Exhaust Bypass Valve for a Homogeneous Charge Compression Ignition (HCCI) Engine

Sponsored by Ruonan Sun, Ph.D.
Environmental Protection Agency

Final Report

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ABSTRACT

Many of the advanced engine technology concepts currently being developed by the Environmental Protection Agency (EPA) require high intake pressure, usually obtained using twin turbochargers. To complete such an operation, an exhaust bypass valve is required to open a desired area of exhaust quickly. The current available valves all have disadvantages: either they are too small (choke the flow), leak (reduce boost), or pop open or jam. We have been asked by Dr. Ruonan Sun of the EPA to design and build a prototype of an exhaust bypass valve that will function at high pressures and temperatures with minimal leakage.

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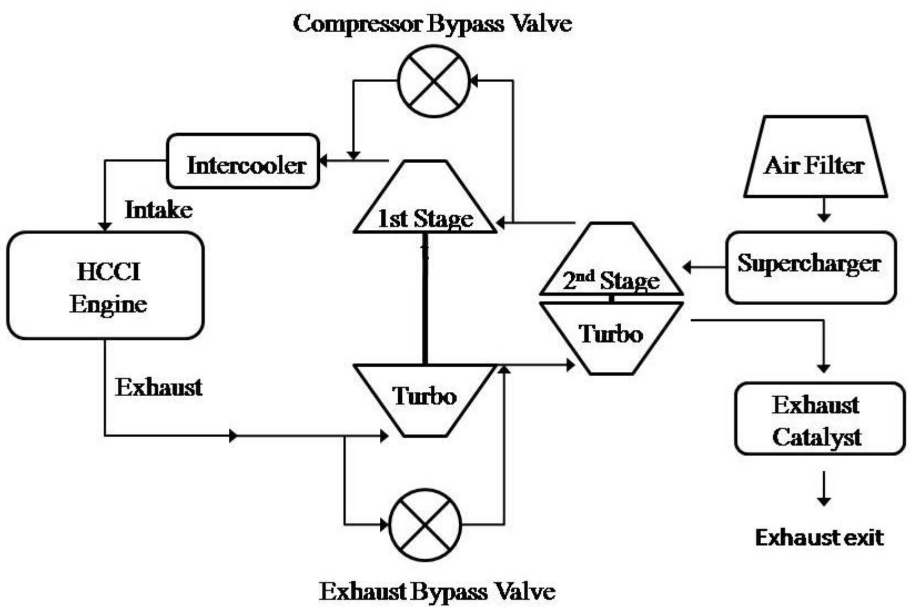
INTRODUCTION

The Environmental Protection Agency (EPA) is in the process of developing several advanced engine technology concepts. One such concept is the Homogeneous Charge Compression Ignition (HCCI) Engine. The HCCI engine uses twin turbochargers to produce high levels of intake pressure needed for operation. A major component for this operation is the exhaust bypass valve. The exhaust bypass valve is used to divert the exhaust flow from the 1st stage, high pressure turbocharger to the 2nd stage, low pressure turbocharger. A diagram of the system can be seen below in Figure 1.

Figure 1: Twin Turbocharger HCCI Engine System



Two turbochargers are necessary in this system because a single turbo has an optimized flow for its particular geometry. This geometry directly corresponds to the engine efficiency. Once the first turbo receives flow greater than its optimized range, the efficiency drops and the exhaust bypass valve opens, directing flow into the 2nd stage turbo. The 2nd stage turbo has a different geometry and therefore a different optimized flow. Diverting the excess exhaust flow into the 2nd stage turbo allows the engine to operate at higher speeds and higher flows while maintaining the highest possible efficiency allowed by turbocharger selection (R. Sun, Personal Communication, September 25, 2007).

The exhaust bypass valve in this operation must withstand very high pressures and temperatures. Current available valves do not meet the needed capabilities. They are either are too small and choke the flow, leak under the high pressures reducing boost, pop open and jam, or do not allow precise control over the timing and proportion of bypassed exhaust gas.

Dr. Ruonan Sun of the EPA has given us the task of developing an exhaust bypass valve that can be used in the HCCI engine operation, specifically the HCCI engine for a Ford Expedition. The valve must be able to function at high pressures and temperatures with minimal leakage and must meet very specific design characteristics provided by Dr. Sun and the EPA. This valve will minimize cost and maximize functionality. We have been asked to provide a detailed design report on the exhaust bypass valve as well as a prototype that conveys our design concept with actual functionality being an added benefit.

INFORMATION SEARCH

Research was completed on the background of the HCCI Engine operation to ensure we understood how the system our exhaust bypass valve is installed in operates. We also needed to get information on any existing valves or patents and how they could relate to our design.

Background on HCCI Engine Operation

The HCCI engine combines the spark ignition (SI) of a traditional gasoline engine and the combustion ignition used in diesel engines.

A gasoline engine uses a discharge of electrical current through a spark plug to ignite a gasoline and air mixture. Fuel is injected into the intake manifold ports leading to the combustion chamber and is mixed with air as it flows into the combustion chamber. During the first stroke, the piston draws this fuel/air mixture into the cylinder as it moves toward bottom dead center (BDC). During the next stroke the piston compresses the mixture with both valves closed. Depending on the engine design some time after the piston reaches top dead center (TDC) the spark is delivered from the ignition system and expansion occurs. The combustion process raises temperatures and pressures inside the cylinder causing the piston to be forced downward producing usable work. The exhaust is then expelled as the piston moves from BDC to TDC thus completing the four stroke cycle. Auto ignition of the mixture is prevented through compression ratio selection, fuel octane level, timing of the spark, and the valve events of the engine. Auto ignition in a gasoline engine is often referred to as knock and can quickly destroy an engine because of the sudden, violent pressure increase associated with it [1].

A diesel engine operates on many of the same principles as the gasoline four stroke engine, however compression is used to ignite the fuel/air mixture. The fuel is injected directly into the cylinder just after compression as opposed to being injected into the intake manifold during the intake stroke. The fuel/air mixture is then compressed to such high pressures that it auto ignites. Increased cylinder pressures resulting from the auto ignition necessitates that diesel engines are more robust in their design than there gasoline counterparts. The injection of fuel directly into the cylinder removes the need to throttle the engine to control speed. The pumping losses associated with the throttle are now gone thus increasing the output of the engine [2].

To obtain the best of both the SI gasoline engine and the compression ignition diesel engine, they are combined in the HCCI engine. The HCCI engine at room temperature starts by SI combustion and as engine temperature increases, the engine controller increases intake manifold pressure for the HCCI combustion and then turns off the SI operation. The combustion process is controlled by adjusting many different parameters with the engine controller such as fuel rate, boost level, exhaust gas recirculation (EGR), intake charge, and cooling temp. A given speed and load has an optimized combination of all these parameters to ensure the best efficiency and operation. This HCCI process offers greater thermal efficiencies and lower NO_x emissions than other engines but has increased HC and CO emissions [3].

Existing Valves
There are several existing valves that we have discovered in our research. Butterfly valves, ball valves, and valves as simple as a sliding plate are the most common. All of these valves provide viable solutions for the exhaust bypass valve design, but none have really been proven to function at the required pressures or temperatures.

Butterfly Valve
The butterfly valve is a reasonable solution to an exhaust bypass valve, and it is commonly used to control the air inlet in gasoline engines as part of the throttle body. The basic idea behind this valve is that it uses a circular plate that rotates about an axis through its center, rotating from fully closed to fully open. This could be applied to an exhaust bypass valve, and it has been used in exhaust valves already in production, which does in fact show that it can withstand the exhaust temperatures. Examples of these butterfly valves are shown below in Figure 2 and Figure 3.

Figure 2: Exhaust Cut-Out Valve

Figure 3: Linear Actuated Butterfly Valve

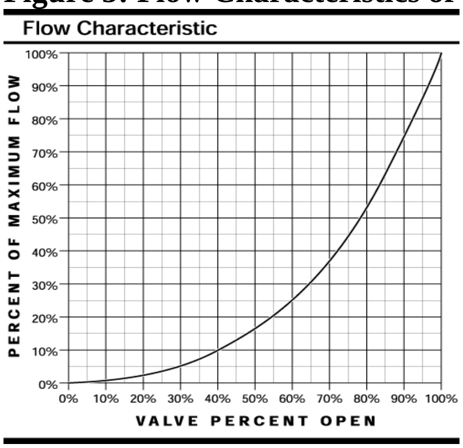
There are multiple downsides to these valves, including issues with creating an effective seal, as well as thermal expansion due to high temperatures. Because of the method in which the valve is opening, it is hard to create a seal that will prevent leaking when it is facing pressures upwards of 3 bars. The fact that the temperatures will exceed 350 °C is a problem also. In order to seal properly the parts require very tight tolerances, however they will show significant signs of thermal expansion, potentially hindering the precise motion needed to achieve the desired accuracy, and further complicating the design of a seal [4].

Ball Valve
Another possible solution for an exhaust bypass valve design is a ball valve. The design of a ball valve resembles that of a butterfly valve. It is comprised of an inlet and outlet path with a sphere blocking the path of the flow. The sphere has a hole cut through the center and as it is rotated, the hole begins to line up with the path of the fluid allowing flow. These valves are excellent for completely closing and stopping all flow through the valve however they are limited in the capability to directly control the flow in throttling applications. There is no direct application of this valve being used in an exhaust bypass system and no evidence that it could withstand the required temperatures. A diagram of a full port ball valve is shown below in Figure 4.

Figure 4: Full Port Ball Valve

A full port ball valve allows no restriction of the flow because the hole cut through the sphere is the same size as the inlet and outlet. Another type of ball valve, a V port ball valve utilizes a V shape to open and close. This allows the flow to be controlled more, as seen in Figure 5 below, which is essential to our exhaust bypass valve [5].

Figure 5: Flow Characteristics of a V Port Ball Valve



Patent Search
Following our evaluation of valves currently on the market, we did a patent search to find ideas that have been patented related to an exhaust bypass valve.

Patent number 20070204616 comprises of a twin stacked poppet valve design where both valves are controlled by the same actuator. This offers some control of the valve, but only in two stages, the first stage being partially open with a smaller poppet valve, and the second stage fully open with both valves engaged. This does not apply to our exhaust bypass valve because it does not offer enough proportional control.

Patent number 06675579 describes the entire twin turbo system for an HCCI engine. It describes an electrically controlled exhaust bypass valve used to divert flow from one turbo to the other. It does not go into explicit detail on the bypass valve, providing no real useful information in our design.

There are many other patents out there, more than can be listed, but all of which factor into our design process. These patents allow us to see the positives and negatives of each design but our exhaust bypass valve has a very unique set of requirements and must be designed entirely independent of any other valve currently on the market.

CUSTOMER REQUIREMENTS
Dr. Ruonan Sun of the EPA has provided us with several requirements, many of them specific engineering specifications, for the HCCI engine exhaust bypass valve. These requirements include:

- The valve must seal against a gage pressure of 3 bars.
- The valve must have proportional controlled opening for the first 20 ± 2% of the area.
- The valve must open quickly, 0 to 20% in 2 seconds.
- The valve must be able to operate in the temperatures of 350 to 550°C.
- The electric motor operating the valve must be kept below 125°C.
- The valve must exhibit minimal wear between moving parts.
- The valve must have a maximum opening area of 1.3 in^2.
- The valve would ideally be low cost, light weight, and have a small profile.

Although all of the customer requirements are important, many contradict each other (e.g. it is difficult for a valve to be proportionally controlled while opening quickly at the same time). Several tradeoffs will have to be made to complete an optimized functional design. For example, Dr. Sun and the EPA are much more concerned with a functional valve than with the final cost.

The final deliverables for the project include a detailed design report and preferably a working prototype, although due to time constraints a model would be acceptable (R. Sun, Personal Communications, September 13, 2007).

ENGINEERING SPECIFICATIONS

A set of engineering specifications was developed to encompass all of the customer requirements into our design of the exhaust bypass valve. We broke down our specifications into three main groups: the physical geometry of the valve, the operational characteristics of the valve, and the material considerations of the valve. The engineering specifications encompassed by the physical geometry of the valve include the overall size/shape of the valve, the valve, opening area, and the manufacturability of the valve. The type of valve, area proportionally opened, opening speed, maximum gage pressure, type of actuator, operating temperature, electronics' temperature, and lubrication will be included in the operational characteristics of the valve. Finally, the engineering specifications included in material considerations are the choice of materials for all components related to the valve and the durability of each component. When choosing appropriate materials for the exhaust bypass valve, the specific material properties will be considered to determine how they will function at these high pressures, temperatures, and repeated loading and unloading.

Quality Function Development (QFD)

To relate the customer specifications to our engineering requirements, a Quality Function Development (QFD) diagram was created. See Appendix A for the complete QFD.

We began creating the QFD by listing the customer requirements provided by Dr. Sun and assigning a weight to each one on a scale of 1 to 10, 10 being very important and 1 being the least. For example, the customer requirements of sealing against high pressures, a proportional controlled opening, and quick opening all received a weight of 10 while the requirement of light weight received our lowest weight of 3. Next, we listed our engineering specifications, their units, and their target values and rated the relationship of each spec to each customer requirement, 1 being a small relationship, 3 being a medium relationship, and 9 being a very strong relationship. For example, the specification of the type of valve was strongly related to the requirement of sealing pressure, so their relationship was assigned a 9. The specification of the opening speed has a very small relationship with the requirement of sealing against high pressures, so their relationship was assigned a 1. The numerical value for each relationship was then multiplied by the weight for each customer requirement and summed for the entire specification. This allows us to see which specifications are more important in the design of the exhaust bypass valve. Our higher rated and most important engineering specifications include the type of valve, area proportionally opened, opening speed, type of actuator, and material choice. Some of the lower rated engineering specifications and less important include the manufacturability and overall size/shape of the valve. Finally each engineering specification was then assigned a relationship to the others. A ++ indicates a strong positive relationship, + a medium positive, - a medium negative, and - - a strong negative. For example, the material choice and operating temperatures had a strong positive relationship while proportionality and speed had a strong negative relationship.

The QFD provides a valuable tool for evaluating the most important aspects of our design of the exhaust bypass valve although it is not always perfect and our team still needs to make our own decisions on each customer requirement and specification.

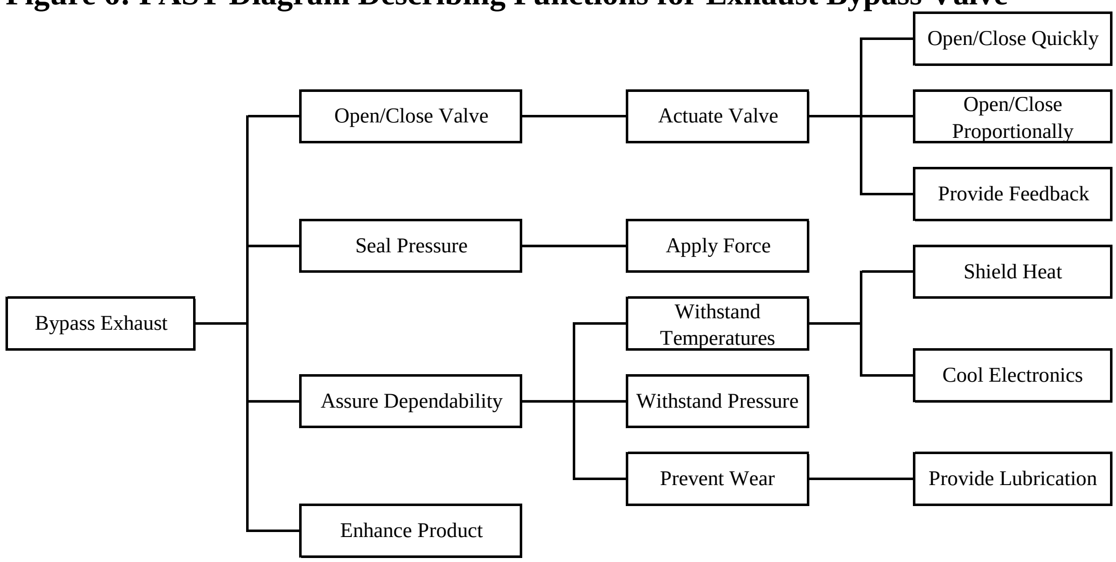
CONCEPT GENERATION

Multiple steps went into the generation of exhaust bypass valve concepts to meet the EPA's customer requirements and the engineering specifications. The steps we went through to develop our design concepts are documented in the sections below.

