and can better handle changes in load. Therefore, a gear type pump will be used.

4.1.2 Frame Design Concepts

The requirements of mobility without heavy machinery directed our design to industrial grade casters rather than forklift mounts. An engineering drawing of the frame is shown in Appendix 9. The drip pan was created by riveting sheet metal to the bottom of the frame and sealing it with an oil resistant caulk.

4.1.3 Hydraulic Accumulator

The reason that an accumulator is used in this hydraulic system is to suppress sudden pressure fluctuations; this is referred to as hydraulic shock suppression. This allows the hydraulic system to respond more quickly to power demand and dampens pulsations. In our design, we do not need to store pressure since our pump and motor will handle the demands of our system.

There are two different styles of accumulators: piston or diaphragm type. Each of these devices uses a separating device (piston or diaphragm) to separate compressed nitrogen from a reservoir of hydraulic fluid. The problem is that it acts as a fluid and pressure reservoir; both are not required in our system. However, an inline hydraulic shock suppressor will effectively handle the hydraulic shock without acting as a pressure or fluid reservoir. This shock suppressor has a cylindrical rubber bladder filled with pressurized nitrogen. The bladder separates the nitrogen from a flow of hydraulic fluid through the device. When hydraulic shock occurs, the rise in fluid pressure causes the nitrogen to compress, thus dampening the impact load incurred on fittings and valves throughout the rest of the hydraulic system.

4.2 Peripheral Systems

4.2.1 Intake System

The intake air will initially pass through an air filter. The primary design issue of the filter is to minimize the pressure drop across the filter. After the filter, the mass flow rate of the air flow will need to be measured. There are two methods of measuring the mass flow rate of the intake flow: mass or volume measurement. As mentioned in the engineering specifications, a component will be added to dampen the pulsations of the intake flow. Benchmarking studies in the AutoLab have shown that either a surge tank or adaptive tuning can reduce the vibrations.

Mass flow measurement can be acquired by a Coriolis mass flow meter. This device measures the resultant magnitude of induced vibrations of a flexible tube. The magnitude of the resultant vibrations is proportional to the mass in the tube. The instrument then calculates the mass flow rate by correlating the frequency of induced vibration and the resultant magnitude to the mass flow rate. Volume flow measurement can be acquired by using a laminar flow element (LFE). This device measures the pressure drop across an array of plates oriented parallel to the flow. The pressure drop associated with the conversion to laminar flow is used to calculate the volume flow rate. Mass flow rate is then calculated using the density of the gas.

Pulsations of the intake flow may interfere with the vibration measurement method of a Coriolis mass flow meter. In addition, observations of which methods are currently used in the AutoLab indicate that the LFE is the preferred instrument used in all test cells. Sizing of this instrument is dependent on the volume flow rate as well as the volumetric efficiency (η_v).

An adaptive tuning system requires ducting the flow through an adjustable length pipe. This system would require greater technical design, cost, and control to obtain the same result as a passive surge tank. Therefore, a passive constant volume surge tank will be designed to dampen pulsations as well as to minimize the pressure loss of the intake system.

4.2.2 Exhaust System

The fittings and ducting must be able to handle the high temperatures of the exhaust gases, 800 K [2]. Therefore, the exhaust system is recommended to be constructed using steel components because they can operate at these high temperatures. In addition to dealing with the high temperature exhaust, this system will have to compensate for the induced pulsations from the exhaust stroke of the engine. This can be handled by adaptive tuning or by a passive surge tank. The passive surge tank was selected for the same reasons as it was for the intake system. The dimensions of the exhaust surge tank will be identical to that of the intake surge tank because it will handle the same mass flow rate. The final design easily accommodates further research; these future technologies that will be implemented will be decisions of future graduate students' work and is, therefore, beyond the scope of this project.

4.2.3 Oil System

The paramount design option for the oil system is the selection of a "dry sump" or "wet sump" design. Initially, we had decided on a dry sump oil system with a low level of oil in the engine block. This would allow a linkage mechanism to follow the piston path and extract data from inside the engine without fully submersing the linkage in oil. This would require an external reservoir to hold the volume of oil and a scavenger pump to remove oil from the block into the reservoir. We can use a wet sump design, since there is a design for the linkage mechanism that utilizes a shorter link segment which keeps the mechanism from being submerged.

4.2.4 Coolant System

The major components of the cooling system are a pump, a cooling heat exchanger, a reservoir, control valves, and a circulation heater to warm the coolant. The design aspects of this system include the sizing of the pump, the circulation heater, and the cooling element. The remaining specifications of this system are quite standard. The type of coolant used will be a 50% water and 50% glycol mixture commonly used in automotive engines. The system will operate at a pressure of 100 kPa, again, a standard pressure used for automotive engines.

4.2.5 Fuel System

The fuel system will need to deliver fuel to the engine at a high pressure, approximately 6.9 MPa. This pressure is required to operate direct injection technology for load control of HCCI. This pressure can be maintained in two manners. One design uses a pump to pressurize an accumulator with fuel. This is then used during testing as a pressurized fuel reservoir. The down side of this design is that there is a finite amount of fuel in the tank and, therefore, the duration of testing would be limited. Another design is to run a high pressure pump continuously while the engine is running. This design would require the implementation of a pressure regulator to depressurize access pressure which is generated when the engine is using less than the provided pressure. However, this will allow longer test durations for the engine. This method is the preferred approach for the fuel system design. This approach to designing the fuel system will require a method to cool the fuel. This is required because the temperature of the fuel increases when it is regulated back to the pump supply pressure.

4.3 Test Cell Layout

Ideally, the test cell will hold as many research engines as possible. This can be done in a few different configurations of the test bed layout. One must also allot proper space for the peripheral systems. Assuming a smaller AVL dynamometer is acquired, the test bed can be designed to have one single cylinder engine on one side of the test bed and a multi-cylinder engine on the other side. This would allow room for both engines' peripheral systems but only one single cylinder engine would be on the test bed. An alternative layout would allow an additional single cylinder engine to be placed on the test bed. This design requires both single cylinder engines to be aligned parallel to each other, as seen in Appendix 2. Both single cylinder engines would be oriented with the flywheel pointing in opposite directions in order for the dynamometer connections to fit.

5. ENGINEERING DESIGN ANALYSIS AND FINAL DESIGN DESCRIPTION

As stated in the project statement, one of the project deliverables is a design manual for the peripheral systems. We have benchmarked various peripheral systems in other engine test cells. Various technologies which may be implemented into each system have been identified and have been included into the designs of each peripheral system. We have generated final designs for all of the systems and have had them reviewed and approved by our Sponsor and Technical

Advisor. For each final concept, we have listed preliminary part types and specifications needed for each system.

5.1 Intake System

As previously mentioned, the intake system is comprised of an air filter, a LFE, a surge tank, and a throttle. The sizing of these components hinges on the volume flow rate of the engine. The calculations for the total volume inducted and required volume flow rate can be found in Appendix 10. The final design schematic of the intake system can be seen in Appendix 11.

5.1.1 Air Filter

A performance level air filter for a production level engine will be used. A performance level filter is being used instead of an OEM part because they are designed to lower the pressure drop across the filter. An additional benefit is that the performance air filter can be cleaned, where OEM air filters must be replaced when dirty. However, a custom flange must be fabricated to mount the air filter to the intake system. The selection of the air filter is related to the selection of the throttle body.

5.1.2 Laminar Flow Element

The idealized intake system should have a maximum capacity of 1175 LPM. However, volumetric efficiency maintains a consistent value between 80-90% for various speeds and engine sizes [2]. Therefore, the capacity of the LFE must be greater than or equal to 1075 LPM. A Meriam 50MH10-2 LFE has a capacity of 1130 LPM and should cover the expected operational range. See Appendix 10 for more details on these calculations.

5.1.3 Throttle Body

The cost and time required to custom build a throttle body can be avoided by purchasing an OEM production throttle body that is appropriately sized to the research. Therefore, the throttle body from a production engine with a similar or larger displaced volume will suffice. This will both ensure the throttle body has the capacity for the research engine and will accommodate potential future changes to the bore diameter or stroke length.

Because the fuel injection system from a GM 2.4 L LE5 I4 engine will be used, the possibility of using the associated throttle body was investigated. The displaced volume of a single cylinder of the GM engine is 0.6 L. Because the engine is an inline four cylinder engine, the intake stroke events do not overlap. This means the throttle manages the airflow for each cylinder separately. Because the production engine's displaced volume is slightly larger than that of the research engine, this throttle body should be used.

An additional benefit is that this throttle body is an "Electronic Controlled Throttle", signifying that it is already mounted with an electronic motor. Installation will only require mounting of the unit and creating a control program for the control computer. This eliminates the need to purchase an additional control motor.

5.1.4 Surge Tank

The surge tank is used to reduce the pressure drop experienced by the engine during the intake stroke. The engine supplies the work to pull the air into the cylinder when the intake valve opens.