Functional Analysis System Technique (FAST)

The concept generation process started with our team listing and relating the functions of our exhaust bypass valve, also known as the Functional Analysis System Technique (FAST). These functions come from looking at the customer requirements, engineering specifications, and the QFD diagram located in Appendix A. The main function of the exhaust bypass valve is to bypass exhaust gas. The supporting functions of the main function of the exhaust bypass valve include opening/closing the valve, seal against pressure, assure dependability, and enhance the product. Each of these supporting functions has a set of supporting functions. The main function and supporting functions are related in the FAST diagram in Figure 6 below.

Figure 6: FAST Diagram Describing Functions for Exhaust Bypass Valve



Morphological Method

To begin generating some concepts and sketches, our team used the morphological method. For each of the functions in the FAST diagram on the previous page, many different concepts were brainstormed and listed in the Morphological Chart seen in Figure 7 on page 10. For the open/close valve function, there were 6 different concepts that could accomplish this including a poppet valve, ball valve, butterfly valve, gate, linear slide, or a rotational slide. This was repeated for each function. By listing the different concepts for each function horizontally you can look vertically and combine different concepts into an exhaust bypass valve concept.

Figure 7: Morphological Chart for Concept Generation

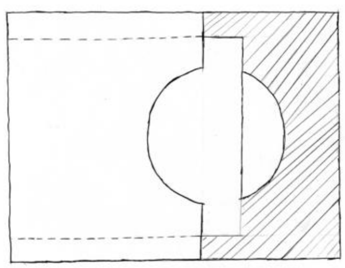
Function	Concept 1	Concept 2	Concept 3	Concept 4	Concept 5	Concept 6
Open/Close Valve	Poppet	Ball	Butterfly	Gate	Linear Slide	Rotational Slide
Actuate Valve	Linear Actuator	Boost Pressure	Spring Actuated	Fluid Actuated	Servo-Motor	
Seal Pressure	Valve Seat	Spring	Locking	Friction		
Withstand Temperatures	Air Cooling	Fluid Cooling	Fins	Thermal Electric	Heat Shields	
Withstand Pressure	Seal Type	Fluid Force	Spring Force			
Prevent Wear	Oil	Other Lubricant				

Exhaust Bypass Valve Concept Sketches

After completing the morphological chart and deciding which concepts to combine, we developed some concept sketches seen below and on the following pages.

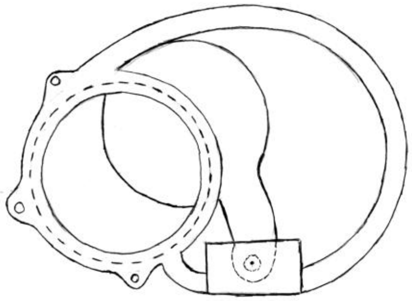
Design Concept #1 is a linear slide valve seen in Figure 8 below. A linear actuator is attached to a sliding rectangular plate that is orientated perpendicular to the flow of exhaust gas. The plate is moved to expose more area for the gas to flow through.

Figure 8: Sketch of Linear Slide Exhaust Bypass Valve



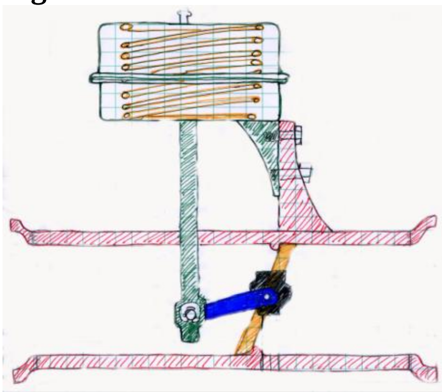
Design Concept #2 is a rotational slide valve seen in Figure 9 below. This concept is similar to Design Concept #1, the only difference being rotational motion and a circular plate. The orientation of the circular plate and the exposed area changes as it pivots. The rotational motion could be controlled with a servo motor either attached directly to the pivot or through a transmission.

Figure 9: Sketch of Rotational Slide Exhaust Bypass Valve



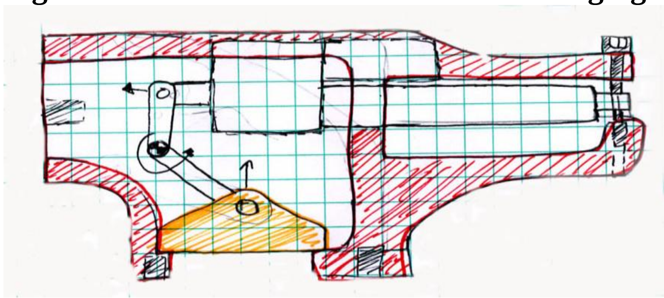
Design Concept #3 is a pressure activated butterfly valve seen in Figure 10 below. The valve is actuated using manifold pressure pushing on a flexible diaphragm. The end of this actuator is attached with a pivoting joint to a lever arm that is secured to the butterfly shaft causing the butterfly to rotate. The rotation of the plate allows exhaust gas to flow through the tubing.

Figure 10: Sketch of Pressure Activated Butterfly Exhaust Bypass Valve



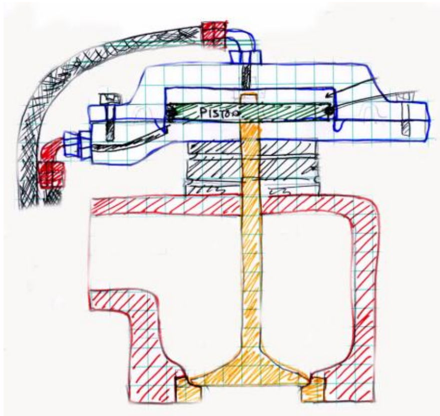
Design Concept #4 is a linear actuated swinging plate valve seen in Figure 11 below. This concept uses a linear actuator coupled with a linkage that when activated "swings" the plate up, removing the plate obstructing the flow, and allowing the exhaust gas to pass.

Figure 11: Sketch of Linear Actuated Swinging Plate Exhaust Bypass Valve



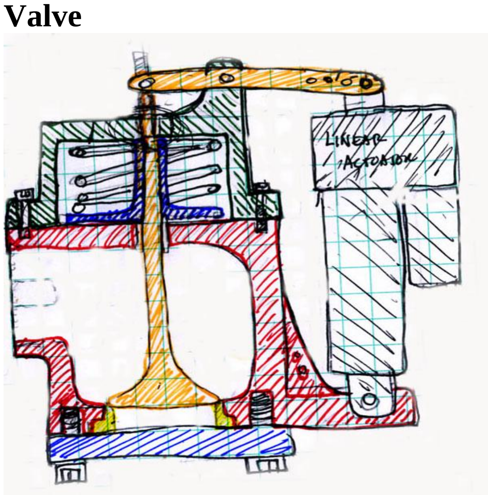
Design Concept #5 is a fluid diaphragm actuated poppet valve seen in Figure 12 on page 12. Using engine oil pressure, combined with a piston cylinder design, the piston is activated, moving the poppet valve up or down, providing the desired area for the flow. Altering of fluid pressures above and below the piston would displace the piston in the cylinder causing smooth movement of the valve off its seat.

Figure 12: Sketch of Fluid Diaphragm Actuated Poppet Exhaust Bypass Valve



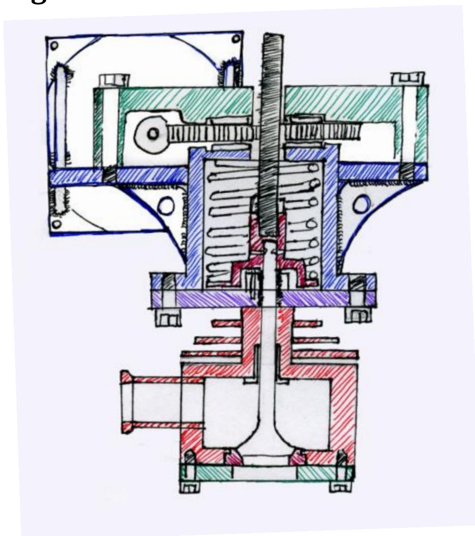
Design Concept #6 is a linear actuated rocker arm poppet valve seen in Figure 13 below. The linear actuated poppet valve functions along the same line as Design Concept #5; however the linear actuator eliminates the need for oil pressure and uses a linkage to move the poppet valve. The push or pull force from the actuator is translated through the rocker arm to act directly on the valve stem. The spring provides the necessary preload to keep the valve sealed and allows better proportional control.

Figure 13: Sketch of Linear Actuated Rocker Arm Poppet Exhaust Bypass Valve



Design Concept #7 is a motor controlled poppet valve seen in Figure 14 on page 13. This design uses a poppet valve that has the same orientation as Design Concept #6, however attached to the spring retainer is a motor controlled screw jack. The rotation of the motor drives a worm gear that reduces the rotational speed. The gear that meshes with the worm is threaded and its rotation casues the stationary screw to thread up and down. The force provided by the threaded screw combined with the spring force determines the position of the valve relative to the amount of exhaust pressure.

Figure 14: Sketch of Motor Controlled Poppet Exhaust Bypass Valve



CONCEPT EVALUATION AND SELECTION

Many of the design concepts meet a majority of the design specifications so it was necessary to break them down and look into their individual advantages, disadvantages, and problem areas. Some of the specicfic areas of interest in the concept evaluation were the simplicity, cost, ease of production, actual performance, and potential complications of each valve. After examining all the concepts, the valves that met all the design requirements, presented the best performance, and remained simple in design were choosen. A brief discussion of each of the concepts follows.

Concept 1: Linear Slide Exhaust Bypass Valve

The linear slide exhaust bypass valve is ideal because it is a very simple design. The linear motion of the plate is easy to control and the entire valve can be manufactured easily. This valve requires tight tolerances to keep the desired seal but the sliding motion has the potential to cause excess wear which in turn could cause leaking. There is no good method to ensure adequate lubrication to combat the wear issue with an effective zero loss seal.

Concept 2: Rotational Slide Exhaust Bypass Valve

Like Concept #1, the rotational slide exhaust bypass valve is a simple design. The rotational motion of the plate is not as simple as the linear motion. The rotational motion allows for very accurate movement, however, the crescent shape of the exposed area would makes proportional control difficult because the change in area per change in rotational angel in not constant. Leakage past the valve is also a concern as some type of frictional seal would be necessary. This valve does satisfy some of the design requirements but like Concept #1 complete sealing would be an issue.

Concept 3: Pressure Activated Butterfly Exhaust Bypass Valve
The pressure activated butterfly valve concept is currently in use as a newly released automobile turbocharger bypass. The simple design and activation type allow for easy implementation. This valve would operate at lower pressures than our design requirements. The butterfly fly will not seal under the higher pressures of our system, as well as not effectively control the opening area. Rotating the plate only a small amount will expose a large opening and imbalance in flow. This could result in unbalanced forces on the butterfly blade resulting in further instability. Proportional control of this valve and its inability to operate under the design specification are reasons why this valve is not optimal for our design.

Concept 4: Linear Actuated Swinging Plate Exhaust Bypass Valve
The main advantage of this design is it has been used to bypass exhaust in turbocharger applications for decades. The rotation of the pivot presses the plate against and opening with no seal, only surface contact as the seal. In conditions where some leaking is allowed the valves are a very simple cost effective choice however, the design specifications mandate that there is no leakage.

Concept 5: Fluid Diaphragm Actuated Poppet Exhaust Bypass Valve
The idea of using a poppet valve to control turbocharger speed is used by many turbocharger manufacturers. The advantage of this design is a positive seal against the valve seat. Internal combustion engines use this concept and seal to pressures far exceeding our design requirement. The complication in this concept lies in the activation. Using fluid in a piston allows for large forces to be applied with low pressures of fluid application. The control of the fluid flow and dynamics is very complex. A valve body would be needed to control the fluid flow and although it was originally thought engine oil could be used for operation, a stand-alone fluid system would most likely be necessary adding further complexity.

Concept 6: Linear Actuated Rocker Arm Poppet Exhaust Bypass Valve
Further improving on the previous concept the linear activated poppet valve seals well and increases the controllabilty of the valve position. The linkage of a rocker arm does impose some problems with wear. Tolerance of each of the pivot points present a problem for controlling very fine movements in that the addition of tolerances for each joint would result in uncertainty of valve position.

Concept 7: Motor Controlled Poppet Exhaust Bypass Valve
By incorporating a transmission as the control, we can convert rotational motion of a motor to linear motion of the valve. This concept is well suited for electronic controls currently used in automobile applications. The transmission of the motors torque into a linear force is complicated. Friction and wear could be a concern if the transmssion were not designed correctly. Prevention of the rotation of the valve itself may also be difficult to accomplish without complexity. The very fine control of position enabled by using this worm gear driven screw and ease of integration into existing engine electronic controls elavate this design above others.

Concept Selection

To select the final design concept for the exhaust bypass valve, a Pugh Chart was used to rate all of our designs to the specific customer requirements provided by Dr. Sun of the EPA. The customer requirements and weights from our QFD diagram were first added to the Pugh Chart. Each of our design concepts, labeled and shown in the previous pages, were given a rating against each customer specification, 0 meaning that the concept does not meet the customer requirement, 3 meaning the concept barely meets the customer requirement, 6 meaning that the concept somewhat meets the customer requirement, and 9 meaning that the concept definitely meets or exceeds the customer requirement. For example, Concept # 1, the linear slide exhaust bypass valve, received a 9 for opening quickly because the concept definitely meets this requirement but received a 0 for minimal wear because this concept does not meet this customer requirement. The completed Pugh Chart can be seen in Figure 15 below.

Figure 15: Pugh Chart for Exhaust Bypass Valve Concepts

		Concept #						
Customer Requirement	**Weight**	**1**	**2**	**3**	**4**	**5**	**6**	**7**
Seal Against High Pressures	10	6	3	3	6	9	9	9
Proportional Controlled Opening	10	6	3	0	0	3	6	9
Open Quickly	10	9	9	9	3	3	9	9
Operate in High Temperatures	8	3	3	9	9	9	9	9
Minimal Wear	7	0	3	6	6	9	9	6
Low Cost	5	9	6	3	3	0	3	3
Light Weight	3	9	9	6	6	3	3	3
Small Profile	6	6	3	3	6	3	3	3
	Total	342	270	285	273	312	417	426
	Rating	3	7	5	6	4	2	1

After rating all concepts against each customer requirements a total was tallied for each concept to determine which one would meet the overall customer requirements the best. The Pugh Chart evaluation revealed that our top design is Concept #7, the motor controlled poppet exhaust bypass valve. Coming in a very close second was Concept #6, the linear actuated rocker arm poppet exhaust bypass valve. The Pugh Chart is by no means the final decision in the design selection process, our team must also decide if the concept is feasible to completely design and manufacture within our resources.

Our team also brought all design concepts to Dr. Sun to receive his input on which design would meet his requirements and function at the highest level. Dr. Sun expressed concern over the linkage in Concept #6, pointing out that the slightest movement in the linkage could throw off the very tight tolerances required for the control of the exhaust bypass valve. He also recommended a screw type design similar to Concept #7 (R. Sun, Personal Communications, October 18, 2007).