Without a surge tank, the engine must work against the resistance of the filter and the LFE. With a surge tank, typically installed close to the engine head, the engine draws air from this reservoir. This reservoir allows the induction process to occur at a faster rate. The surge tank facilitates this by using the compressibility of air to generate a slight vacuum in the tank once the engine inducts the air. This slight vacuum then provides the work to pull the air through the filter and LFE during the time at which the engine is not inducting air.

The surge tank must be custom sized to the displaced volume of the engine. Our research engineer advised us that the surge tank should have approximately 10 times the displaced volume. However, benchmarking studies indicate that this volume is a conservative value and most surge tanks are 50-100% percent larger. In addition to this size requirement, the ratio of the diameter and height were selected to provide area on the top and bottom of the tank for additional components to be fastened. This is also the reason why the entrance and exit port of the tank are offset. An engineering drawing of the surge tank can be seen in Appendix 12.

5.2 Exhaust System

All the components in the exhaust system must be able to handle the high temperatures created during the combustion process. A temperature approximation was done using a constant pressure cycle with no heat losses during compression and expansion. The maximum temperature that the exhaust will need to withstand is approximately 930 K. Details of this calculation are shown in Appendix 13. Since heat loss will occur during the compression and expansion cycles, this approximation will be higher than the actual temperature.

The inducted volume will not change between the intake and exhaust. Although the pressure, temperature, and density of the reactants of combustion will differ compared to the inducted air, the volume flow rate will only marginally increase due to the addition of fuel. Therefore, the surge tank should be identical to the intake surge tank already described.

A back pressure valve will simulate the back pressure that an on-vehicle engine would normally experience. This back pressure is caused by mufflers, exhaust ducting, and/or catalytic converters. The back pressure must be adapted to various testing conditions. Therefore, the valve will require a control motor to be mounted to it. This will allow the test cell operator to remotely change the valve setting without entering the test cell during testing.

The vibrations from the engine during firing will propagate along any system that is attached to the engine. This is an issue in the exhaust system because it is rigidly attached to the engine by steel tubing and non-flexible hoses. Therefore, a bellow must be attached between the surge tank and the exhaust piping. A bellow is a semi-flexible piece of tubing which dampens any vibration which it experiences. In addition, the system must also have extra space in the ducting for future research components to be added, i.e., turbochargers, EGR elements, or throttle plates. The final design schematic of the exhaust system can be seen in Appendix 14.

5.3 Oil System

The oil cooler could either be a custom built design or may be purchased from an existing vehicle. Due to no standard oil cooling system existing for this size engine, it will be cheaper and more time effective to purchase an OEM oil cooling system. The design calls for an oil cooler from the 5.4 L V8 Ford Triton engine. This will have plenty of capacity to remove the excess heat from the oil.

The size of the circulation heater is limited by the highest capacity circuit, 240 V and 15 A, in the test cell. Benchmarking studies of similar sized oil heating systems in the AutoLab have indicated that the electrical heating element will require an approximate resistance of 60 Ohms. According to Ohm's law, the maximum current drawn by the heater will be 4.0 A. The benchmarking study indicated that two heaters of this size will be required to heat the oil during the engine start up period (15 min). According to Ohm's Law a circulation heater with an output capacity of 2 kW will be sufficient. The size of the oil pump was also obtained from the benchmark studies. The required pump capacity is 2 GPM at approximately 414 kPa.

Further explanation on how the oil system was sized and selected is given in Appendix 15. The final design schematic of the oil system can be seen in Appendix 16.

5.4 Coolant System

The cooling pump selected will be a high volume centrifugal pump that will have a flow rate capacity of 120 g/s. Market research has shown that a pump of this capacity will require a 375 W motor. The heater will be rated at 3kW with a Watt density of 120 W/in^2. The heater will feature a mechanical thermostat which will regulate the coolant to 350 K during testing. The heater will have a safety feature which will automatically turn off the heater if the temperature exceeds 370 K. This limit was set forth by the limitation of other components in this system.

The heat exchanger will use city tap water to cool the 50/50 glycol-water coolant. The cooling heat exchanger will be a counter flow heat exchanger. The required surface area of the heat exchanger will be 1.3 sq. m with a flow capacity of 11 GPM. This was calculated from the convection coefficients of the coolant and the water, the temperature of the coolant and the water entering and exiting the heat exchanger, and the heat produced from the engine, 3.7 kW max. The calculations for the specified heat exchanger can be found in Appendix 17. The final design schematic of the coolant system can be seen in Appendix 18.

5.5 Fuel System

The purpose of the boost pump is to supply enough pressure so that the fuel can overcome the resistance induced by the filter and mass flow meter. The pressure of the fuel supply of the test cell is 21 kPa. Accounting for the manufacturer's provided pressure drop of the components, a pressure of 70 kPa will be adequate. The filter rating will need to be between 2 to 10 microns. This is required because the fuel injector system has very tight clearances and any macroscopic contaminate can cause the injector to seize.

As mentioned in the engineering specifications section, the rail pump will be required to pressurize the fuel to 7 MPa. This pressure is required because the test engine will be utilizing gasoline direct injection (GDI) as its method of load control. Because GDI is a recent automotive technology, customizing the fuel pump and injection system requires the design ability of an automotive OEM. Therefore, the high-pressure pump and fuel rail will be purchased from Bosch performance, a leading supplier of GDI technology. This system will include the pressure regulator to maintain the required pressure. Further explanation on how the fuel system was sized and selected is given in Appendix 19. The final design schematic of the fuel system can be seen in Appendix 20.

6. MANUFACTURING PLAN

6.1 Dynamometer Frame

The customer requirement of mobility of the dynamometer eliminated the possibility of bolting the dynamometer to the test bed to isolate vibrations generated by the dynamometer. Therefore, a rigid frame was manufactured to act as a mobile foundation for the dynamometer. The frame was made out of square three inch steel tubing. All joints were welded together to minimize the chance that a fastener may work itself free during the dynamometer's lifetime. Four casters were attached to the bottom of the frame to fulfill the customer's requirement of mobility. Further vibrations were prevented by the installation of a dampening flange between the pump and the electric motor as well as dampening bars between the electric motor and the frame.

6.2 Hydraulic Motor to Engine Mounts

The part of the dynamometer which is coupled to the engine must be rigidly attached to the engine. This is to ensure that the engine block will not move when power is supplied to the engine. These brackets were all made from aluminum bar stock. It required drilling hole patterns to ensure the hydraulic motor axis lined up with the crankshaft axis. Previous designs implemented made engine setup an arduous task because it required a "guess and check" method of selecting shims to obtain the correct alignment. The new design has one spacer that was machined to the correct dimension, thus eliminating and "guess and check" aligning method.

6.3 Dynamometer Hydraulic System

Once the dynamometer cart has been fully assembled, time must be spent to assemble the components to fit together inside of the cart. Care must be taken to ensure that high pressure fitting are not swapped with return (low) pressure fittings. Once all the parts have arrived, some additional fitting may be required to be purchased to couple the hydraulic system to the electronic control system; this coupling is beyond the scope of the project.

7. VALIDATION PLAN

7.1 Hydraulic Dynamometer

Complete validation of the hydraulic dynamometer can only be accomplished when the test cell is fully operational. At the moment, the engine to which the dynamometer is coupled is missing its flywheel and valve train assembly. Therefore, validation of operation is impossible. Once these missing pieces are procured, the engine can be assembled and coupled to the dynamometer and validation can occur.

Complete assembly of the mechanical systems of the dynamometer has been delayed due to long lead times for key components of the dynamometer, electric motor, surge suppressor and hydraulic filter. Our dynamometer has been validated and approved by a U of M Research Engineer, Kevin Morrison. He approved the manufacturing and assembly of the engine mounting brackets, dynamometer frame, internal purchased components and plumbing connections. The size requirement of fitting through a standard door opening was validated by wheeling the dynamometer cart through several doors in the AutoLab, thus, also validating the ability to move without heavy machinery.

7.2 Peripheral Systems Project Validation

Validation of the peripheral systems designs is at the discretion of our Project Sponsor and Research Engineer. We have completed all final designs of the peripheral systems along with the sizing, part type, and manufacturer of each system part. The approximate cost and manufacturer can be found in Appendix 21. Following this completion, our Sponsor and Engineering Specialist have approved our designs and part selection. While the design portion of our systems has been validated, the manufacturing and operational validation of the system has not yet begun. This construction and assembly is beyond the scope of our project. A design manual of our peripheral systems designs will be used for manufacturing projects for future Independent or Senior Projects. The designs in the manual will be enough to run the engine and validate the hydraulic dynamometer.

8. PROJECT TIMELINE AND PLAN

The deliverables of the project are the assembly of the mechanical system of the hydraulic dynamometer. This will include the frame, all the hydraulic components, and the mounting brackets to couple the dynamometer to the engine. In addition to the dynamometer, finalized design of each peripheral system was completed. The major components for each system have been specified and are ready to order. Therefore, when the engine test cell is renovated, the peripheral systems can be purchased and installed when requested.

The frame of the dynamometer is fully assembled and the mounting brackets are machined and attached to the engine. All of the internal dynamometer components have been assembled into the frame except for the electric motor, hydraulic filter assembly and surge suppressor. We do expect to receive these components and complete assembly of the dynamometer by the end of the semester.

In order to effectively undertake the design of the peripheral systems, we assigned a functional leader to each system with group support when necessary. The system design responsibility is listed below. The fuel system has the most complexity and, therefore, required a focused team effort from the entire group.

System	Design Responsibility
Intake	Dave Ault
Coolant	Chris Marchese
Oil	Randy Jones
Exhaust	Dan Murray
Fuel	Entire Group

The original Gantt chart has been updated and we have followed each task up to and including the Design Expo. To date, the following tasks have been completed: benchmark studies of existing hydraulic dynamometers, benchmark studies of existing peripheral systems, ordering of dynamometer parts, procurement of raw material for dynamometer frame and mounting brackets, and assembling the dynamometer. We have the dates of completion organized in a Gantt chart in Appendix 22. Our target was to have the mechanical systems for the hydraulic dynamometer and finalized design of the peripheral systems completed by the Design Expo and, as previously stated, this was completed.

In the future, more work will be needed to mechanically validate the hydraulic dynamometer and the peripheral systems. These projects will be undertaken through future Independent and Senior Design Projects. For the hydraulic dynamometer, the electronic controls will need to be installed on the cart and hooked to a power supply to validate its operation. For the peripheral systems, all the parts for each system will need to be ordered and assembled in the test cell. This assembly and installation of the systems will hinge on the test cell layout. The old engine and dynamometer must be replaced with new hardware before the peripheral system installation can occur.

9. DESIGN CRITIQUE AND PROBLEM ANALYSIS

Upon physical validation (operation in conjunction with the engine), our dynamometer and peripheral systems will meet all of the customer's requirements. The hydraulic dynamometer has a load range that is equivalent to the single cylinder engine's output, it is compact in size and can fit through a standard doorframe. The peripheral systems have been designed to be modular, require limited space, and be easily accessible for maintenance.

While we are pleased with the results of this project, there are many areas of design and manufacturing that could have been improved. For the hydraulic dynamometer, more advanced ordering of the components would have allowed complete assembly of the mechanical system of the dynamometer. A few key items were delayed in the ordering and delivery process. The dynamometer was loosely assembled in such a manner that will allow the missing components to be easily attached when they arrive.

The dynamometer cart fits through all doors and entrance ways in the Auto Lab facility. However, our cart does not fit through some doors to buildings where testing will not be performed. A more compact dynamometer would thus be able to fit through all doors on campus. Additional time would have allowed us to design a narrower frame for the dynamometer.

Our designs of the peripheral systems have been approved by our Research Assistant and our Project Sponsor. Therefore, there are no flaws to the systems from a design perspective. Functional flaws will be made apparent when the systems are assembled. Given more time, each system could be procured and assembled; thus allowing functional validation.

10. INFORMATION SOURCES

Our project required significant research from various sources. We used the internet to research the basics on commercially available dynamometers [6][7]. We found a 1968 SAE paper which explained the details of how hydraulic dynamometers function. There was an operation handbook for a similar type of dynamometer in the AutoLab. The textbooks from the prerequisite mechanical engineering classes of thermodynamics, heat transfer, statics, and fluid mechanics were used in the engineering analysis for each component of the peripheral systems. Component selection and sizing information was obtained from vendor information, benchmarking research, and advice from University research personnel.

11. ACKNOWLEDGEMENTS

We would like to acknowledge certain individuals who made this project possible and supported our progress. They include the following: Professor Dennis Assanis, Kevin Morrison, Professor Alan Wineman, Professor Zoran Filipi, Brad Zigler, Steve Busch, Andrew Ickes, Orgun Guralp, Jonathan Hagena, Mike Smith, Chandrasekaran Sethu, Alberto Lopez, and Mark Hoffman.

12. CONCLUSIONS

At the conclusion of our project, all deliverables as agreed to by our Project Sponsor are completed and submitted. The main goals achieved are the following:

- Procured the mechanical components for hydraulic dynamometer
- Manufactured and assembled dynamometer frame
- Designed and assembled engine-hydraulic motor mounting brackets
- Completed thorough benchmarking and study of existing peripheral systems facilitated correct system design and sizing of major components of peripheral systems
- Created budget for test cell setup
- Proposed designs for new test cell layout
- Started the process of setting up engine; obtained correct size engine head, cylinder liner, crankshaft and piston as desired by Project Sponsor

More work is still needed to be done in the test cell before research can be done. Our group has started the assembly of the test cell. Hopefully, in the near future, professors, graduate students and researchers can benefit from the work that we have done to facilitate assembly of this test cell.

13. BIBLIOGRAPHY

[1] Pulkrabek, Willard W. "Engineering Fundamentals of the Internal Combustion Engine, 2nd Edition." Pearson: Prentice Hall, 2004. Upper Saddle River, NJ 07458.

[2] Heywood, John B. "Internal Combustion Engine Fundamentals." McGraw-Hill, 1988. New York, NY.

[3] Narayan Rao, N.N. "Basic Theory of Hydraulic Dynamometers and Retarders." SAE International Technical Paper #680178. 1968

[4] DeWitt, David P. and Incropera, Frank P. "Introduction to Heat Transfer, 4th Edition" John Wiley & Sons, 2002. New York, NY.

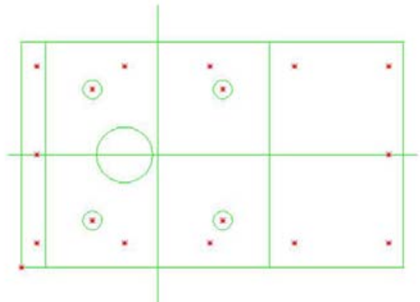
[5] Bornakke, Claus, Sonntag, Richard E., and Van Wylen, Gordon J. "Fundamentals of Thermodynamics." Wiley, 2002. New York, NY.

[6] AVL. http://www.avl.com.

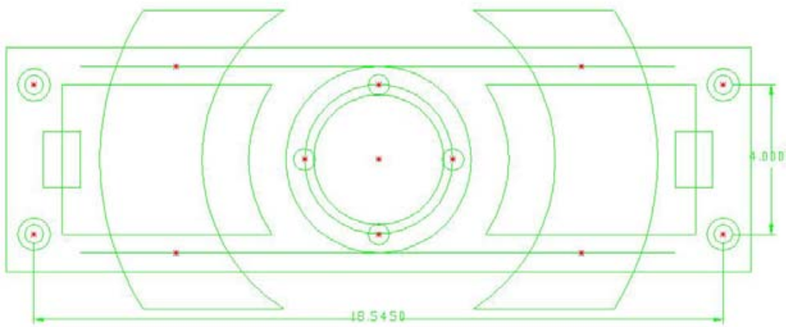
[7] FEV. http://www.fev.com.