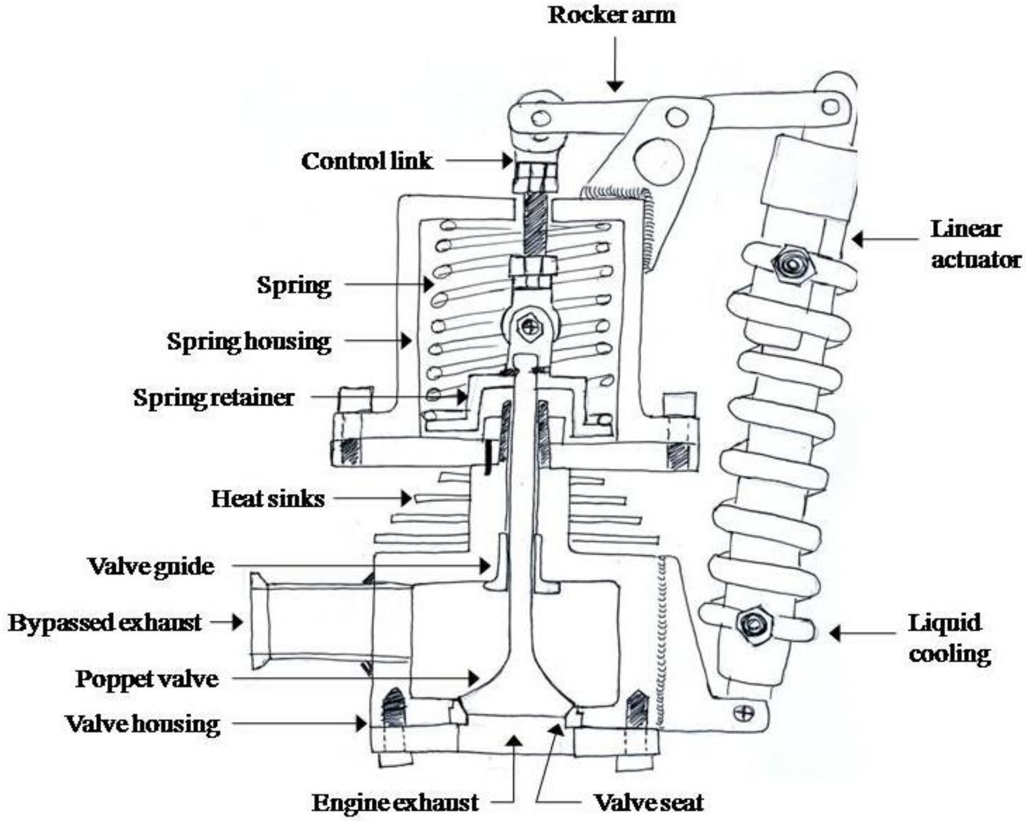
After taking into account the Pugh Chart evaluation and Dr. Sun's feedback, at this moment our team has Concept #7, the motor controlled poppet exhaust bypass valve, as our 1a choice and Concept #6, the linear actuated rocker arm poppet exhaust bypass valve, as our 1b choice. Before the meeting with Dr. Sun we were leaning towards Concept #6 and had determined that this design was well within our manufacturing capabilities and resources. After the meeting, we are now leaning towards Concept #7 but have concerns over the complexity of the design and if we will be able to successfully manufacture a prototype. Both designs are very similar, only differing in the type of actuation, so within the next few days a final design will be decided upon.

SELECTED CONCEPTS

Our design has been narrowed down into two concepts; both are very similar in function and design, the main difference being the method of actuation for the poppet valve. The two final concepts we chose are shown in Figures 16 on page 17 and Figure 17 on page 18. In each figure, the different components of each design are labeled, giving an accurate description of how the entire valve is assembled.

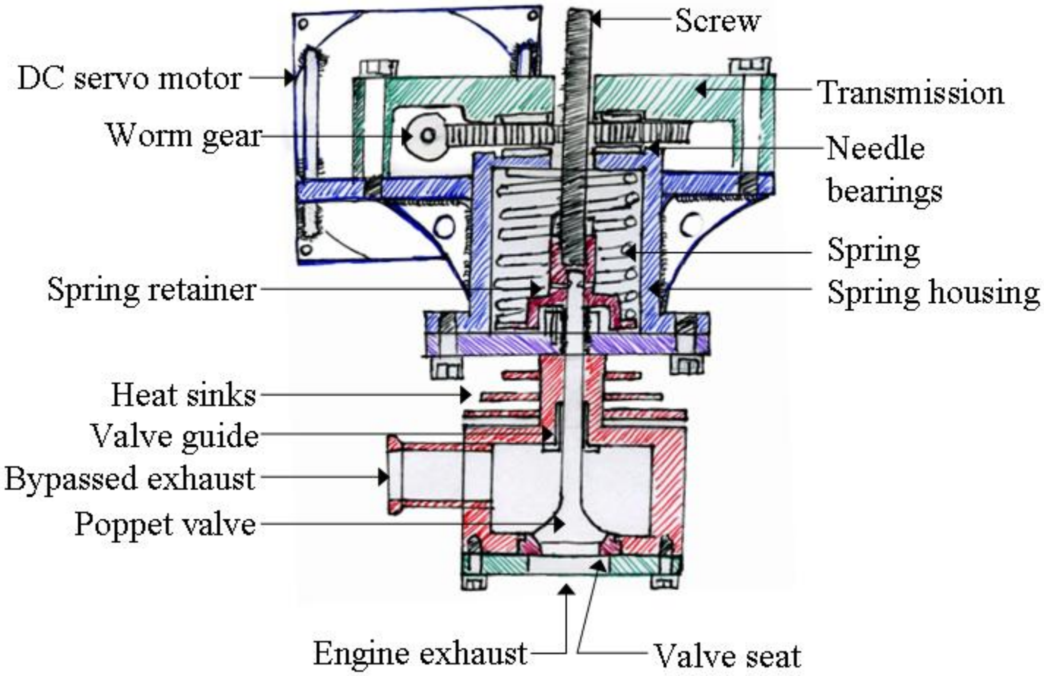
The linear actuated rocker arm poppet exhaust bypass valve is shown in Figure 16 on page 17. There are several components that are going to be purchased, mainly the valve, valve seat, spring, and the linear actuator. The valve seat is designed for the specific poppet valve, allowing us to achieve the best possible seal. The spring is designed specifically to resist high temperatures so that the actual spring constant and the force it applies is not affected by thermal expansion. The linear actuator is only required to produce small linear movements to open the valve, and is capable of being proportionally controlled by the electrical input to achieve this goal. The remainder of the components will be machined by hand. The housing will be manufactured out of steel, with fins between the housing and the spring housing to help dissipate the heat. As you can see in Figure 16, once these components are assembled, the linear actuator uses a rocker arm to actuate the valve that is held in place by the spring.

Figure 16: Detailed Linear Actuated Rocker Arm Poppet Exhaust Bypass Valve



Our second choice, the motor controlled poppet exhaust bypass valve, is shown in Figure 17 on page 18. The main components are identical to those as the linear actuated valve. The difference with this valve is that is uses a simple transimission driven by a DC servo motor and a worm gear to actuate a screw jack that is attached to the valve itself. This valve still has a spring pressed against the valve using a spring retainer in order to fight backlash against the motor, providing more accurate control of the movement. As with the linearly actuated poppet valve, the housing, spring housing, and the other main components will be machined by hand, and the same components as before are to be purchased. The gears and threaded rod will be purchased if available to our required specifications, otherwise they will be machined by hand.

Figure 17: Detailed Motor Controlled Poppet Exhaust Bypass Valve



FINAL CONCEPT SELECTION

After carefully reviewing the manufacturability, cost, and reliability of both the linear actuated rocker arm poppet valve and the motor controlled poppet valve our team determined that the motor controlled poppet valve was ideal for our exhaust bypass valve application. We felt that this valve would meet all requirements and would be able to be manufactured with our available resources.

In order the ease the manufacturability, research was done into the different types of gears and transmission systems that were readily available for purchase. Our research revealed that purchasing all of our required gears would be very expensive and far outside our prototype budget but we could purchase a screw jack that included all required gears and the screw in a pre-made component.

Dr. Sun later informed us that he and the EPA would be providing us with a very precise DC motor, specifically a Maxon RE 40 DC motor, to cut our prototype costs. A drawing and all of the motors specifications can be seen in Appendix C. We were informed that this motor should handle any of our needs for our valve and if the engineering analysis revealed that a different motor was required we could still be use this to show the functionality of our prototype.

We met with our sponsor after completing all of this research to get his input on our final design choice. He expressed concern with the proportional control of the screw jack and recommended we abandon the transmission system altogether and mount the motor vertically with the screw directly coupled between the motor and poppet. The Maxon DC motor was also given to us as this time which let us further visualize our final design (R. Sun, Personal Communications, November 1, 2007).

Taking into account all of our research and input from Dr. Sun we have chosen for a final design a modified version of the motor controlled poppet valve presented in the previous pages of this report. The transmission system has been abandoned for a vertical motor controlling a screw that is coupled to the poppet. We have also decided to include a liquid cooling system around the heat sinks to further dissipate heat and protect our controls of the exhaust bypass valve.

In this final design an existing wastegate valve would be purchased and modified to connect the DC motor and screw to the valve. The manufacturing of the connecting components will be completed in the student machine shop.

The following sections of the report will provide more detailed information on our final design for the motor controlled poppet exhaust bypass valve.

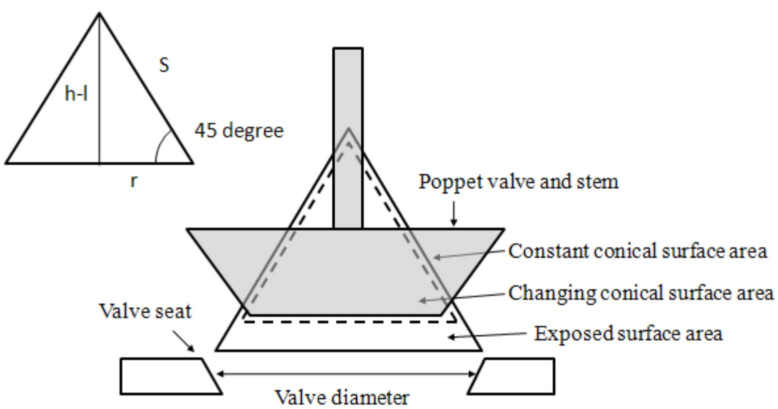
ENGINEERING ANALYSIS

Detailed engineering analysis was necessary to ensure that our design for an exhaust bypass valve met all customer requirements and engineering specifications. The following sections detail the engineering analysis that went into the design of our motor controlled exhaust bypass valve.

Area Analysis

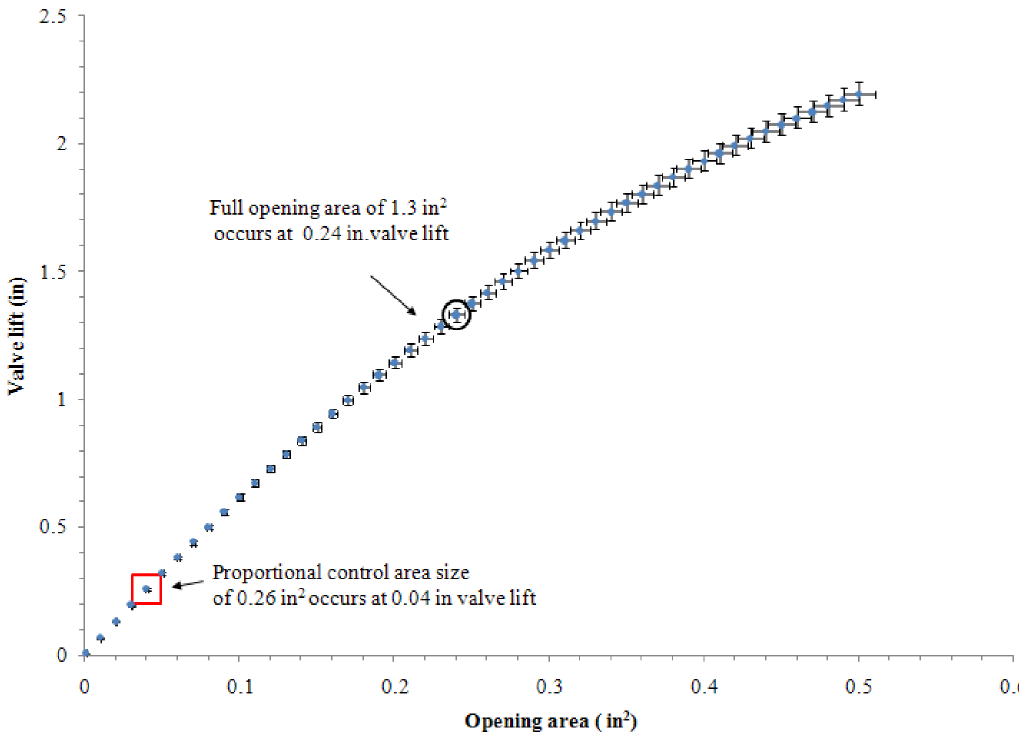
In order to meet the proportionality and speed requirements given by Dr. Sun, an area analysis had to be performed for our valve. The valve seat angle and the valve shape were critical to determining the shape of the exposed area when the valve was displaced. The diameter of the valve was chosen so the valve area was approximately equal to the 1.3 in^2 area requirement. The valve we are using has a valve seat angle of 45 degrees. As the valve opens a small circular area is exposed. This area is shaped like an annulus of a cone as seen in Figure 18 below. To calculate the area at each individual valve position, we took the difference between the fully exposed cone and the smaller cone. In this case the changing cone height is the constant height cone minus the valve lift. Because the valve angle is 45 degrees, the valve radius of the changing cone is also the constant height cone minus the valve height.

Figure 18: Conical shape Approximation for Flow Area of Bypass Valve



The relationship between valve lift and exposed area for the flow of exhaust gases is shown in Figure 19 below. As the valve lift increases, the area available for the exhaust to pass increases non-linearly. The valve lift required to reach the specified area for the proportional control is 0.0402”. The magnitude of the lift here necessitates the use of the fine threaded rod to activate the valve as opposed to an acme screw or rod with course thread pitch. Small displacements in the valve result in large changes in flow area under these conditions. The change in flow area is not as sensitive to changes in valve lift when nearing the maximum flow area of 1.3 in^2. Proportionally controlling the flow is directly related to how precisely we can move the valve at these low lifts. In order to control the area within the requested uncertainty we must determine how the tolerance in machined components and precision of movement will affect the area exposed. Errors of ± 2% are shown on the graph.

Figure 19: Exposed Flow Area Increases Non-Linearly with Vertical Valve Lift



Now that the relationship between the valve lift and the valve opening area is known, how the motor controls the opening area can be analyzed. The design requires that the valve be able to porportionally control and area opening of 20 ± 2% of the maximum area of 1.3 in^2 within 2 seconds. To determine whether or not this could be accomplished with the existing motor, we needed to calculate how much lift the valve obtains with each revolution of the screw. Once this was determined, the loading conditions and torque characteristics of the motor were used to determine the speed required to reach the 20% opening area of 0.26 in^2. Because of the increasing force given by the spring upon compression, the response of the system as lift increases goes down. The balance for the spring force and the fluid pressure force of the hot exhaust gases allows the motor to easily control the low lift portion of the valve movement with significant precision and accuracy. This follows the customer requirement imposed by the EPA that only the first 20% of the area needs to be proportionally controlled.

Lift Force Analysis

The poppet of the valve and how it related to the exhaust pressure needed to be analyzed to determine the lift force required for our exhaust bypass valve. The diameter of the valve is 1.496" (*0.038 m*) and will operate under an exhaust pressure of 45 psi (*310300 Pa*) therefore it will experience a lift force of 79 lbs (*352 N*). Refer to Appendix D for free body diagrams, equations, and analysis for the lift force.

Spring Analysis

To close the spring-screw housing the spring is compressed 0.375" (*0.010 m*). To ensure the valve is closed, the spring force must equal that of the lift force. From the equation of a spring force and the free body diagram in Appendix E, the constant of the spring can be derived. The spring constant required to keep the valve closed under an exhaust pressure of 45 psi (*310300 Pa*) is 210.95 lbs/in (*36946 N/m*).

Torque Analysis

To determine the motor torque required to lift the valve, a screw analysis was performed. To reach 20% of an opening area of 1.3in^2 ($8.4 \times 10^{-4}\ m^2$), the valve must be lifted 0.040" (*0.001 m*) which implies that the spring must be compressed by the same amount. To fully open to the area of 1.3in^2 ($8.4 \times 10^{-4}\ m^2$), the valve must be lifted 0.231" (*0.006 m*).