14. APPENDICES

Appendix 1: Mounting Brackets

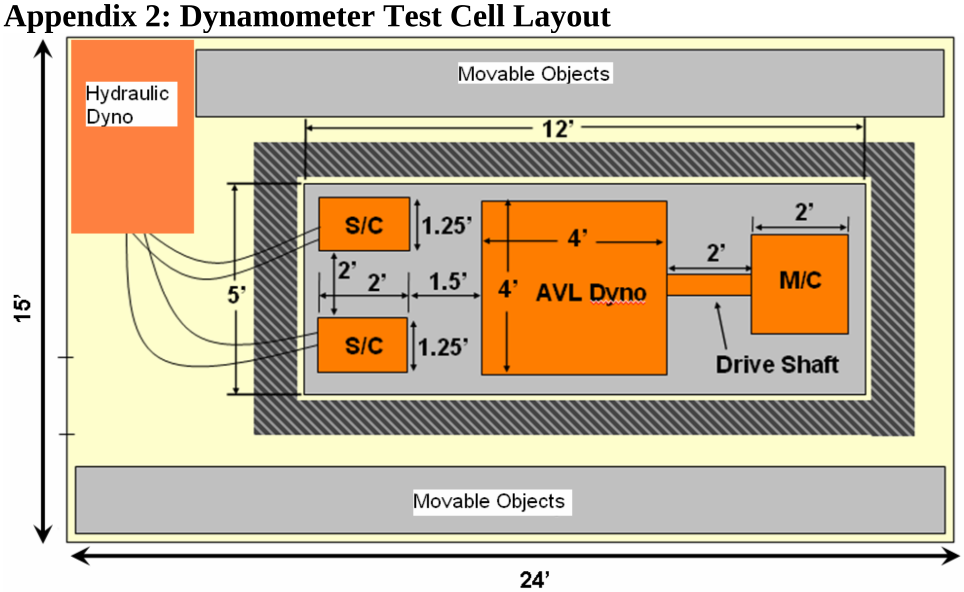


Ricardo Hydra Crankcase Mounting Bracket

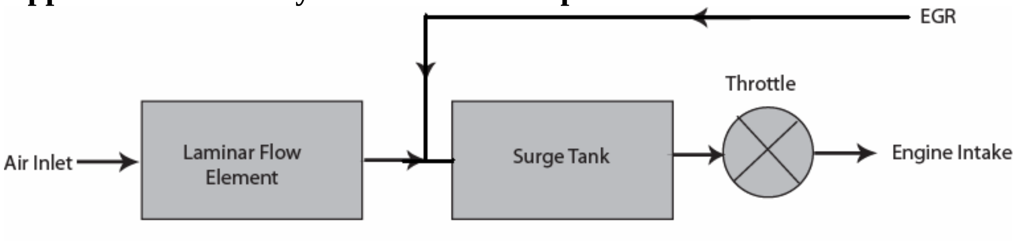


Hydraulic Dynamometer Motor Mounting Bracket

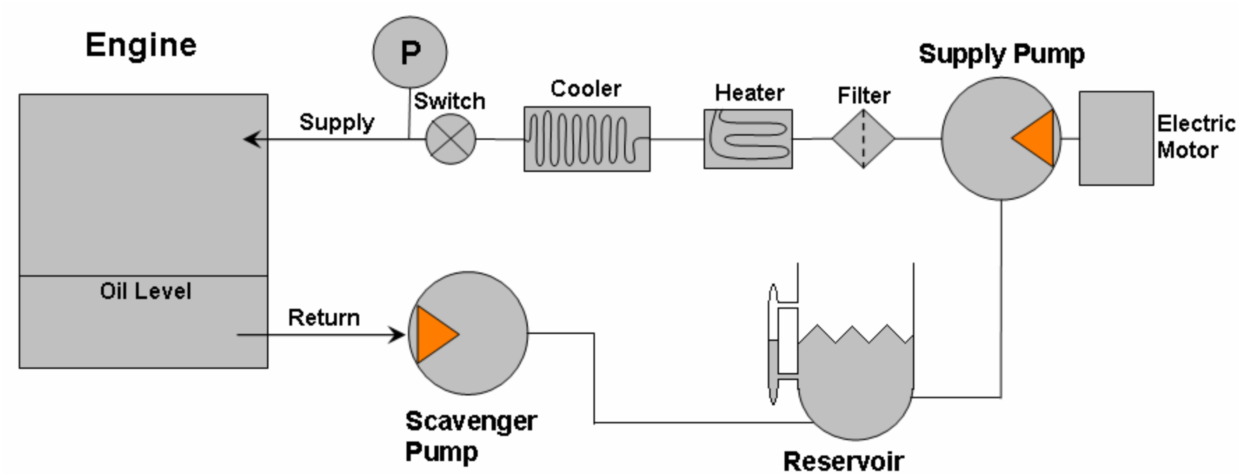
Appendix 2: Dynamometer Test Cell Layout



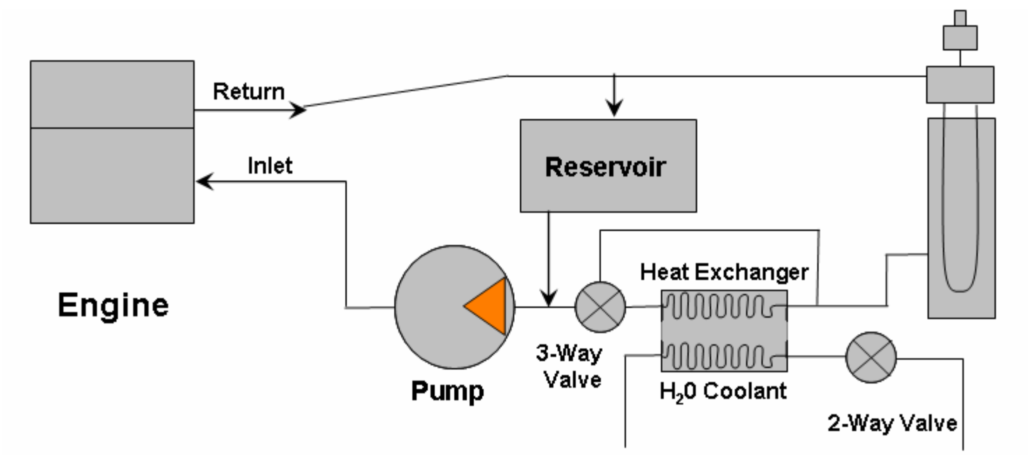
Appendix 3: Intake System Initial Concepts



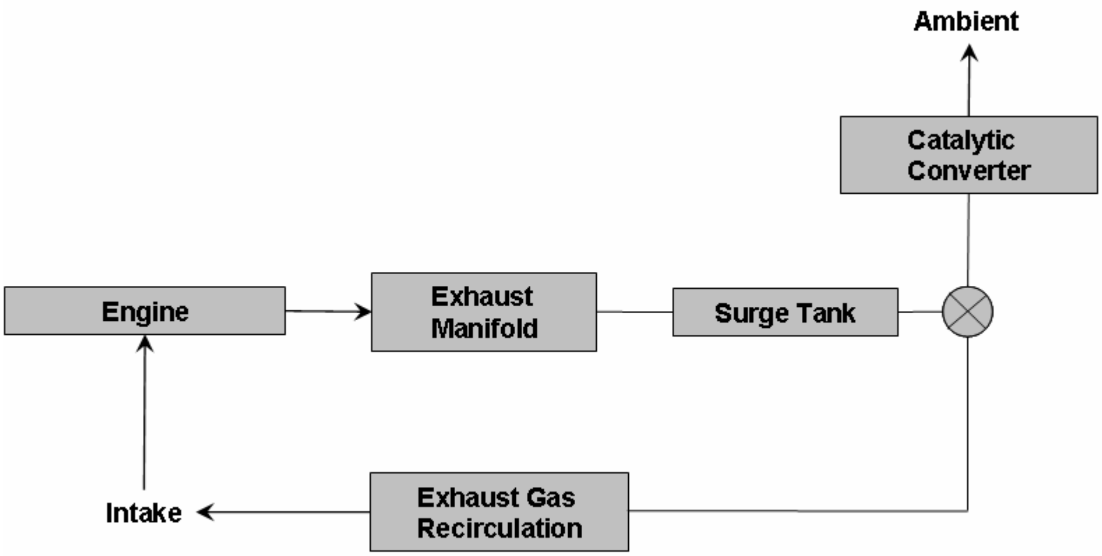
Appendix 4: Oil System Initial Concepts



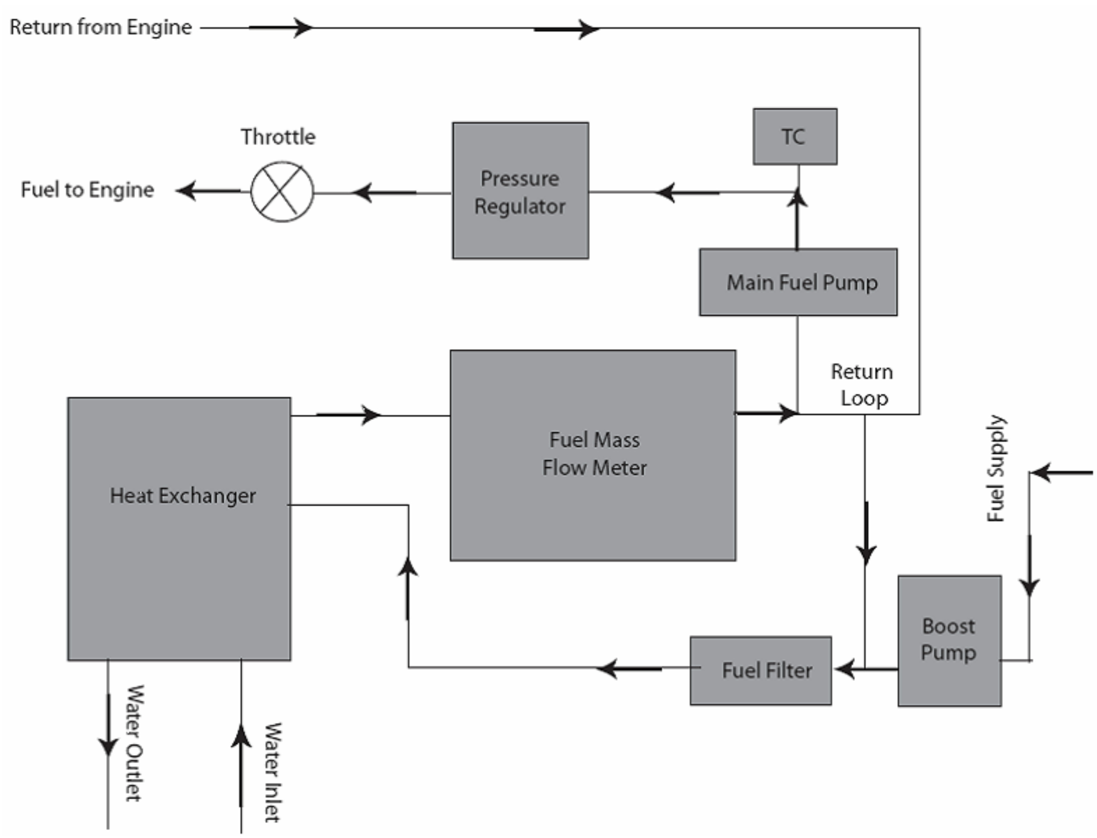
Appendix 5: Coolant System Initial Concepts



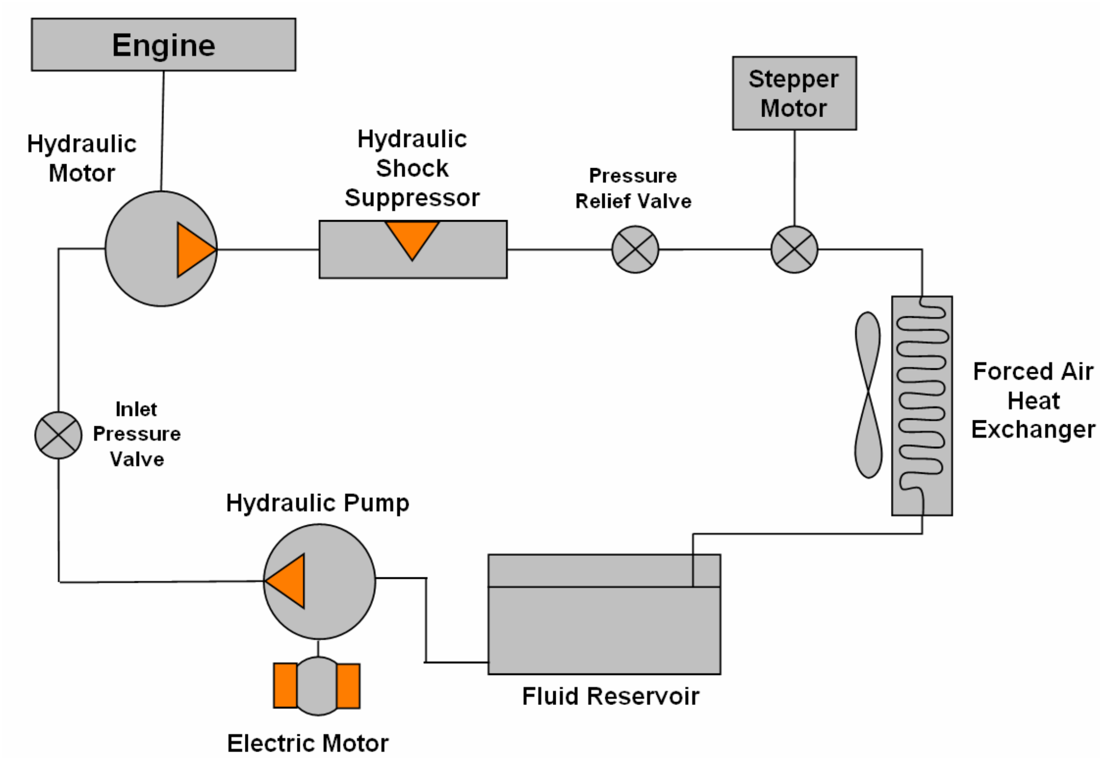
Appendix 6: Exhaust System Initial Concepts



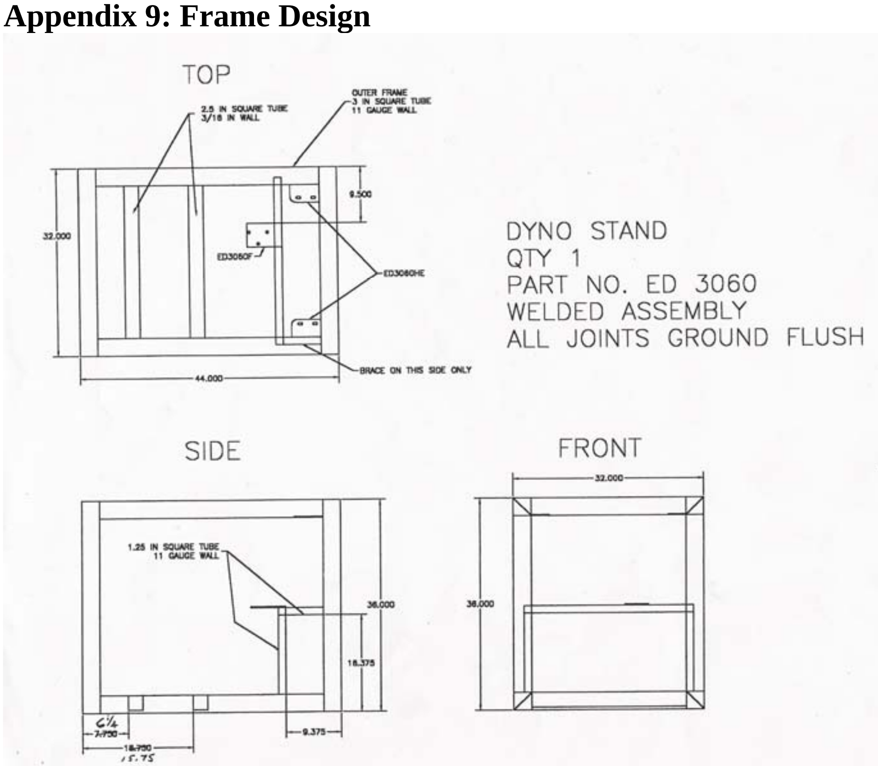
Appendix 7: Fuel System Initial Concepts



Appendix 8: Hydraulic Motor Flow Charts



Appendix 9: Frame Design



Appendix 10: Intake System Design

The following calculations produce the required volume flow rate. The factor of ½ in the volume flow rate equation is to account for the basic function of an internal combustion engine; the intake event occurs every other revolution.

Engine Bore Diameter:	D = 86 mm
Engine Piston Stroke Length:	L = 92 mm
Expected Maximum Engine Speed:	S = 4400 RPM

$$V_{DISP} = \frac{\pi}{4} D^2 \times L = 0.534 \text{ Liters}$$

$$\dot{V} = V_{DISP} \times S \times \frac{1}{2} = 1195 \text{ LPM}$$

Air Filter

A performance level air filter for a production level engine will be used. A performance level filter is being used instead of an OEM part because they are designed to increase the efficiency of the component; in this case, it causes a lower pressure drop. The selection of exactly which model of air filter is related to the selection of the throttle body.

Laminar Flow Element

Sizing of the LFE is dependent on the volume flow rate calculated above as well as the volumetric efficiency (η_v). The volumetric efficiency is defined below as the ratio of the mass of the air inducted into the intake system to the mass of air of displaced volume of the piston.

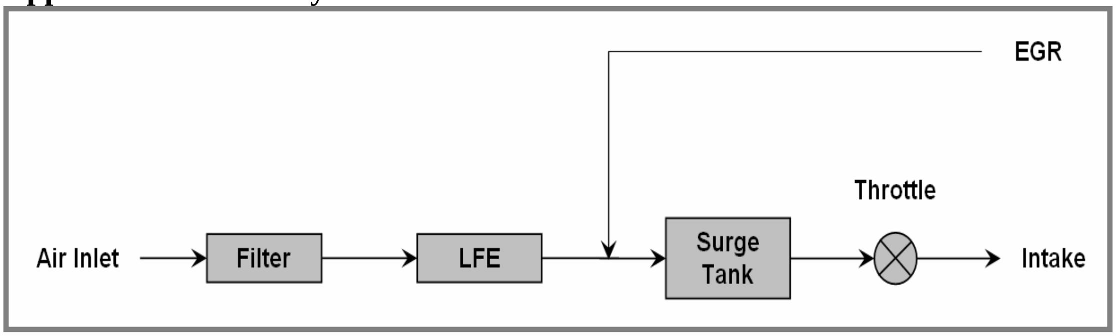
$$\eta_v = \frac{m_a}{\rho_{a,i} V_{DISP}}$$

The difference occurs because air is heated and, thus, expanded as it enters the cylinder. This expansion causes a lower amount of mass to be trapped in the cylinder compared to the amount that is expected due to the displaced volume of the piston. By definition, this efficiency is always less than 100%, typically 80-90% [2]. Therefore, the capacity of the LFE must be greater than or equal to 1075 LPM. A Meriam 50MH10-2 LFE has a capacity of 1130 LPM, and should cover the expected operational range.