The net forces applied on the screw to lift the valve 20-100% of an opening area of 1.3in^2 are 2.46 lbs (*10.9 N*) to 14.2 lbs (*63 N*), respectively.

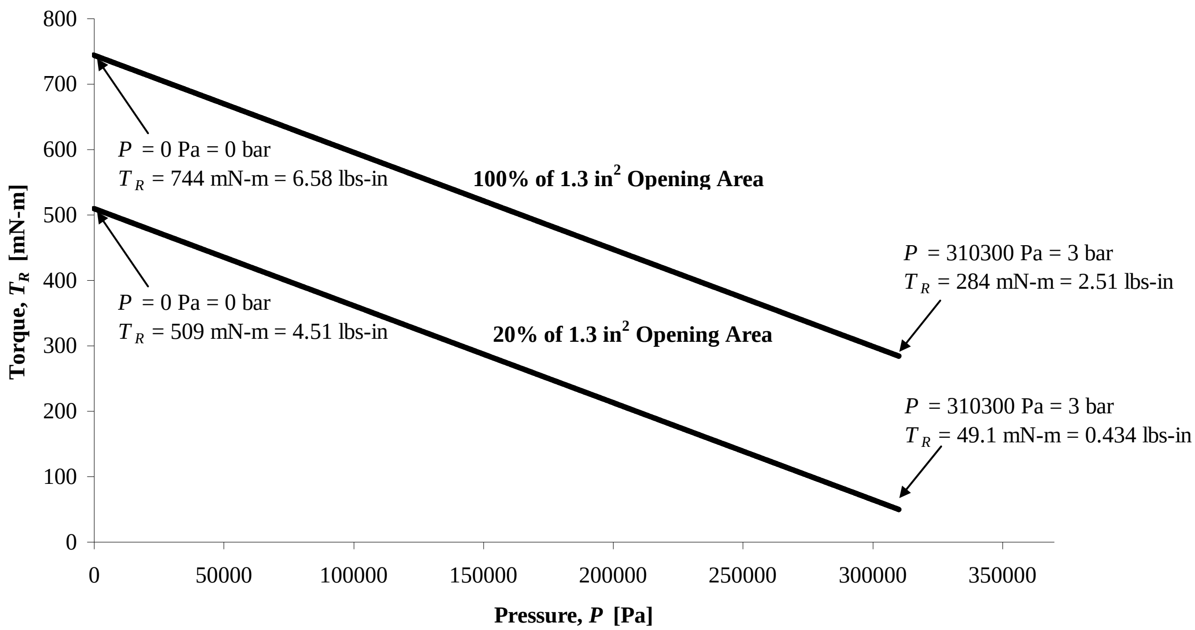
To compute the required force and torque needed to raise the valve, an analysis for an unrolled thread for one revolution of the screw was performed. The force applied on an unrolled thread is the same net force applied on the screw [7].

The screw's material is Stainless Steel and the material of the block the screw is threaded into is Kevlar/Nylon. The coefficients of friction for Nylon on Nylon for dry static friction are 0.25 to 0.15 [8]. The coefficients of friction between the screw and Kevlar/Nylon are assumed to be less. To ensure our system operates under various conditions ranging from dry to lubricated, we assume the friction to be that of Nylon on Nylon under dry static friction. The forces and torques required to raise the valve are summarized in the tables in Appendix F for various conditions.

The torque required to raise the valve for 20% of the maximum 1.3in^2 opening area was found to be 0.434 lbs-in (*49.1 mN-m*) without accounting for any pressure drop when the valve is opened. The torque required to raise the valve 100% of the maximum 1.3in^2 opening area was found to be 2.51 lbs-in (*284 mN-m*) without accounting for any pressure drop.

As the valve is opened the pressure acting on the poppet will drop, causing the required torque to increase. Figure 20 below shows how torque increases as exhaust pressure at the poppet drops for the 20% and 100% of the maximum opening area. We were not able to compute the actual pressure drop so we varied the pressure from a maximum of 3 bars to 0. As this pressure varies, the force acting on the poppet of the valve also changes. The change in exhaust pressure and force will result in a different torque required by the motor. At a 0 bar pressure, the maximum torque increases to 6.58 lbs-in (*744 mN-m*).

Figure 20: As Pressure Drops the Required Torque for 20% or 100% of 1.3 in^2 Maximum Open Area Increases.



Realistically, the pressure will not drop all the way to 0 bar, meaning the required torque will be less than the maximum value of 6.58 lbs-in. The provided Maxon DC motor provides 0.84 lbs-in (*94.9 mN-m*) of torque which is below the required torque meaning a gear reduction of approximately 8:1 is needed for the maximum torque required by the motor.

The complete analysis and free body diagrams of the forces acting on the screw can be seen in Appendix F.

Speed Analysis

Included in the customer requirements was that the valve needs to proportioanlly control the first 20% or area with and accuracy of ± 2% within 2 seconds. The time the sytem needs to respond is critical to safely controlling the switch between the two turbochargers without overspeeding either one. Using the 24 thread per inch screw directly coupled to the motors output, the screw only needs to turn once to achieve the required lift to reach the 20% flow area. The encoder that is currently on our motor performs 64 counts per turn. This means that at 64 locations in one revolution the angular position the motors output is at can be determined. At this resolution of 64 turns per revolution we would have an error in valve lift on the magnitude of 10E-3” from the encoder. The tolerance in the thread also has an error that we must account for in the valve lift, however it is also on the magnitude of 10E-2”. With any electric motor there will be a time constant and a transition time until steady state, however with our design requiring only small angular changes to produce the correct changes in valve position, a steady state rpm will not be reached. It is important for us to incorporate the encoder and a position contoller. The position controller will not be purchased nor will it be incorported into our prototype, however it could successfully be implemented by the EPA on future work with this design to operate the motor. These position controller are available from Maxon Motors USA. They allow full rotational and speed control of the motor. Because the motor could be precisely controlled with the encoder and position controller, the mechanical tolerance and threaded connection allow for positive movement of the valve.

To determine the time to reach the one rotation of the screw we computed the moment of inertia of the rotating parts such as the coupler and the screw. Noting that torque is related to angular acceleration through moment of inertia, we were able to compute the required angular acceleration for the maximum required torque of 6.58 lbs-in (*744 mN-m*). Once we had the angular acceleration, we were able to compute the angular speed which allowed us to determine the time required to reach 20% of the maximum opening area. This time is on the order of 0.005 seconds which is well below our costumer requirement of 2 seconds. The complete speed analysis can be found in Appendix G.

Heat Transfer Analysis

In order to determine whether or not our design would be able to satisfy the temperature requirements imposed by Dr. Sun, we needed to perform heat transfer analysis of our valve and its components. The complete heat transfer analysis, complex geometry, and associated heat flux and temperatures can be seen in Appendix H. The critical dimensions for analyzing the individual types of heat transfer are also shown. Because the heat transfer calculations we performed were not crucial to satisfying our design requirements, we simplified our heat transfer model.

An order of magnitude calculation was performed to allow us to estimate the amount of heat transfer that would be needed to cool the valves electrical components to the design specification. The heat transfer required in an aluminum cylinder block similar in geometry to our heat sink made of aluminum was found using the equation in Appendix G. The temperatures used were the maximum operating temperature of the exhaust gases (550°C) and the required electrical component operating temperature specified by Dr. Sun (125°C). The thermal conductivity for aluminum alloy was used along with the external dimensions of the heat sink modeled as a cylinder. The magnitude of the heat transfer to achieve the desired temperature drop was 60 kW.

The heat transfer model used to calculate the heat transfer in the water coolant passages was as a single bounded fluid heat exchanger is also shown in Appendix H. Because of the complex geometry of the coolant passage, we also needed to make simplifications to the mathematical model. We chose to model the flow as that through a cylindrical tube with a diameter equal to that of the average side length of our passage. We needed to calculate the Nusselt number for the flow through the heat exchanger. To get this number we first looked up the Prandtl number and calculated the Reynolds number. The flow equation required that we know the fluid velocity in the passage, the kinematic viscosity and the diameter. This velocity was approximated using a percentage volume flow rate of engine coolant and water as well as the area of flow of the simplified model. The kinematic viscosity was found taking an average of glycol and water at an operating temperature of 80º C. Our Reynolds number was found to be 3940. In this geometry, any Reynolds number over 2300 is considered to be turbulent flow. It is required to know the type of flow in order to calculate the Nusselt number which is found using the following equation. Again we used and average value for the Prandtl number. Our Nusselt number was found to be 292.7.

Next we found the number of transfer units, NTU. This number correlates to the ratio of convection resistance to conduction resistance in the heat exchanger and helps to determine the heat exchanger effectiveness. The area used for finding the NTU was the inner area of a tube of with the averaged diameter and a length equal to that of the circumference of the heat sink. The thermal conductivity and specific heat will be the averaged values for water and glycol at the engines operating temperature. The mass flow rate will be found using the coolant density and the volume flow rate. Using the NTU we can get the effectiveness of the heat exchanger ε_{he}. The effectiveness was found to be 0.554. This effectiveness relates how well the heat exchanger transfers the heat energy from one source to another. Using the effectiveness, the prescribed temperatures, the material properties, and the simplified geometry, we can calculate the amount of heat that this simplified version of our heat exchanger would be able to transfer and compare it with the order of magnitude of original example of pure conduction through an aluminum cylinder.

Using the effectiveness, mass flow rate, and the coolants specific heat capacity we can calculate the average convection resistance. This convection resistance can be used to find the amount of heat transfer through our model using the following equation. The temperatures used in this equation are the far field temperature and the inlet temperature of the water coolant. Our heat transfer through our exhaust bypass valve was found to be 51 kW.

The amount of heat transfer needed to meet our design requirement was on the same order of magnitude as the heat transfer that a simplified version of our heat sink could remove. It should be noted that because the motor and encoder are mounted away from the heat sink that some radiation heat losses from the spring housing will occur and further improve the safety of our design. More detailed analysis of these calculations and removing some of the simplifying assumptions would be needed to ensure a successful design in reality. Adjustments to the coolant passage size and the flow rate through the coolant passage may be all that is necessary. Actually testing of the model would present the best results because of the complex shapes and geometries that are part of our final design.

Another important aspect of the system is that the coolant does not boil under these conditions. If boiling of the coolant were to occur, cavitations in the engines water pump could destroy it and then lead to overheating of the engine. To ensure that the water does not boil, we can look into the exit temperature of the water circulating through the coolant passage. The point at which the coolant boils is higher than that of water alone because the pressure that builds up in the cooling system. The boiling point of coolant can be increased safely to 250ºC. Using the conduction heat transfer equation we can solve for the exit temperature, in this case 103 ºC.

Because we cannot test our valve in the real application and get some experimental data on the temperatures throughout our system, we used mathematical models and engineering assumptions to estimate the functionality of our design. Dr. Sun emphasized that our design shows the functionality and that we give support to show that it would perform in the real operating conditions. This heat transfer analysis supports our design as a concept that is more than feasible in terms of satisfying the design requirement of keeping the electronic components below 125ºC when encountering exhaust temperatures on the order of 350 to 550 ºC [9]. Once again please refer to Appendix H for the complete heat transfer analysis.

Thermal Expansion Analysis

The thermal expansion of the individual components that make up our valve need to be considered in order to ensure that they do not introduce an error in position greater than the mechanical error and that the geometry and sealing of the valve remains intact. The operating temperatures are in the range of 350-500°C.

Valve Stem Growth

Thermal expansion of the valve results in a change in length that is greater than the mechanical accuracy of the design. This change in length is not large enough that a position change in the rotation of the motor could not correct. The growth in length of the valve for an increase in temperature from ambient temperatures before start up of the engine to operating temperatures can be found using the thermal expansion coefficient for stainless steel, the temperature change, and the geometry of the valve stem. The length that the stem would change during this change in temperature would be 0.71 mm. This amount of length change can be accounted for in the programming of the motor and the encoder. This number may seem larger at first but it is shown as a worst case scenario. This amount of growth would not occur instantaneously. The steady-state temperature of the valve would be relatively constant.

Other Component Growth

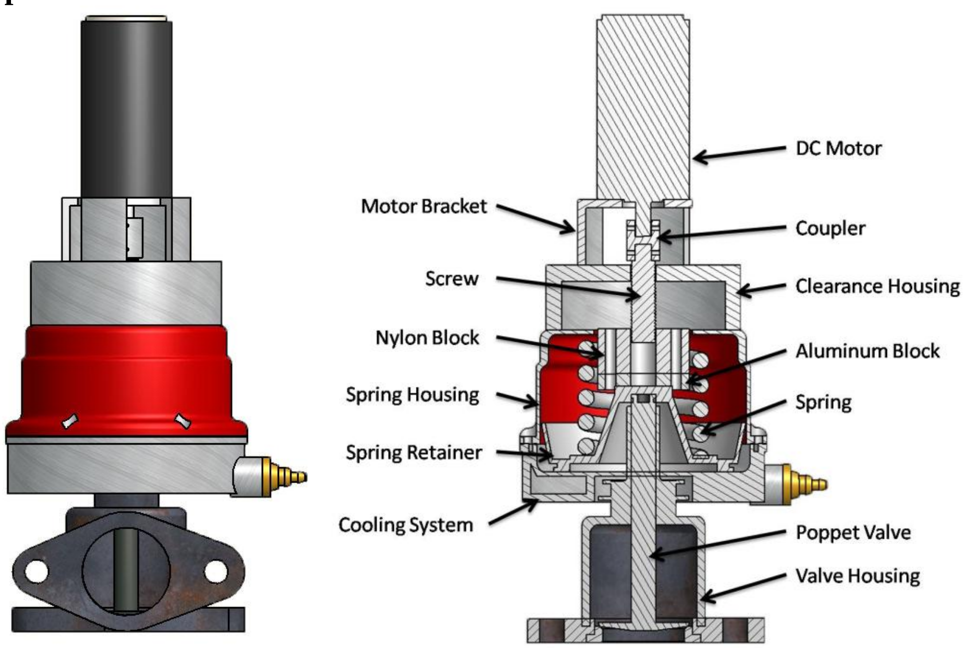
The valve seat and valve sealing surface are critical surfaces where thermal expansion could play a role in operation of the valve. Because we purchased a wastegate that was designed to operate under these design temperatures, we neglected to perform any calculations in this area. The manufacturer has allowed for tolerance in manufacturing and thermal expansion in these components. Using components with similar thermal expansion rates that are interacting in the valve is important for successful operation.

Manufacturing the control housings and all the internal components out of aluminum helped us ensure that there would not be an issue with dissimilar rates of expansion.

FINAL DESIGN

Our final exhaust bypass valve design was selected because it satisfied all customer requirements and engineering specifications. Design improvements from our preliminary motor controlled exhaust bypass valve concept allowed us to create the best possible design. A completed CAD model of the design can be seen below in Figure 21. The main portion of our design will be a purchased Tial 38mm wastegate. A complete technical drawing by Tial can be seen in Appendix I. This wastegate will include the lower housing for the valve, the poppet valve, the valve seat, the spring, spring retainer, and the upper and lower housings for the spring.

Figure 21: Final Design CAD Model of Motor Controlled Poppet Exhaust Bypass Valve



Modifications will be made to the purchased wastegate, along with manfucturing additional components to complete the final design for an exhaust bypass valve. Welded to the bottom of the spring housing is a cooling system comprised of a heat sink that will be machined out of aluminum with a canal in it. The heat sink will be attached to the engine radiator to circulate coolant around the valve housing to absorb large portions of the heat produced by the exhaust gases. The only other direct modification to the wastegate will be a hole drilled through the top of the housing to allow the threaded rod and nylon block to pass through as explained below.

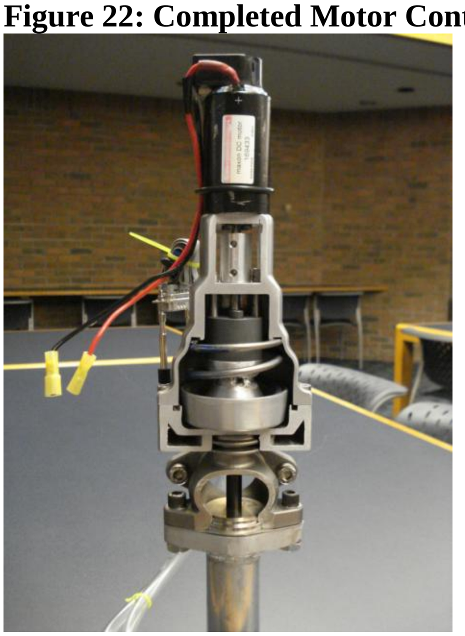
The other purchased components will be the DC motor and the threaded rod. All other features of the design will be machined by hand. Inside the wastegate we will be attaching an aluminum plate to the top of the spring retainer. Attached to this aluminum plate, is a threaded kevlar-nylon block that will be threaded onto the screw. The screw is a 3/8” threaded rod turned down to ¼” at the top. It is directly coupled to the shaft of the DC motor. As the screw is turned by the motor, the kevlar-nylon block will move either up or down the rod, providing compression or extension of the spring and therefore positioning the valve as desired. In order to allow for this movemement there is a clearance housing welded to the top of the spring housing. This clearance housing containes the threaded rod, and provides the space required for the kevlar-nylon block to move up and fully-open the valve. An additional component needs to be welded on top of the clearance housing, the motor bracket, in order to support the motor and provide a spot to couple the shaft of the motor to the threaded rod. The motor bracket has quarter-sections machined out of it to allow access to the screws attaching the motor as well as to tighten the coupler to both the shaft and the screw.

Detailed engineering drawings of the complete assembly, purchased wastegate with modifications, and all machined components can be found in the Appendices J through S. The prototype bill of materials which includes every component that will be purchased or supplied by the University of Michigan or EPA can be found in Appendix T. The total prototype cost is approximately $313.

PROTOTYPE MANUFACTURING

Manufacturing of the exhaust bypass valve prototype was performed in the student machine shop at the University of Michigan. The components were manufactured using the tools and machines available. Some of the major machines used in the manufacturing were: vertical mills, lathes, drill presses, TIG welder, grinders, band saws, and the laser cutter. Plans were developed for the manufacturing of all of the parts related to the creation of our prototype motor controlled exhaust bypass valve, seen in Figure 22 below.

Figure 22: Completed Motor Control Exhaust Bypass Valve Prototype



The following sections outline the manufacturing processes involved in the fabrication of all components related to our design. The dimensions of the components were measured with dial calipers, scales, height gauges, and digital readouts on the vertical mills to ensure accurate sizing of the individual components. Assembly of these components was done with the aid of various hand tools available in the tool crib of the student machine shop.

Aluminum Plate Manufacturing Plan

The aluminum plate consists of round stock blank welded to the original spring retainer. The adapter that was welded to the spring cup was turned in the lathe to a rough diameter of 1.40”. A 0.8” diameter 0.1” deep recess was cut into the base of the part to position it on the spring cup. Next, the adapter was cut off the blank and turned to a rough length of 0.31”. The adapter was then welded to the spring cup with a TIG welder at 120 amps using 0.0625” diameter 5056 filler material. In order to ensure that the surface of the adapter was perpendicular to the valve stem hole in the spring cup, it was placed in the lathe and faced off. After facing off the adapter, it was placed on the table on the bed of the vertical mill. T-slots were used with a ¾” nut and washer. Angle brackets with toe clamps were used to clamp the spring cup to the table. A locator was then used to center the head of the mill over the center of the adapters face. Once centered, the digital readout of the mill was zeroed and used to locate and drill 4 holes 0.190” on a 1” bolt circle, chamfer them with a counter 45 degree sinking bit, and thread the four holes with a 10-32 tap with 90 degrees between the holes. Lastly a clearance hole for the threaded rod was located and drilled in the center of the adapter to a depth of 0.25” and a diameter of 0.390”. All edges and holes were de-burred and filed smooth. Please refer to Figure 23 on page 29 to see the aluminum plate adapter.

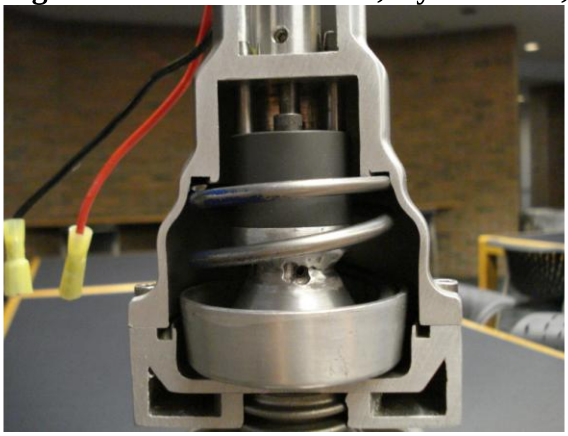
Threaded Nylon Block Manufacturing Plan

A similar process was used to manufacture the threaded nylon cylinder spacer as the aluminum adapter. A rough round stock of nylon was loaded into the lathe and turned to a finish diameter of 1.40”. Next, the nylon cylinder was parted off to 0.65”. Before unloading the part from the lathe, the edges were chamfered and a clearance hole for a 3/8” bolt was drilled through the block. In order to drill the remaining hole pattern in the nylon block, an aluminum fixture was made. A 1” thick block of aluminum was located in the vise of the mill and a hole was drilled ½” in from the front left corner. This would be drilled with a 5/16” drill and tapped for 3/8”-16 UNC thread. The nylon block was then bolted to the fixture and the same four hole pattern was drilled through the cylinder only with a diameter of 0.201” clearance for the 10-32 bolts. Next a center drill and a “Q” drill of diameter 0.332” were used to drill thru the center of the spacer. The hole was chamfered and tapped with a 3/8”-24 tap. Lastly two 0.246” diameter holes were drilled 180 degrees apart between the 4 hole pattern. These holes were then reamed to a final diameter of 0.25”. Please refer to Figure 23 on page 29 to see the threaded nylon block.

Threaded Rod Manufacturing Plan

The threaded rod was manufactured from a 3/8”-24 bolt that was 1.5” long. In order to machine the bolt to the desired shape, a fixture was made in the lathe. The fixture just consisted of a 2” piece of steel round stock that was drilled and tapped for 3/8-24 thread about 1.5” deep. A nut was threaded onto the bolt and then the bolt was threaded into the fixture. The nut of the bolt was then tightened against the fixture to lock the bolt from spinning. Now the head of the bolt could be faced off. Once the head of the bolt was faced off, a small center drill was used to make an indicating hole in the end of the bolt. The tail stock of the lathe was then brought up to the bolt and pressure was applied. Now the diameter of the bolt could safely be turned without needing the small diameter bolt. The outer diameter of the bolt was turned down to 0.236". A flat was also filed onto the threaded rod so set screws can hold it in place. Please refer to Figure 23 below to see the threaded rod.

Figure 23: Aluminum Plate, Nylon Block, Threaded Rod Connection



Motor Coupler Manufacturing Plan

To manufacture the motor coupler a piece of 1" aluminum round stock was turned down to 0.75" diameter and 0.750" length on the lathe. The cylinder was center drilled and then drilled with a 0.236" bit. This is the diameter of the motor and the turned down diameter of the threaded rod. The coupler was loaded into a vice and drilled in the drill press for four set screws. The set screw holes were made by center drilling 0.125" in from each end and then drilling to a depth of 0.375" with a 0.159" diameter drill. Each hole was then tapped with a 10-32 tap. All edges were filed and de-burred. Please refer to Figure 24 on page 30 to see the motor coupler.

Spring Housing Modifications

Modifications to the spring housing were necessary to begin attaching the motor bracket and clearance housing. The spring housing was bolted to the bottom plate of the original wastegate and this assembly was loaded into the lathe. The top surface of the spring housing was drilled through with a 1" drill. Next a boring bar was used to open up the spring cup top to a diameter of 2.5". The hole was then de-burred and filed smooth. Please refer to Figure 24 on page 30 to see the spring housing.

Clearance Housing Manufacturing Plan

Manufacturing of the clearance housing started with a piece of 1" long, 3" diameter round stock aluminum. It was placed in a lathe and turned down to a 2.75" diameter. Once the outer diameter was turned the round stock was then hollowed out to an approximate wall thickness of 0.125". The hollow tube was then removed from the lathe and cut off to a rough length of 2" in the horizontal band saw. The rough cut ring was then loaded back into the lathe and faced off to a final length of 1.5". Next a circular plate of aluminum was rough cut on the band saw with a speed of 325 fpm. The plate was then sandwiched between the tailstock and the chuck of the lathe and turned round to a diameter of 2.60”. This round disc was then welded to the ring. The completely welded ring and plate was then welded to the spring housing. This welded assembly was then loaded back into the lathe. The face of the circular plate was then faced off to ensure that it was parallel to the base of the spring cup. A center drill was also used to drill through the face of the clearance housing. When the whole prototype was welded together a 7/16” hole to clearance the threaded rod was drilled to ensure an accurate location in the center so that all pieces were aligned. This welded assembly was then clamped to the bed of the vertical mill with a t-slot and a ¾” bolt. Using the previously drilled hole in the face of the clearance housing the part was located under the center of the mill head. The digital read out was zeroed and two ¼” holes were drilled on a 1” bolt circle. Please refer to Figure 24 below to see the clearance housing.

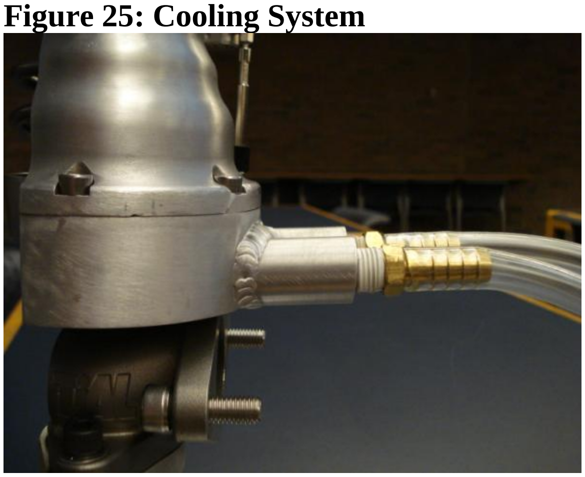
Motor Bracket Manufacturing Plan

For the motor bracket a piece of 1.5” aluminum round stock was turned to a diameter of 1.45” and a length of 1.25”. The round stock was bored with a boring bar to have a wall thickness of 0.125”. This ring was faced off and de-burred on both ends. Next, a circular plate of aluminum was rough cut on the band saw with a speed of 325 fpm. The plate was then sandwiched between the tailstock and the chuck of the lathe and turned round to a diameter of 1.25”. This plate was then welded to the ring that was made earlier. The ring and plate was then welded to the top of the clearance housing. This welded assembly was then loaded back into the lathe. The top of the motor bracket was then faced off in the lathe to ensure that the motor would be perfectly aligned with the components below it. A centering hole was also drilled in the face of the motor bracket and bored to a diameter of 0.625”. Now that the top surface of the motor bracket and the bottom surface of the spring cup were made parallel, the assembly was then bolted back to the bed of the vertical mill for the drilling of the hole pattern. Using the center drilled hole in the part and a locator, the head of the mill was centered over the center of the part. Using geometry and the bolt circle of the hole pattern given by the Maxon DC motor blue print, we located and drilled the holes for the motor. Center drills and clearance holes for 3 mm bolts were drilled with 60 degree spacing on a 22 mm bolt circle. All edges were de-burred and smoothed. Please refer to Figure 24 below to see the motor mount.

Figure 24: Motor, Motor Coupler, Threaded Rod Connection

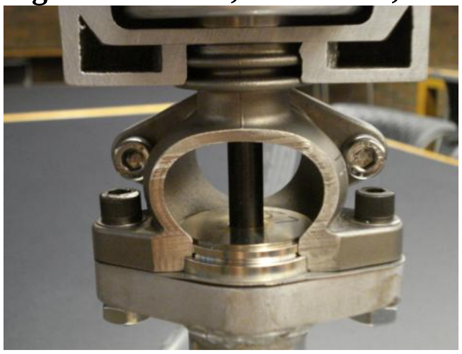
Cooling System Manufacturing Plan
For the cooling system manufacturing, we cut a 4"x 4" piece of 1" thick aluminum flat stock. Using the lathe, we turned the piece down to a round washer, with a diameter of 3.75". We turned the inside of the cooling system on the lathe out to a diameter of 1.5" to allow it to fit over the valve housing. From there we put the piece into the rotary table on the mill, and using a 3/4" end mill we machined out a cavity to provide a path for the coolant to flow through. On the rotary table, we only machined the cavity to approximately 330° around to leave a division providing an inlet and outlet for the coolant. We drilled two 1/4" holes on each side of the division where the hose barbs would be attached. Once the cavity was machined out, we used the mill to take the inside edge down to 0.440", and the outside edge down to 0.680" allowing it to fit to the bottom of the spring housing. Using 1" round stock on the lathe, we turned down two 3/4" long pieces to 5/8" providing an extension to attach the hose barbs into the cooling system. We drilled through each of these pieces with a Q drill and tapped them with a ¼" pipe thread. We welded these onto the cooling system to line up with the inlet and outlet holes, and then welded the entire piece to the bottom of the spring housing. Please refer to Figure 25 below to see the cooling system.

Figure 25: Cooling System



Bottom flange
A bottom flange was made to hold the valve seat into the housing, as well as a place to attach a mount for holding the valve for display. This was cut from 0.75" aluminum to mirror the shape of the bottom of the wastegate. Next, this was loaded into the mill and drilled with the pattern of holes that match the bottom of the valve housing. The bolt spread for the mounting holes is 2.38" and the diameter is 0.375". All the edges and holes were de-burred. Please refer to Figure 26 on page 32 to see the bottom flange.

Figure 26: Valve, Valve Seat, Bottom Flange Connection



Round Stock Stand Piece Manufacturing Plan

For display purposes, a 1.5” piece of round stock was rough cut on the horizontal band saw to a length of 6.5”. This blank was the loaded into the lathe and both ends were faced off to an overall length of 6”. This piece was preheated and welded to the bottom flange using a TIG welder with 275 amps and 1/8” 5056 filler material. Please refer to Figure 27 below to see the piece of round stock for the stand.

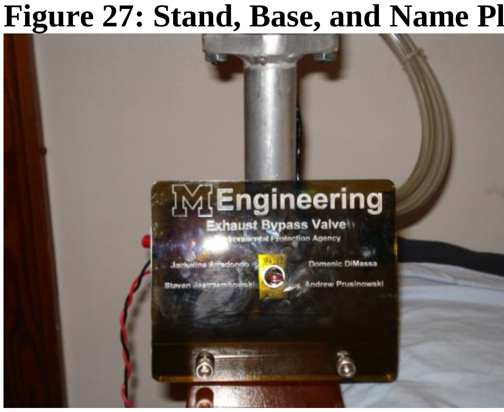
Base Plate for Stand Manufacturing Plan

The base plate for the display was manufactured from a 1” thick plate of aluminum. A rough cut square of aluminum was cut to 6.3”x6.3”. This piece was then squared off in the vise of the vertical mill using a ¾” two flute end mill. After de-burring the edges the base plate was preheated and welded to the round stock and bottom flange. Please refer to Figure 27 below to see the base plate.

Name Plate Manufacturing Plan

A name plate for displaying the project name and our team members was manufactured using the laser cutter in the student machine shop. The lettering and designs were made using BOBCAD software located on the machine shop computer. Both laser etching and cutting were performed to complete the name plate and its features. After all the work was done on the laser cutter, the name plate was bent using a heat gun and a vise. 2 5/16” holes were drilled in the nameplate ½” in from each lower edge to mount it to the aluminum base plate and a ½” hole was drilled to mount a control switch. Please refer to Figure 27 below to see the name plate.