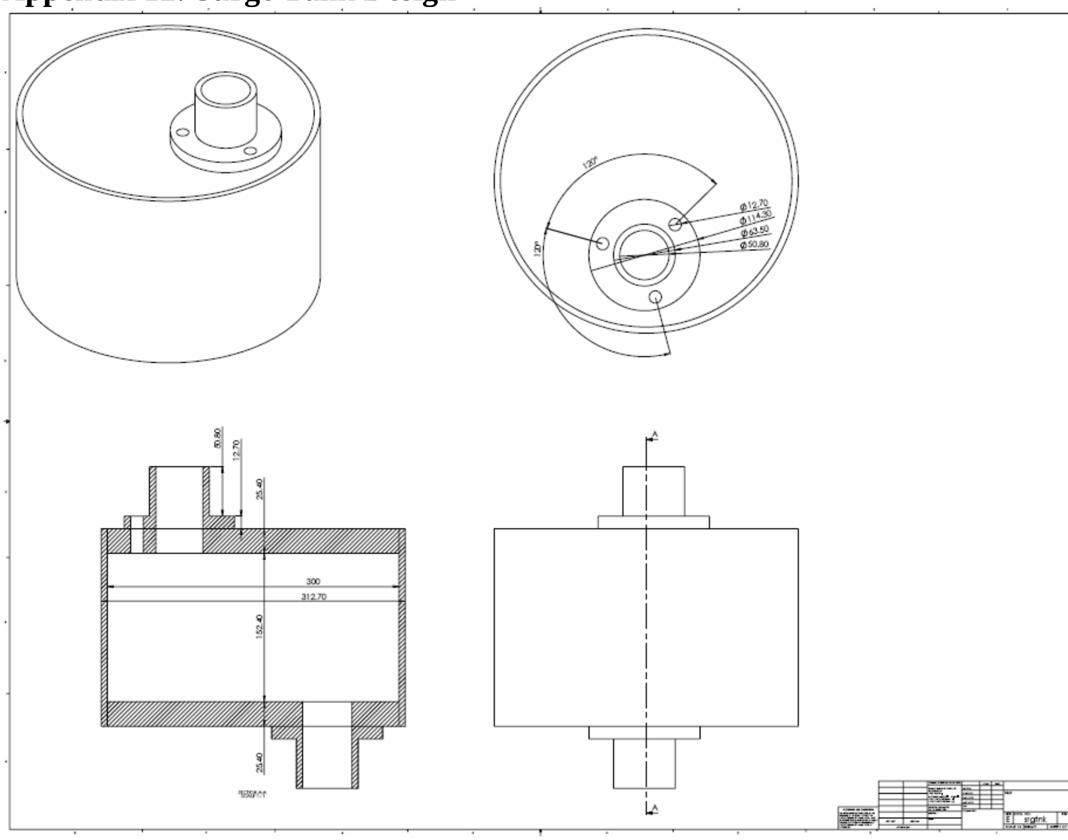
Throttle Body and Surge Tank

These components were sized from the above volume flow rates. The details of each component were already discussed in Section 5.1.3 and 5.1.4.

Appendix 11: Intake System Flow Chart



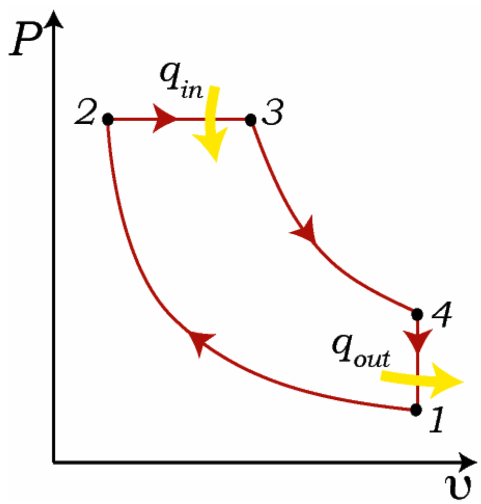
Appendix 12: Surge Tank Design



Appendix 13: Exhaust System Design

The temperature that the exhaust system must be able to withstand is calculated by assuming that the engine is operating with constant pressure combustion. Our single cylinder engine will be using HCCI technology which is not exactly a constant pressure cycle but will give a good approximation. We must assume that the temperature and pressure of the intake is ambient. We used 300 K and 100 kPa.

Since we are attempting to approximate the highest pressure and temperature possible for the exhaust gas, we can assume that the expansion and compression stroke are adiabatic and that the combustion process happens by a constant heat addition. The compression ratio of the engine can be varied but, at the moment, the compression ratio is 15. The ratio of specific heats for the working fluid will be constant approximately 1.3.



Using the Second Law of Thermodynamics, the temperature at the end of the compression stroke can be found.

$$\frac{T_2}{T_1} = r^{\gamma - 1}$$

$$\left(\frac{P_2}{P_1}\right)^{\gamma - 1/\gamma} = r^{\gamma - 1}$$

r – compression ratio
γ – ratio of specific heats

To calculate the heat input from the combustion process, the mass of fuel and the heating value of the fuel must be obtained. Since both of these variables will vary during future testing situations, a heat input of 1600 kJ is a good approximation. From the First Law of Thermodynamics, the temperature at the end of combustion (T_3) can be calculated.

$$m_f Q_{HV} = c_p (T_3 - T_2)$$

Using T_3 and the ideal gas law and setting the mass equal for state 2 and 3, one can find the volume at the end of combustion (V_3).

$$\frac{P_3 V_3}{RT_3} = \frac{P_2 V_2}{RT_2}$$

R – gas constant

With V3 and T3 calculated from above, the exhaust gas temperature can be found using the Second Law understanding that V4 can be calculated from the compression ratio.

$$r = \frac{V_D + V_C}{V_C}$$

$$V_4 = V_D + V_C$$

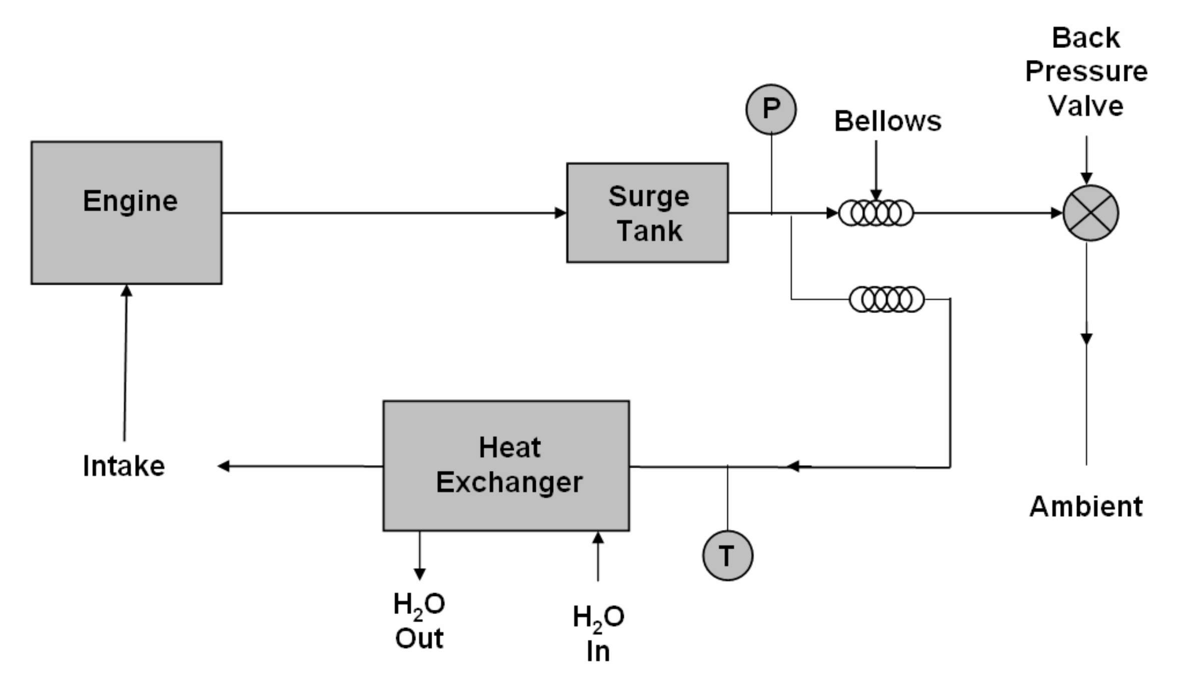
The displaced volume (VD) is calculated in intake analysis in Appendix 12.

$$\frac{T_3}{T_4} = \left(\frac{V_4}{V_3}\right)^{\gamma - 1}$$

Please see the below for more detail on temperature calculation.

T_1	300	K
P_1	100	kPa
Gamma	1.3	
Compression Ratio	15	
Specific Heat CP	1.7	kJ/kg*K
T_2	676	K
P_2	3380	kPa
Q_input	1600	kJ
T_3	1617	K
V_D	0.00055	m^3
V_C (V_2)	0.000039	m^3
V_4	0.00059	m^3
V_3	0.00009	m^3
T_4 (Exhaust Temperature)	932	K

Appendix 14: Exhaust System Flow Chart



Appendix 15: Oil System Design

The oil system will support the function of lubricating, cooling, cleaning and protecting the engine during operation. The oil system will require a heating element to warm up the oil during engine start up (motoring), as well as a cooling element to cool the oil during normal operation (firing). Specifically, this thermal management system must maintain an engine operating temperature between 340 ºK - 400 ºK. This temperature range maintains a relatively constant oil viscosity [2]. Benchmarking studies of similar oil systems in the AutoLab, (primarily the GM-CRL HCCI Heat Transfer Research Lab) have indicated that the oil pressure to maintain proper lubrication of the engine will be 414 kPa.

The oil system has been designed to fit inside a modular cabinet which can be located at the perimeter of the test cell while being easy to access for maintenance or repair. Below is a discussion of some of the key design aspects of the oil system.

Circulation Heater

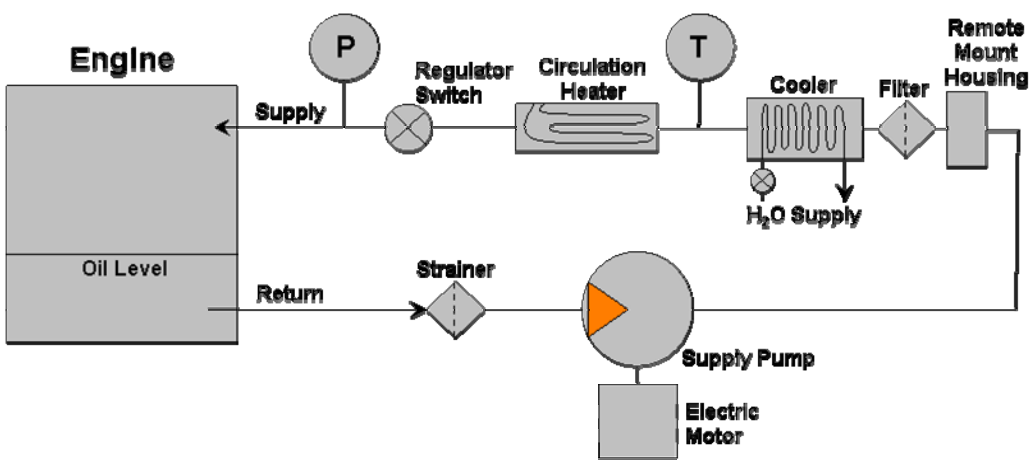
The size of the circulation heater is limited by the highest capacity circuit, 240 V and 15 A, in the test cell. Benchmarking studies of similar sized oil heating systems in the AutoLab have indicated the electrical heating element will require an approximate resistance of 60 Ohms. According to Ohm's law, the maximum power drawn by the heater will be 4.0 A. The benchmarking study indicated that two heaters of this size will be required to heat the oil during the engine start up period (15 min). According to Ohm's Law, a circulation heater with an output capacity of 2-2.5 kW will be sufficient.

Supply Pump
The size of the oil pump was obtained from the benchmark studies of other test cells in the AutoLab. The required pump capacity is 2 GPM at 1800 RPM and approximately 414 kPa. This will allow for adequate lubricant flow throughout the engine cavities.

Oil Cooler
The critical requirement of the oil cooler is capacity. However, for this single cylinder engine's relatively low temperature output, a custom ordered part to fit the application would cost more than a controlled service part from an existing automotive OEM. The part ordered is for an eight cylinder 5.4 liter engine and will have plenty of capacity.

Oil Filter Assembly
The oil cooler and filter will attach together and can be remote mounted in the oil cabinet with the Hayden 22mm remote mount filter holder. The model of filter selected, a Fram PH2, was selected to fit both the oil cooler and remote mount filter holder and can be purchased from any automotive parts distributor.

Appendix 16: Oil System Flow Chart



Appendix 17: Cooling System Calculations

Cooling Pump

Based on a heat transfer and energy balance for an internal combustion engine, the sizing and capacity of the pump for the cooling system can be found. In Equation 1 below, the engine energy equation can be found along with the equation for heat transfer of the coolant (Equation 2). These two equations and inputs from the engine and fuel parameter, along with numbers obtained from competitive benchmarking, can be used to solve for the mass flow rate of the coolant. When this analysis is completed, the mass flow rate of the coolant is found to be 7 kg per minute. A centrifugal motor with a low power output of 370 W will suffice in meeting the mass flow rate requirement.

Equation 1: $$P_{brake} + \dot{Q}_{cool} + \dot{Q}_{misc} + \dot{m}h_{exhaust} = \dot{m}_f Q_{LHV}$$

Equation 2: $$\dot{Q}_{coolant} = \dot{m} \cdot c_{v\_coolant} \cdot (T_{out\_coolant} - T_{in\_coolant})$$

Coolant Reservoir

The amount of coolant needed to supply the system is based on of the amount needed to circulate through the system. This amount is determined by the amount needed to flow through the peripheral parts, coolant lines, and engine parts which include the engine block and cylinder head. Using competitive benchmarking of other peripheral test cells and benchmarking coolant requirements of single cylinder engines, the amount of coolant for the system should be 5 liters. Taking into account the amount of coolant that will be in the system, the coolant reservoir will need to have the capacity to hold 3 liters of coolant.

Circulation Heater

As previously stated, the engine will need a circulation heater to warm the engine up to optimal conditions during initial operation. From previous internal combustion knowledge, engines are inherent to under perform until the system is heated. For all internal combustion engines, performance graphs and data output show that initial engine speeds produce much lower levels of heat transfer. This is due to the fact that more time is needed for heat transfer to occur to reach optimal conditions. With single cylinder engines, since only one cylinder undergoes combustion as opposed to engines with 4, 6, or 8 cylinders, heat transfer to optimal conditions takes even longer to occur. Given this fact, a circulation heater is required to heat the coolant.

Base on this knowledge, a thermodynamic analysis of the first law can be used to find the appropriate energy input to raise the initial coolant temperature to optimal conditions. The equation below, can be used to find the needed amount of heat input, given the defined mass flow rate of coolant (7 kg/min), specific heat of coolant (1.796 kJ/(kg*K)), initial operating temperature of the coolant (500 K) and room temperature of the coolant (300 K). This equation shows that the amount of heat input needed will be 2.6 kW.

$$\dot{Q} = \dot{W} = \dot{m} \cdot c_v (T_2 - T_1)$$

Leaving room for error, the circulation heater will be rated at 3 kW with a Watt density of 120 W/in^2.

Coolant System Heat Exchanger

From the data that our Engineering Specialist supplied us, we were able to determine what surface area in the heat exchanger we would need in order to dissipate the proper amounts of heat. The Engineering Specialist gave us some approximate data on what typical specifications a one cylinder engine will produce. From these specifications, control of the mass flow rate of water will output a suggested surface area.

The temperature of the coolant after it goes through the engine will be approximately 450 degrees Kelvin and will have a mass flow rate of 3.8 GPM. After it goes though the coolant system, the heat back into the engine should be 380^{o}K. The engine will be producing 3.7 kW. The convection coefficients of the coolant and water from the Red Line Oil Corporation website are 1829 kW/(m^2*C) and 897 kW/(m^2*C), respectively. We applied a safety factor of 2 to our final surface area. The temperature of the water will be that of tap water. We assumed 290 ^{o}K. The specific heats of water and the coolant were found to be 4.186 kJ/(kg*K) and 1.796 kJ/(kg*K), respectively [3].

To find the temperature of the water after it leaves the heat exchanger, we calculate the amount of heat transfer with the change in temperature of the coolant, shown by Equation 1. Assuming that all the heat generated by the coolant must be dissipated into the water (Equation 2), we were able to find the temperature of the water as it leaves the heat exchanger (Equation 3).

Surface area, as given in Equation 8, is a function of the log mean temperature difference (T_{lm}), heat generated (q) and the overall heat transfer coefficient (U). We used the horsepower to find the amount of heat (q) that the heat exchanger would need to exchange. Equation 6 with the inlet and outlet temperatures of the coolant and water was used to find the log mean temperature difference. Equation 7 was used to find the overall effective heat transfer coefficient using the heat transfer coefficients of water and the coolant. From these, we solved Equation 8 to find an approximate surface area needed.