Figure 27: Stand, Base, and Name Plate



TESTING PLAN

Experimental testing of the prototype will not be conducted, because the valve we are manufacturing will be used solely for the purpose of displaying a design concept and not used in the HCCI engine directly. It was our sponsor's suggestion that we focus more on how the prototype design will demonstrate our suggested concept. To enhance the ability of our prototype to show the functionality of the valve, we will use the mill to cut away some of the outer structure in order to show the inner working of the valve and how the components machined, welded, and manufactured work together to solve our sponsors engineering problem. A basic control system was also added to our prototype to display the functionality. A three-position switch was wired to a 12 VDC connection as well as to the Maxon DC Motor to show the movement of our valve. The torque that the motor supplies when connected to a 12 volt power source was not enough to show substantial movement of the valve with out the assistance of the pressure forces that exhaust gases would apply to the bottom of the valve.

DESIGN/PROTOTYPE EVALUATION

We feel that our design for an exhaust bypass valve sufficiently meets all of our sponsor's requirements, but there are several things that need to be addressed to further improve our design.

The first proposed improvement would to be use a gear reduction system to increase the torque of the provided Maxon RE 40 DC motor to the needed level. Gear reduction systems are readily available by Maxon in a wide range of ratios and can be purchased to fit their motors. The required level of gear reduction would be 8:1 for our design.

The second proposed improvement would be to modify the motor mount to allow easier attachment or to re-position the motor altogether to reduce the valve's profile. This could easily be accomplished by using a transmission system that provides the required torque and angular velocity for our design. Bolting the motor to a plate that could be attached to top of the motor bracket would allow for much easier accessibility to the small bolt circle of bolts that is machined into the motors chassis.

The final proposed improvement would be reducing the spring constant to reduce the required motor torque. Our design required the spring to be in equilibrium with the maximum exhaust pressure to prevent the motor from having to provide the force to seal the valve as requested by our sponsor. We feel that the motor could at least provide a portion of this force and we would then be able to reduce the spring constant thereby reducing the required motor torque. An alternative to this would be to devise some type of system to have the spring not coupled to the motor in any way so the motor does not have to compress the large spring force.

We feel that after these improvements have been made our design of an exhaust bypass valve will function at a high level in the HCCI engine application and could be scaled up or down to be used in other applications if desired. Since the interaction between the exhaust pressure and the force required to move the poppet valve of its seat is complex, more dynamic analysis of the valve as a system would be necessary to perfect our component selection and implement a control system to control the exhaust flow in the HCCI engine application.

CONCLUSIONS

Our team was presented with the task of designing an exhaust bypass valve for an HCCI engine system that can function at high pressures and temperatures with minimal leakage. Research was conducted on the system our exhaust bypass valve is being installed in as well as existing valves and patents so we can design the best possible valve. Our sponsor, Dr. Sun of the EPA, has provided us with detailed customer requirements that the exhaust bypass valve must meet. Based on these requirements, we were able to develop a set of engineering specifications to focus on for our design. A project plan was developed to ensure that we are on task throughout the entire design and manufacturing process. We have since come up with 7 preliminary design concepts, evaluated each one, and narrowed our final design down to two very similar concepts, the motor controlled poppet exhaust bypass valve and the linear actuated rocker arm poppet exhaust bypass valve. Further research and meetings with our sponsor led us to choose a modified version of our preliminary motor controlled poppet exhaust bypass valve for our final design. Engineering analysis and drawings have been completed for all components related to our design as well as detailed manufacturing plans for our prototype. Our prototype was manufactured, assembled, and wired up to display the functionality of our design. Our team feels that our design of a motor controlled exhaust bypass valve meets all design requirements and with a few slight modifications can be successfully integrated into an HCCI engine application or any other similar application.

ACKNOWLEDGEMENTS

We would like to thank the following people for assisting us with the design and manufacturing of our exhaust bypass valve prototype:

- Robert “Bob” Coury
- Professor Katsuo Kurabayashi
- Professor Kazuhiro Saitou
- Dr. Ruonan Sun

We would not have been able to accomplish this project without the help and guidance of all of these people.

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TEAM MEMBER BIOS

Jackeline Arredondo

Jackeline Arredondo grew up in Los Angeles and Inglewood, CA where she graduated from Saint Mary’s Academy in 2002. While in high school she actively participated in student council as Treasurer and Secretary and also participated in various community service events. She was a member of California’s Scholarship Federation and earned the Baush & Lomb Science Award. At the age of eleven she decided she wanted to become an engineer since this profession incorporates her interest in math, science, and directly impacts society. Her college options ranged from UCLA, University of Rochester, to the University of Michigan-Ann Arbor and five more. She decided to attend the College of Engineering at U of M since it was top 5 at the time. She also decided to major in Mechanical Engineering since it provided an overall overview of engineering. Her interests in ethical problems lead her to pursue a minor in Political and Moral Philosophy. She interned with Bechtel’s Oil, Gas, and Chemicals Global Unit in Houston, TX the summer of 2007 where her experience sparked an interest in the Energy sector and is now seeking full-time employment in related industries. In the near future she seeks to attend graduate school where she plans to earn an MBA and a master in International Affairs concentrating in Foreign Policy. She hopes to one day hold an executive position in a corporation or build her own in order to directly impact individuals by working towards the greater good for society.

Domenic Alexander Dimassa

Domenic grew up in South Lyon, Michigan where he graduated from South Lyon High School in the year 2000. While in high school he competed in Vocational Welding Competitions and was ranked 3rd in the state for overall welding. After high school he continued competition welding at Washtenaw Community College, eventually earning an Associate's degree in Welding and Fabrication. During the last year at Washtenaw he began working for a Livernois Vehicle Development, a prototype fabrication shop, as a metal model maker. Still wanting to continue his education, Domenic enrolled at the University of Michigan-Dearborn in the Mechanical Engineering department. In the winter term of 2006, Domenic transferred to University of Michigan-Ann Arbor. While working towards earning his Bachelors degree, he accepted a work study position as a machine shop technician. Domenic currently is in his senior year, works for the university machine shop, is working part-time for a HVAC company, and is proud to be an expecting father in February 2008. Outside of work and school he enjoys drag racing, metal fabrication, and hanging out in the garage working on cars.

Steven James Jastrzembowski

Steven grew up in Pinckney, Michigan, graduating from Pinckney Community High School in 2004. In high school he was a 4-year member of the tennis team and competed in Robotic Work Cell competitions, winning best in show his sophomore and junior years for work cells assembling utility knives and manufacturing screw drivers respectively. After high school Steven enrolled at the University of Michigan-Ann Arbor in the College of Engineering. In the next year he made the decision to declare a major of Mechanical Engineering. Currently, as a work study job, Steven works for the Data Sharing for Demographic Research (DSDR) division of the Inter-University Consortium for Political and Social Research (ICPSR) at the University of Michigan where he does basic statistical analysis and other related tasks. In May, Steven will graduate with a Bachelor's degree in Mechanical Engineering and begin working full-time. Outside of work and school he enjoys hunting, fishing, boating, and pretty much every other sport.

Andrew John Prusinowski

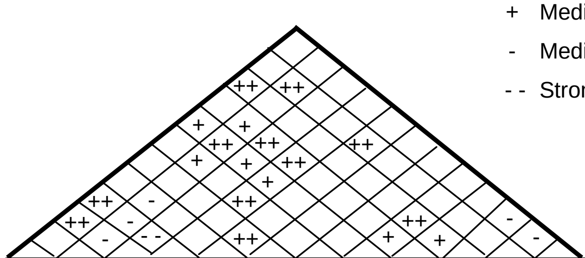
Andrew grew up in Royal Oak, Michigan where he graduated from Kimball High School in 2004. In high school he was the co-leader of his FIRST National Robotics Team where they were one of the top teams in the state and went to the national finals in Atlanta, Ga. He was also the captain of the varsity swim team at his high school. When he graduated he enrolled at the University of Michigan-Ann Arbor to pursue a degree in Mechanical Engineering. Andrew is the Product/Sales Manager for a company called Solid State Racing, which specializes in electronic components for race cars. He began working at Pratt & Miller Engineering this past summer who builds the Corvette, Cadillac, and Pontiac GTO race cars, and he has plans to continue working there upon graduation in May. Outside of school he enjoys working on cars, rebuilding engines, racing, and anything that involves going fast.

APPENDICES

Appendix A: Quality Function Development (QFD) Diagram

Relationships

++ Strong Positive
+ Medium Positive
- Medium Negative
- - Strong Negative



Benchmarks

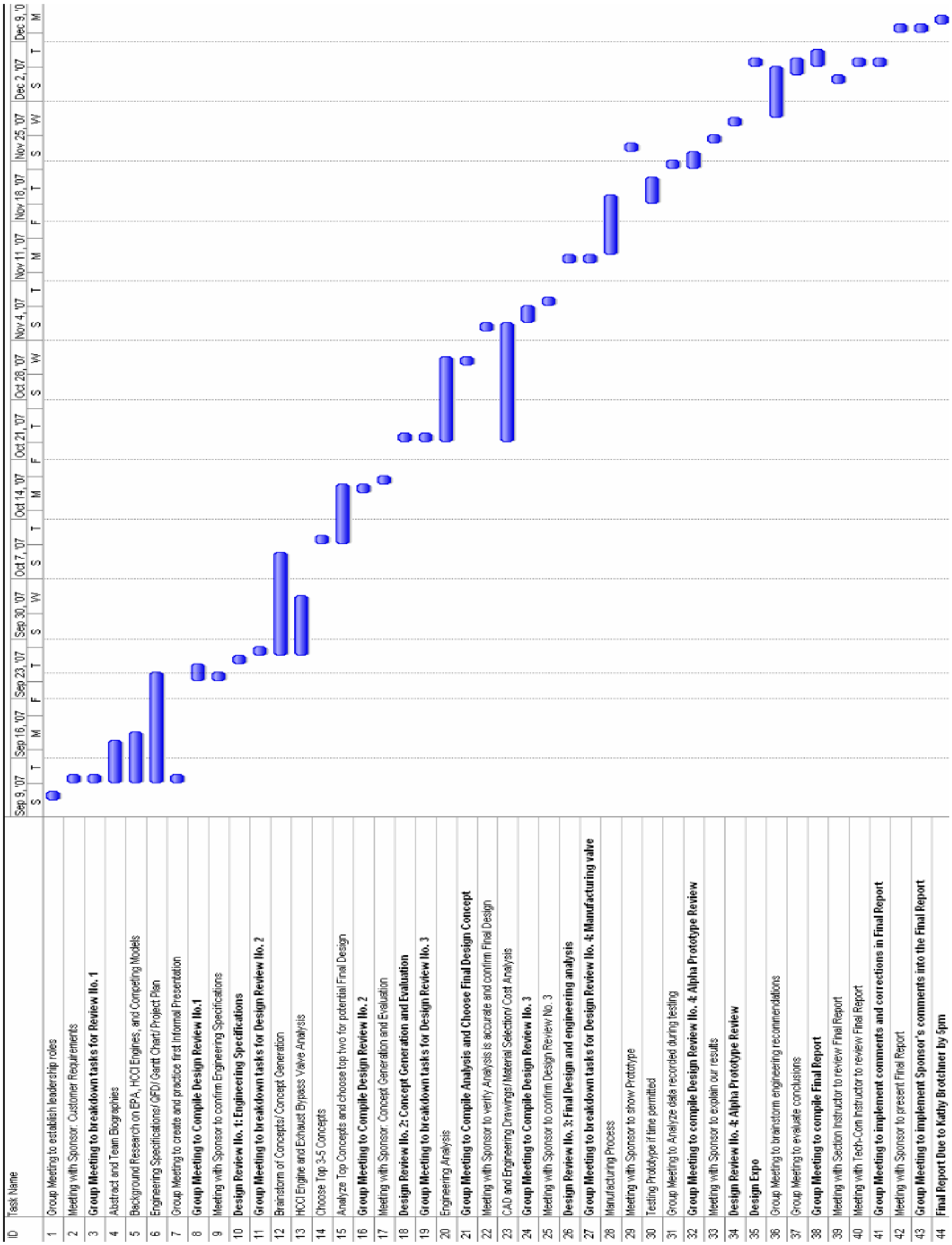
(+) => more is better
(-) => less is better

	Weight*	Opening Area	Maximum Gage Pressure	Area Proportionally Opened	0 to 20% Opening Speed	Operating Temperature	Electronics Temperature	Lubrication	Type of Valve	Type of Actuator	Overall Size/Shape of Valve	Material Choice	Manufacturablility
Seal Against High Pressures	10	3	9	3	1	3		1	9			3	
Proportional Controlled Opening	10	9	3	9	9			3	9	9			
Open Quickly	10	3	3	9	9			3	9	9			
Operate in High Temperatures	8		1			9	9		1	1		9	
Minimal Wear	7		3		1	9	9	9	3			9	
Low Cost	5		3	3	3	1	1	1	3	3	1	9	3
Light Weight	3											1	1
Small Profile	6	3							1	1	9		1
Measurement Unit		in^2	bars	%	sec	°C	°C						
Target Value		1.3	3	20	2	350 to 550	125						
Importance Rating		**7**	**6**	**2**	**3**	**7**	**9**	**9**	**1**	**3**	**11**	**3**	**12**
Total		168	194	225	212	170	140	138	320	209	59	213	24
Normalized		0.08	0.09	0.11	0.10	0.08	0.07	0.07	0.15	0.10	0.03	0.10	0.01

Key:
9 => Strong Relationship
3 => Medium Relationship
1 => Small Relationship
(blank) => Not Related

***Weights are figured on a scale of 1 to 10**
(ten being most important)

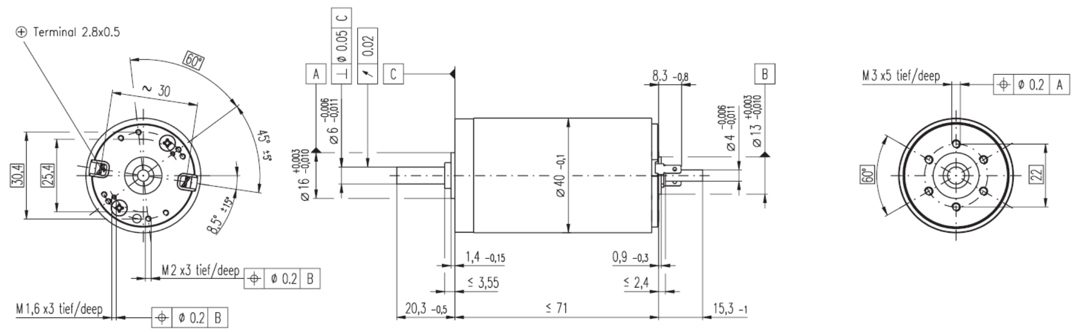
Appendix B: Gantt Chart



Appendix C: Maxon RE 40 DC Motor Drawing/Specifications [6]

RE 40 ∅40 mm, Graphite Brushes, 150 Watt

maxon DC motor



M 1:2

- Stock program
- Standard program
- Special program (on request)

Order Number

Motor Data	Unit	148866	148867	148877	218008	218009	218010	218011	218012	218013	218014	218015
Values at nominal voltage												
1 Nominal voltage	V	12.0	24.0	48.0	48.0	48.0	48.0	48.0	48.0	48.0	48.0	48.0
2 No load speed	rpm	6920	7580	7580	6420	5560	3330	2690	2130	1710	1420	987
3 No load current	mA	241	137	68.6	53.7	43.7	21.9	16.7	12.5	9.67	7.77	5.16
4 Nominal speed	rpm	6370	6930	7000	5810	4920	2700	2050	1500	1080	774	339
5 Nominal torque (max. continuous torque)	mNm	94.9	170	184	183	177	187	187	189	189	188	188
6 Nominal current (max. continuous current)	A	6.00	5.77	3.12	2.62	2.20	1.38	1.12	0.898	0.721	0.593	0.413
7 Stall torque	mNm	1680	2280	2500	1990	1580	995	796	641	512	415	289
8 Starting current	A	102	75.7	41.4	28.0	19.2	7.26	4.68	3.00	1.92	1.29	0.627
9 Max. efficiency	%	88	91	92	91	91	89	88	87	86	85	83
Characteristics												
10 Terminal resistance	Ω	0.117	0.317	1.16	1.72	2.50	6.61	10.2	16.0	24.9	37.1	76.6
11 Terminal inductance	mH	0.0245	0.0823	0.329	0.460	0.612	1.70	2.62	4.14	6.40	9.31	19.2
12 Torque constant	mNm / A	16.4	30.2	60.3	71.3	82.2	137	170	214	266	321	461
13 Speed constant	rpm / V	581	317	158	134	116	69.7	56.2	44.7	35.9	29.8	20.7
14 Speed / torque gradient	rpm / mNm	4.15	3.33	3.04	3.23	3.53	3.36	3.39	3.35	3.37	3.44	3.45
15 Mechanical time constant	ms	6.03	4.81	4.39	4.36	4.35	4.31	4.31	4.31	4.31	4.32	4.33
16 Rotor inertia	gcm^2	139	138	138	129	118	123	121	123	122	120	120

Specifications

Thermal data

17	Thermal resistance housing-ambient	4.65 K / W
18	Thermal resistance winding-housing	1.93 K / W
19	Thermal time constant winding	41.6 s
20	Thermal time constant motor	1120 s
21	Ambient temperature	-20 ... +100°C
22	Max. permissible winding temperature	+155°C

Mechanical data (ball bearings)

23	Max. permissible speed	12000 rpm
24	Axial play	0.05 - 0.15 mm
25	Radial play	0.025 mm
26	Max. axial load (dynamic)	5.6 N
27	Max. force for press fits (static)	110 N
	(static, shaft supported)	1200 N
28	Max. radial loading, 5 mm from flange	28 N

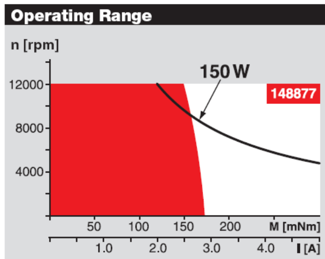
Other specifications

29	Number of pole pairs	1
30	Number of commutator segments	13
31	Weight of motor	480 g

Values listed in the table are nominal.
Explanation of the figures on page 47.