Equation 1: $$q_{coolant} = c_{v\_coolant} \cdot (T_{out\_coolant} - T_{in\_coolant})$$

Equation 2: $$q_{coolant} = q_{H_2O}$$

Equation 3: $$q_{H_2O} = c_{v\_H_2O} \cdot (T_{out\_H_2O} - T_{in\_H_2O})$$

Equation 4: $$\Delta T_2 = (T_{in\_coolant} - T_{out\_H_2O})$$

Equation 5: $$\Delta T_1 = (T_{out\_coolant} - T_{in\_H_2O})$$

Equation 6: $$\Delta T_{lm} = \frac{\Delta T_2 - \Delta T_1}{\ln(\Delta T_2 / \Delta T_1)}$$

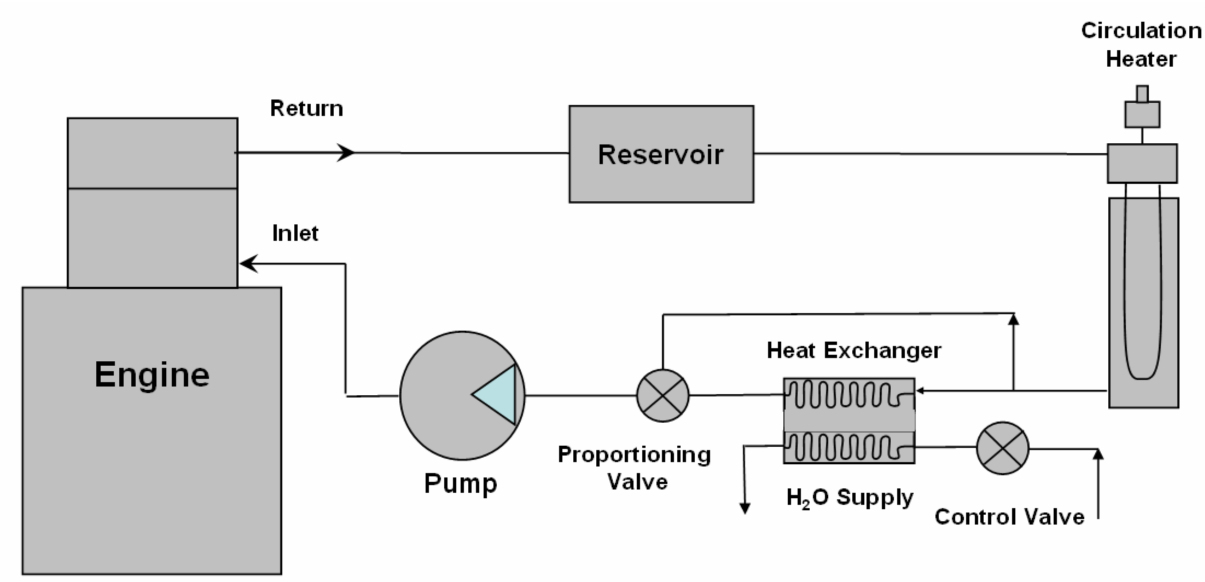
Equation 7: $$U = \frac{1}{{}^1\!/_{h_{coolant}} + {}^1\!/_{h_{H_2O}}}$$

Equation 8: $$q_{engine} = U \cdot A \cdot \Delta T_{lm}$$

Appendix 17 (cont.): Coolant System Calculation

Max Flow Capacity	11	Gallons/min
HP	5	Hp
Safety Factor	2	
Convection Coefficients		
Water	1829	kW/(m^2*K)
Coolant (glycol/h2o)	897	kW/(m^2*K)
Mass Flow Rates		
Water	41.635	kg/min
Coolant (glycol/h2o)	6.92655	kg/min
Specific Heats		
Water	4.186	kJ/(kg*K)
Coolant (glycol/h2o)	1.796	kJ/(kg*K)
Temperatures		
Coolant (glycol/h2o)		
T_in FROM THE ENGINE	453	K
T_out TO THE ENGINE	350	K
H2O		
T_in	294	K
T_out	301.3519533	K
Density		
Coolant (glycol/h2o)	915	kg/m^3
Water	1000	kg/m^3
Heat Transfer From Coolant		
-1281.328631	Joules/min	
U	602	
Q	3728	W
Change T_2	151.65	
Change T_1	56.00	
Change in T_im	96.01	
Surface Area	**0.65**	**m^2**
Sufrace Area	**13.89**	**ft^2**

Appendix 18: Cooling System Flow Chart

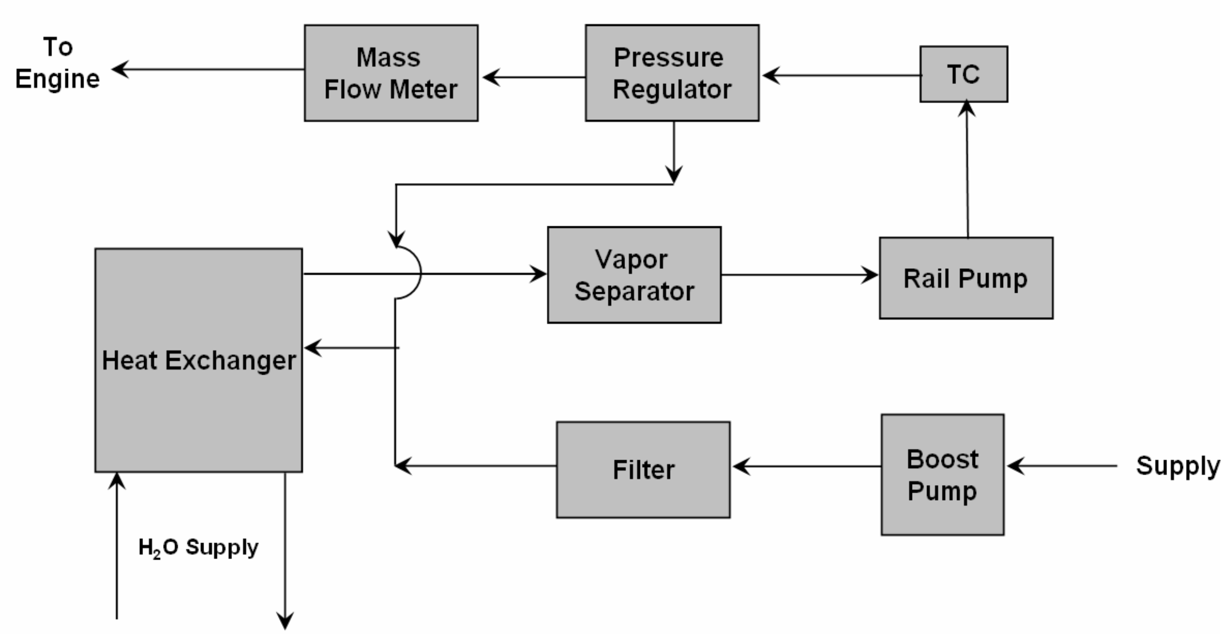


Appendix 19: Fuel System Calculations

<u>Boost pump</u>
The size of the boost pump was found using benchmarking guides of other test cells in the Auto Lab. The building fuel supply is rated at 21 kPa and the pressure is needed to be raised to flow the fuel through the system. From our benchmarking, raising the pressure to 70 kPa is adequate to supply the system.

<u>High Pressure Fuel Pump</u>
As previously stated, the fuel pressure needed to be delivered to the engine will be 7 MPa, which is required for HCCI. HCCI utilizes compression ignition which normally used direct injection. In direct injection, the pressure of the fuel must be at a high pressure due to the fact that the pressure inside of the cylinder will be high. This means that, at the injection, pressure must be higher than that of the cylinder in order to have a dispersed fuel spray from the injector.

Appendix 20: Fuel System Flow Chart



Appendix 21: Peripheral Systems Parts List with Suppliers and Pricing

ME450 AutoLab
Peripheral System Component Budget
10-Dec-06
D. Ault
R. Jones
C. Marchese
D. Murray

Peripheral System		Cost
Fuel		**$8,280**
Cooling		**$871**
Oil		**$1,127**
Exhaust		**$2,480**
Intake		**$2,400**
Cabinets		**$3,000**
	Total	**$18,158**

Fuel

Part	Supplier		Price
Coriolis Mass Flow Me	Automatic Controls		7000
Fuel Filter	McMaster		30
Fuel Pump	Bosch		750
Pressure Regulator	Bosch		150
Boost Pump	Holley		200
Cooling System			150
		Sub Total	8280

Cooling

Part	Supplier		Price
Resevior	McMaster		50
Centrifugal Pump	Grainger		221
Heat Exchanger	McMaster		250
3-Way Valve	McMaster		30
2-Way Valve	McMaster		20
Circulation Heater	McMaster		300
		Sub Total	871

Oil

Part	Supplier		Price
Circulation Heater	McMaster		570
Oil Cooler	Ford		120
Oil Filter	AutoZone		2
Remote Mount Housin	Hayden		60
Supply Pump/Motor	McMaster		300
Strainer			75
		Sub Total	1127

Exhaust

Part	Supplier		Price
Surge Tank	Alro Steel		1000
2 Bellows	McMaster		460
2 Way Valve	McMaster		20
Butterfly Valve	McMaster		1000
		Sub Total	2480

Intake

Part	Supplier		Price
LFE	Meriam		500
Filter	K&N		100
Proportioning Valve	JCI		500
Surge Tank	Alro Steel		1000
Throttle	GM	Sub Total	300
			2400
		Total	15158

Appendix 22: Gantt Chart