Option
Preloaded ball bearings

Operating Range



Comments

Continuous operation
In observation of above listed thermal resistance (lines 17 and 18) the maximum permissible winding temperature will be reached during continuous operation at 25°C ambient.
= Thermal limit.

Short term operation
The motor may be briefly overloaded (recurring).

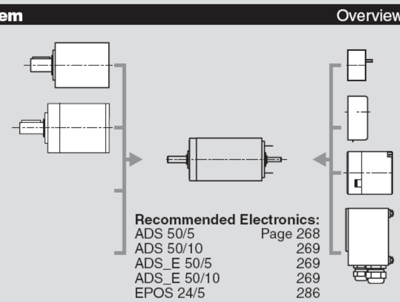
Assigned power rating

maxon Modular System

Overview on page 16 - 21

Planetary Gearhead
∅42 mm
3 - 15 Nm
Page 235

Planetary Gearhead
∅52 mm
4 - 30 Nm
Page 238



Recommended Electronics:

ADS 50/5	Page 268
ADS 50/10	269
ADS_E 50/5	269
ADS_E 50/10	269
EPOS 24/5	286
EPOS P 24/5	287
EPOS 70/10	287
MIP 50, MIP 100	289
Notes	**18**

Encoder MR
256 - 1024 CPT,
3 channels
Page 251

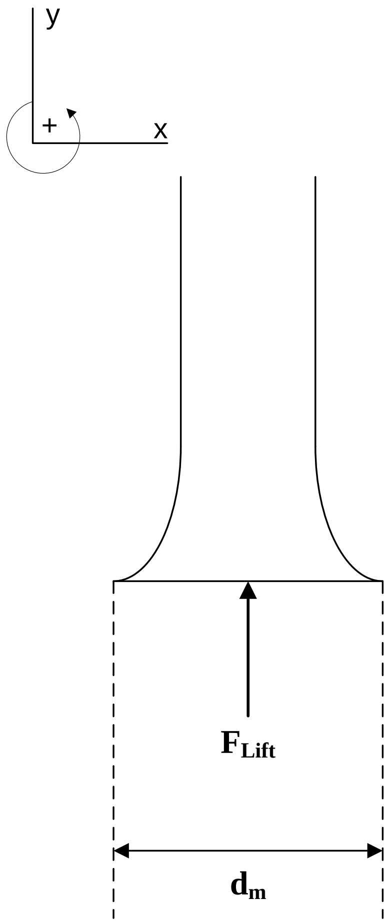
Encoder HED_ 5540
500 CPT,
3 channels
Page 254 / 256

Brake AB 28
∅28 mm,
24 VDC, 0.4 Nm
Page 300

Industrial Version
Encoder HEDL 9140
Page 259
Brake AB 28
Page 301

Appendix D: Lift Force Analysis

Figure D.1: Valve Free Body Diagram



Force Applied by Exhaust Pressure:

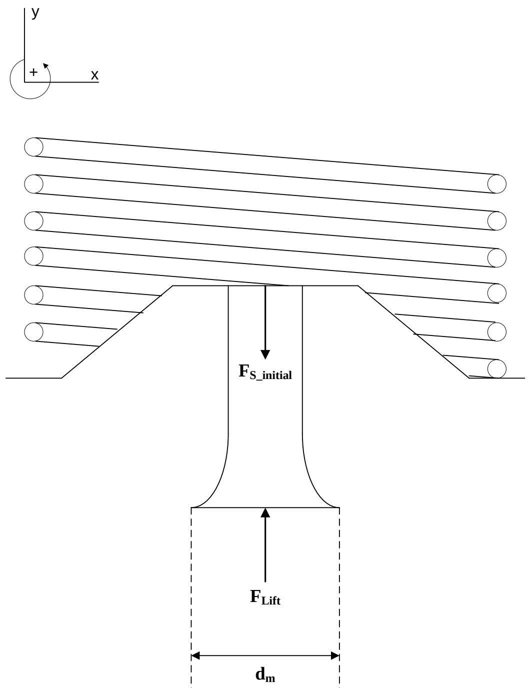
$$P = 45psi = 310300 Pa$$

$$A = \frac{\pi \cdot d_m^2}{4} = 1.75in^2 = 0.011m^2$$

$$F_{Lift} = P \bullet A = 79.1\,lbs = 352\ N$$

Appendix E: Valve Spring Analysis

Figure E.1: Valve Spring Free Body Diagram



Equilibrium Equations for Valve Spring System:

$$\sum F_y = 0 = F_{Lift} - F_{S\_initial}$$

$$\therefore F_{S\_initial} = F_{Lift} = 79.1\, lbs = 352\ N$$

$$F_{S\_initial} = k \bullet \Delta x_{initial}$$

$$\Delta x_{initial} = 0.375\, in = 0.009\, m$$

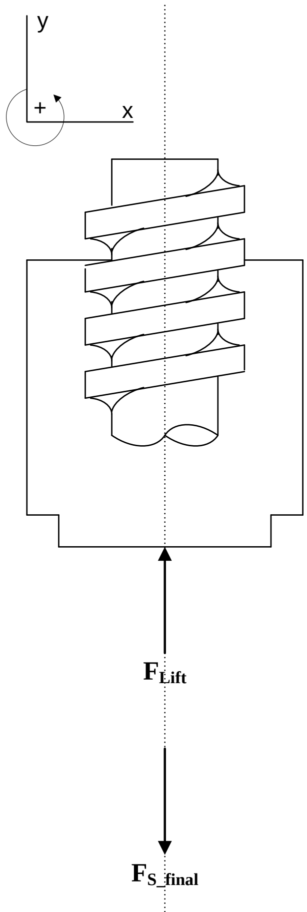
$$\therefore k = \frac{F_{Lift}}{\Delta x_{initial}} = 210.95\, lbs/in = 36946\ N/m$$

Table E.1: EXHAUST FORCE AND SPRING CONSTANT FINDINGS

CONSTANTS	VARIABLE	VALUE	ENGLISH UNITS	VALUE	METRIC UNITS
Exhaust Pressure	P	45.000	psi	310300.000	Pa
Valve Diameter	d_{Valve}	1.496	in	0.038	m
Valve Area	A_{Valve}	1.758	in^2	0.001	m^2
Exhaust Lift Force	$\mathbf{F_{Lift}}$	**79.105**	**lbs force**	**351.916**	**N**
Initial compression of Spring required to close housing	$\Delta x_{Initial}$	0.375	in	0.010	m
Spring Force required to close housing	$F_{S\_initial}$	79.105	lbs force	351.916	N
Spring Constant required to close housing	$\mathbf{k_S}$	**210.946**	**lbs force/in**	**36946.548**	**N/m**

Appendix F: Torque Analysis

Figure F.1: Screw and Kevlar/Nylon Block Free Body Diagram



Derivation of Force on Screw:

$$F_{S\_final} = k \bullet \Delta x_{final}$$

$$\sum F_y = F_{Net} = F_{Lift} - F_{S\_Final}$$

$$F_{Net} = F_{Screw}$$

$$\therefore F_{Screw} = F_{Lift} - F_{S\_Final}$$

Equations for Raising the Valve:

$$p = 24.0\ threads/in = 945\ threads/m$$

$$l = \frac{1}{p}$$

$$d_m = 0.355 in = 0.009 m$$

$$\tan\lambda = \frac{l}{\pi \cdot d_m}$$

$$\sum F_x = P_R - N \cdot \sin(\lambda) - f \cdot N \cdot \cos(\lambda) =$$

$$\sum F_y = -F_{Screw} - f \cdot N \cdot \sin(\lambda) + N \cdot \cos(\lambda) = 0$$

$$\therefore P_R = \frac{F_{Screw} \cdot [\sin(\lambda) + f \cdot \cos(\lambda)]}{\cos(\lambda) - f \cdot \sin(\lambda)} = \frac{F_{Screw} \cdot [f + (\frac{l}{\pi \cdot d_m})]}{1 - (\frac{f \cdot l}{\pi \cdot d_m})}$$

$$T_R = \frac{d_m}{2} \cdot P_R$$

Torque Required to Raise Valve when neglecting friction:

$$T_0 = \frac{F_{Screw} \cdot l}{2 \cdot \pi}$$

Screw Efficiency:

$$e = \frac{T_0}{T_R}$$

Figure F.2: Free Body Diagram for Raising the Valve

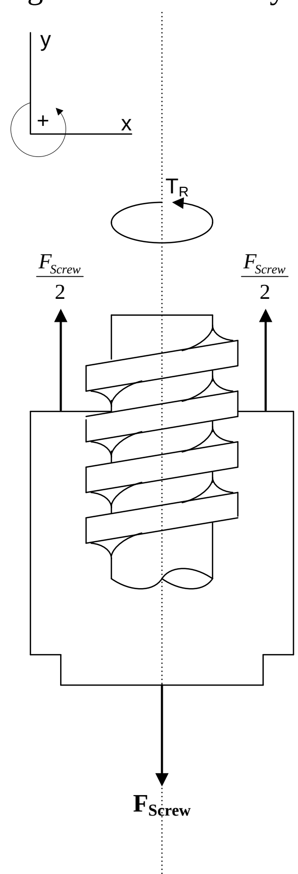


Figure F.3: Free Body Diagram of an Unrolled Thread

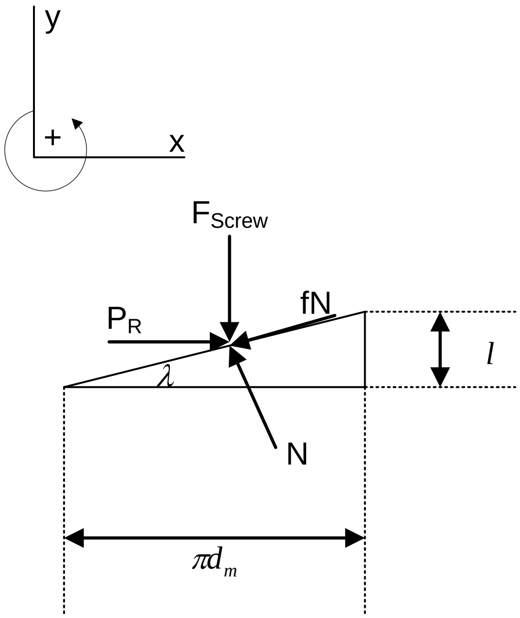


Table F.1: 100% OF 1.3in^2 OPEN AREA WITH MAX DRY STATIC FRICTION

	VARIABLE	VALUE	ENGLISH UNITS	VALUE	METRIC UNITS
Final compression of Spring	Δx_{Final}	0.606	in	0.015	m
Final Spring Force	F_{S_final}	128	lbs force	569	N
Force applied on Screw	F_{screw}	48.8	lbs force	217	N
Screw pitch	p	24.0	threads/in	945	threads/m
Thread Height	l	0.042	in	0.001	m
Mean Diameter of Screw	d_m	0.355	in	0.00902	m
Friction of Nylon	f	0.250		0.250	
Force required to raise Screw	P_R	14.2	lbs force	63.0	N
Torque required to raise Screw	$\mathbf{T_R}$	**2.51**	**lbs force in**	**0.28399**	**Nm**
Torque with zero friction	T_0	0.324	lbs force in	0.037	Nm
Efficiency	e	12.9	%	12.9	%

Table F.2: 100% OF 1.3in^2 OPEN AREA WITH MIN DRY STATIC FRICTION

	VARIABLE	VALUE	ENGLISH UNITS	VALUE	METRIC UNITS
Final compression of Spring	Δx_{Final}	0.606	in	0.015	m
Final Spring Force	F_{S_final}	128	lbs force	569	N
Force applied on Screw	F_{screw}	48.8	lbs force	217	N
Screw pitch	p	24.0	threads/in	945	threads/m
Thread Height	l	0.042	in	0.001	m
Mean Diameter of Screw	d_m	0.355	in	0.00902	m
Friction of Nylon	f	0.150		0.150	
Force required to raise Screw	P_R	9.20	lbs force	40.9	N
Torque required to raise Screw	$\mathbf{T_R}$	**1.63**	**lbs force in**	**0.184**	**Nm**
Torque with zero friction	T_0	0.324	lbs force in	0.037	Nm
Efficiency	e	19.8	%	19.8	%

Table F.3: 100% OF 1.3in^2 OPEN AREA WITH MAX SLIDING FRICTION

	VARIABLE	VALUE	ENGLISH UNITS	VALUE	METRIC UNITS
Final compression of Spring	Δx_{Final}	0.606	in	0.015	m
Final Spring Force	F_{S_final}	128	lbs force	569	N
Force applied on Screw	F_{screw}	48.8	lbs force	217	N
Screw pitch	p	24.0	threads/in	945	threads/m
Thread Height	l	0.042	in	0.001	m
Mean Diameter of Screw	d_m	0.355	in	0.00902	m
Friction of Nylon	f	0.125		0.125	
Force required to raise Screw	P_R	7.96	lbs force	35.4	N
Torque required to raise Screw	$\mathbf{T_R}$	**1.41**	**lbs force in**	**0.160**	**Nm**
Torque with zero friction	T_0	0.324	lbs force in	0.037	Nm
Efficiency	e	22.9	%	22.9	%

Table F.4: 100% OF 1.3in^2 OPEN AREA WITH MIN SLIDING FRICTION

	VARIABLE	VALUE	ENGLISH UNITS	VALUE	METRIC UNITS
Final compression of Spring	Δx_{Final}	0.606	in	0.015	m
Final Spring Force	F_{S_final}	128	lbs force	569	N
Force applied on Screw	F_{screw}	48.8	lbs force	217	N
Screw pitch	p	24.0	threads/in	945	threads/m
Thread Height	l	0.042	in	0.001	m
Mean Diameter of Screw	d_m	0.355	in	0.00902	m
Friction of Nylon	f	0.075		0.075	
Force required to raise Screw	P_R	5.50	lbs force	24.5	N
Torque required to raise Screw	$\mathbf{T_R}$	**0.976**	**lbs force in**	**0.110**	**Nm**
Torque with zero friction	T_0	0.324	lbs force in	0.037	Nm
Efficiency	e	33.2	%	33.2	%

Table F.5: 20% OF 1.3in^2 OPEN AREA WITH MAX DRY STATIC FRICTION

	VARIABLE	VALUE	ENGLISH UNITS	VALUE	METRIC UNITS
Final compression of Spring	Δx_{Final}	0.415	in	0.011	m
Final Spring Force	F_{S_final}	87.6	lbs force	390	N
Force applied on Screw	F_{screw}	8.48002	lbs force	37.7	N
Screw pitch	p	24.0	threads/in	945	threads/m
Thread Height	l	0.042	in	0.001	m
Mean Diameter of Screw	d_m	0.355	in	0.00902	m
Friction of Nylon	f	0.250		0.250	
Force required to raise Screw	P_R	2.46	lbs force	10.9	N
Torque required to raise Screw	$\mathbf{T_R}$	**0.437**	**lbs force in**	**0.049**	**Nm**
Torque with zero friction	T_0	0.056	lbs force in	0.006	Nm
Efficiency	e	12.9	%	12.9	%

Table F.6: 20% OF 1.3in^2 OPEN AREA WITH MIN DRY STATIC FRICTION

	VARIABLE	VALUE	ENGLISH UNITS	VALUE	METRIC UNITS
Final compression of Spring	Δx_{Final}	0.415	in	0.011	m
Final Spring Force	F_{S_final}	87.6	lbs force	390	N
Force applied on Screw	F_{screw}	8.48002	lbs force	37.7	N
Screw pitch	p	24.0	threads/in	945	threads/m
Thread Height	l	0.042	in	0.001	m
Mean Diameter of Screw	d_m	0.355	in	0.00902	m
Friction of Nylon	f	0.150		0.150	
Force required to raise Screw	P_R	1.60	lbs force	7.11	N
Torque required to raise Screw	$\mathbf{T_R}$	**0.284**	**lbs force in**	**0.03205**	**Nm**
Torque with zero friction	T_0	0.056	lbs force in	0.006	Nm
Efficiency	e	19.8	%	19.8	%

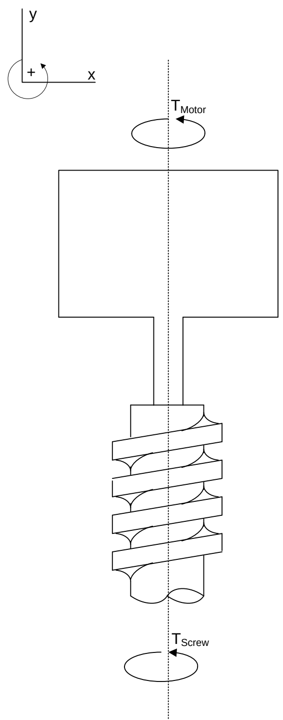
Table F.7: 20% OF 1.3in^2 OPEN AREA WITH MAX SLIDING FRICTION

	VARIABLE	VALUE	ENGLISH UNITS	VALUE	METRIC UNITS
Final compression of Spring	Δx_{Final}	0.415	in	0.011	m
Final Spring Force	F_{S_final}	87.6	lbs force	390	N
Force applied on Screw	F_{screw}	8.48002	lbs force	37.7	N
Screw pitch	p	24.0	threads/in	945	threads/m
Thread Height	l	0.042	in	0.001	m
Mean Diameter of Screw	d_m	0.355	in	0.00902	m
Friction of Nylon	f	0.125		0.125	
Force required to raise Screw	P_R	1.38	lbs force	6.15	N
Torque required to raise Screw	$\mathbf{T_R}$	**0.246**	**lbs force in**	**0.028**	**Nm**
Torque with zero friction	T_0	0.056	lbs force in	0.006	Nm
Efficiency	e	22.9	%	22.9	%

Table F.8: 20% OF 1.3in^2 OPEN AREA WITH MIN SLIDING FRICTION

	VARIABLE	**VALUE**	**ENGLISH UNITS**	**VALUE**	**METRIC UNITS**
Final compression of Spring	Δx_{Final}	0.415	in	0.011	m
Final Spring Force	F_{S_final}	87.6	lbs force	390	N
Force applied on Screw	F_{screw}	8.48002	lbs force	37.7	N
Screw pitch	p	24.0	threads/in	945	threads/m
Thread Height	l	0.042	in	0.001	m
Mean Diameter of Screw	d_m	0.355	in	0.00902	m
Friction of Nylon	f	0.075		0.075	
Force required to raise Screw	P_R	0.955	lbs force	4.25	N
Torque required to raise Screw	$\mathbf{T_R}$	**0.170**	**lbs force in**	**0.019**	**Nm**
Torque with zero friction	T_0	0.056	lbs force in	0.006	Nm
Efficiency	e	33.2	%	33.2	%

Figure F.4: Free Body Diagram of Motor and Screw



Equations for Motor used in Prototype:

$$\omega_{Motor} = \omega_{Screw}$$

$$\therefore T_{Screw} = T_{Motor} = 0.840\, lbs\, in = 0.095\, Nm$$

$$T_{Screw} = P_R \bullet \frac{d_m}{2} = P_R \bullet r_m$$

$$\therefore P_R = \frac{T_{Screw}}{r_m} = 21.1N = 4.73\, lbs\ \ force$$

Gear Reducer Analysis:

$$T_{Motor} = F_{Motor} \cdot R_{Motor}$$
$$T_R = P_R \cdot R_{Screw}$$
$$F_{Motor} = P_R \Rightarrow \frac{T_{Motor}}{R_{Motor}} = \frac{T_R}{R_{Screw}} \Rightarrow \frac{R_{Screw}}{R_{Motor}} \cdot T_{Motor} = T_R \Rightarrow N \cdot T_{Motor} = T_R$$
$$\therefore Gear\ Ratio,\ N = \frac{T_R}{T_{Motor}}$$
$$P_{Motor} = T_R \cdot \omega \Rightarrow \omega = \frac{P_{Motor}}{T_R}$$
$$\omega = \frac{\Delta\theta}{\Delta t} \Rightarrow \Delta t = \frac{\Delta\theta}{\omega}$$

Table F.9:

	VARIABLE	VALUE	ENGLISH UNITS	VALUE	METRIC UNITS
Torque Required to Raise Valve 100% of 1.3 in^2 Maximum Opening Area with zero Pressure and Dry Static Friction	T_R	6.58	Lbs-in	744	mN-m
Motor Torque	T_{Motor}	0.840	lbs-in	94.9	mN-m
Gear Ratio	N	7.84		7.84	
Motor Power	P_{Motor}	560	lbs-in/s	63.3	W
Continuous Rotational Speed Required	ω	813	RPM	85.1	rad/s
Time to Reach 20% of 1.3 in^2 Maximum Opening Area	t	0.148	s	0.148	s
Torque Required to Raise Valve 20% of 1.3 in2 Maximum Opening Area with zero Pressure and Dry Static Friction	T_R	4.51	Lbs-in	509	mN-m
Motor Torque	T_{Motor}	0.840	lbs-in	94.9	mN-m
Gear Ratio	N	5.36		5.36	
Motor Power	P_{Motor}	560	lbs-in/s	63.3	W
Continuous Rotational Speed Required	ω	1780	RPM	186	rad/s
Time to Reach 20% of 1.3 in^2 Maximum Opening Area	t	0.067	s	0.067	s

Appendix G: Speed Analysis

Equations for Moment of Inertia of Screw:

$$M_{Screw} = \rho_{Steel} \cdot V_{Screw} = \rho_{Steel} \cdot \pi \cdot r_{Screw}^2 \cdot l_{Screw}$$

$$I_{Screw} = M_{Screw} \cdot r_{Screw}^2$$

Table G.1: Screw Moment of Inertia

Screw: 1030 Carbon Steel	**Variable**	**Value**	**English Unit**	**Value**	**Metric Unit**
	Density	0.284	lbs/in^3	7861	kg/m^3
	Diameter	0.375	in	**0.010**	m
	Radius	0.188	in	0.005	m
	Length	1.50	in	**0.0381**	m
	Volume	0.166	in^3	0.000003	m^3
	Mass	0.047	lbs	0.021	kg
	Moment of Inertia	0.002	lbs-in^2	4.84E-07	kg-m^2

Equations for Moment of Inertia of Coupler:

$$M_{Coupler} = \rho_{Aluminum} \cdot V_{Coupler} = \rho_{Aluminum} \cdot \pi \cdot (r_{Coupler\_o}^2 - r_{Coupler\_i}^2) \cdot l_{Coupler}$$

$$I_{Coupler} = M_{Coupler} \cdot (r_{Coupler\_o}^2 - r_{Coupler\_i}^2)$$

Table G.2: Coupler Moment of Inertia

Coupler: 6061 Aluminum	**Variable**	**Value**	**English Unit**	**Value**	**Metric Unit**
	Density	0.098	lbs/in^3	2713	kg/m^3
	Outer Diameter	0.75	in	0.019	m
	Total Radius	0.375	in	0.010	m
	Length	0.750	in	**0.019**	m
	Volume	0.331	in^3	0.000005	m^3
	Mass	0.032	lbs	0.015	kg
	Moment of Inertia	0.005	lbs-in^2	1.34E-06	kg-m^2

Equations Determining Required Time to Reach 20% of Opening Area:

$$I_{Total} = I_{Screw} + I_{Coupler} = 1.82 \times 10^{-6}\ kg \cdot m^2$$

$$T_R = I_{Total} \cdot \alpha \Rightarrow \alpha = \frac{T_R}{I_{Total}} = \frac{0.744\ N \cdot m}{1.82 \times 10^{-6}\ kg \cdot m^2} = 408791\ rad/s^2$$

$$\theta = 2 \cdot \pi\ rads = 1\ rev\ to\ reach\ 20\%\ of\ Maximum\ Opening\ Area$$

$$\omega_o = 0\ rad/s \Rightarrow Starts\ from\ rest$$

$$\omega^2 = \omega_o^2 + 2 \cdot \alpha \cdot \theta \Rightarrow \omega = \sqrt{2 \cdot \alpha \cdot \theta} = 2266\ rad/s$$

$$\omega = \omega_o + \alpha \cdot t \Rightarrow t = \frac{\omega}{\alpha} = 0.005\ \sec$$

Appendix H: Heat Transfer Analysis

Figure H.1: Thermal Exhaust Bypass Valve Model with Nodal Temperatures and Dimensions

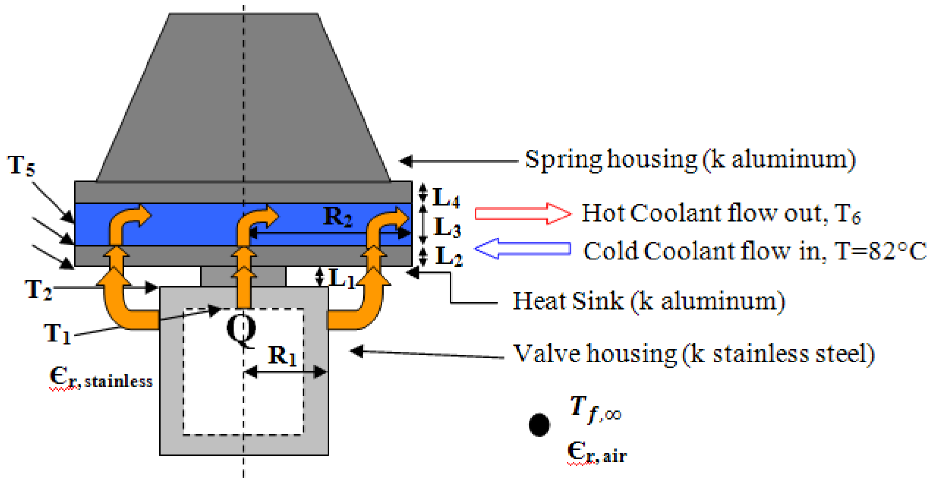


Figure H.2: Thermal Circuit Diagram for Heat Transfer throughout Exhaust Bypass Valve

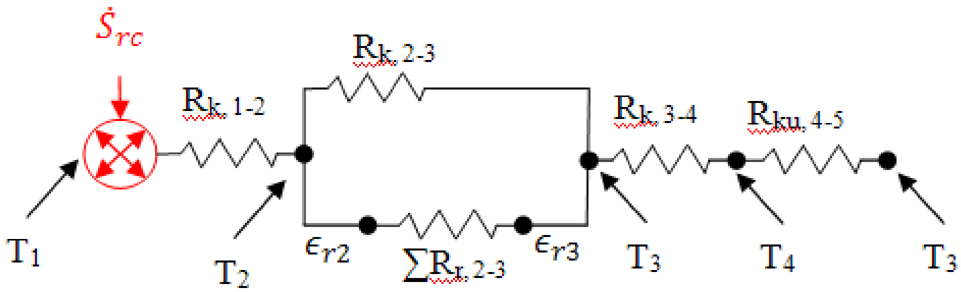
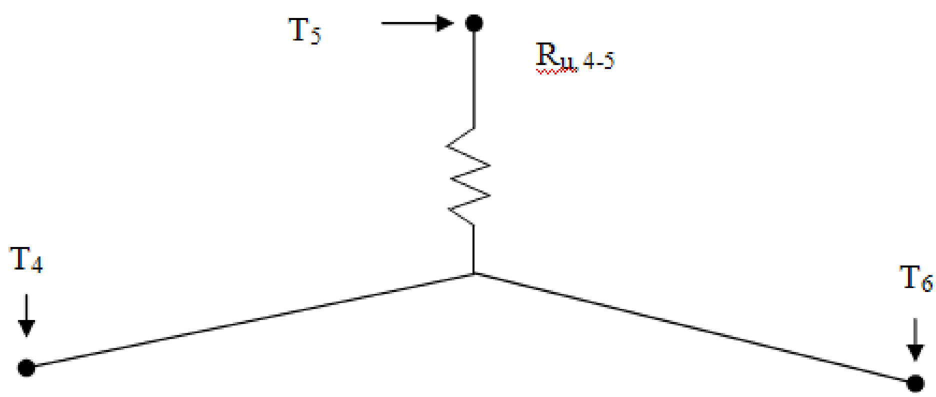


Figure H.3: Simplified Thermal Circuit Diagram for Heat Transfer through Cooling Pathway

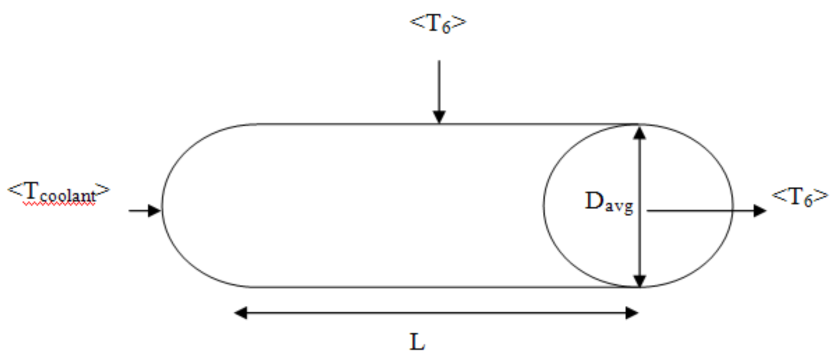


Heat Transfer in an Aluminum Cylinder Block:

$$Q_{k,1-2} = \left[\frac{T_1 - T_2}{R_{k,1-2}}\right] \; where \; R_{k,1-2} = \frac{L}{A_k k}$$

$$Q_{k,1-2} = 60KW$$

Figure H.4: Heat Transfer Model for Single Bounded Fluid Heat Exchanger



Reynolds Number:

$$R_{e,D} = \frac{\langle u_f \rangle D}{\nu}$$

$$\langle R_{e,D} \rangle = 3940$$

Nusselt Number:

$$\langle N_{uD} \rangle = 0.023 \langle R_{e,D} \rangle^{\frac{4}{5}} P_r^n \; where \; n = 0.4 \; for \; turbulent \; flows$$

$$\langle N_{uD} \rangle = \; 292.7$$

Number of Transfer Units and Effectiveness:

$$NTU = \frac{R_{u,f}}{\langle R_{ku} \rangle} = \frac{A_{ku} \langle N_{u,D} \rangle \frac{k_f}{D}}{\dot{M} C_p} = 0.808$$

$$\varepsilon_{he} = 1 - e^{-NTU} = 0.554$$

Convection Resistance and Heat Transfer through Model:

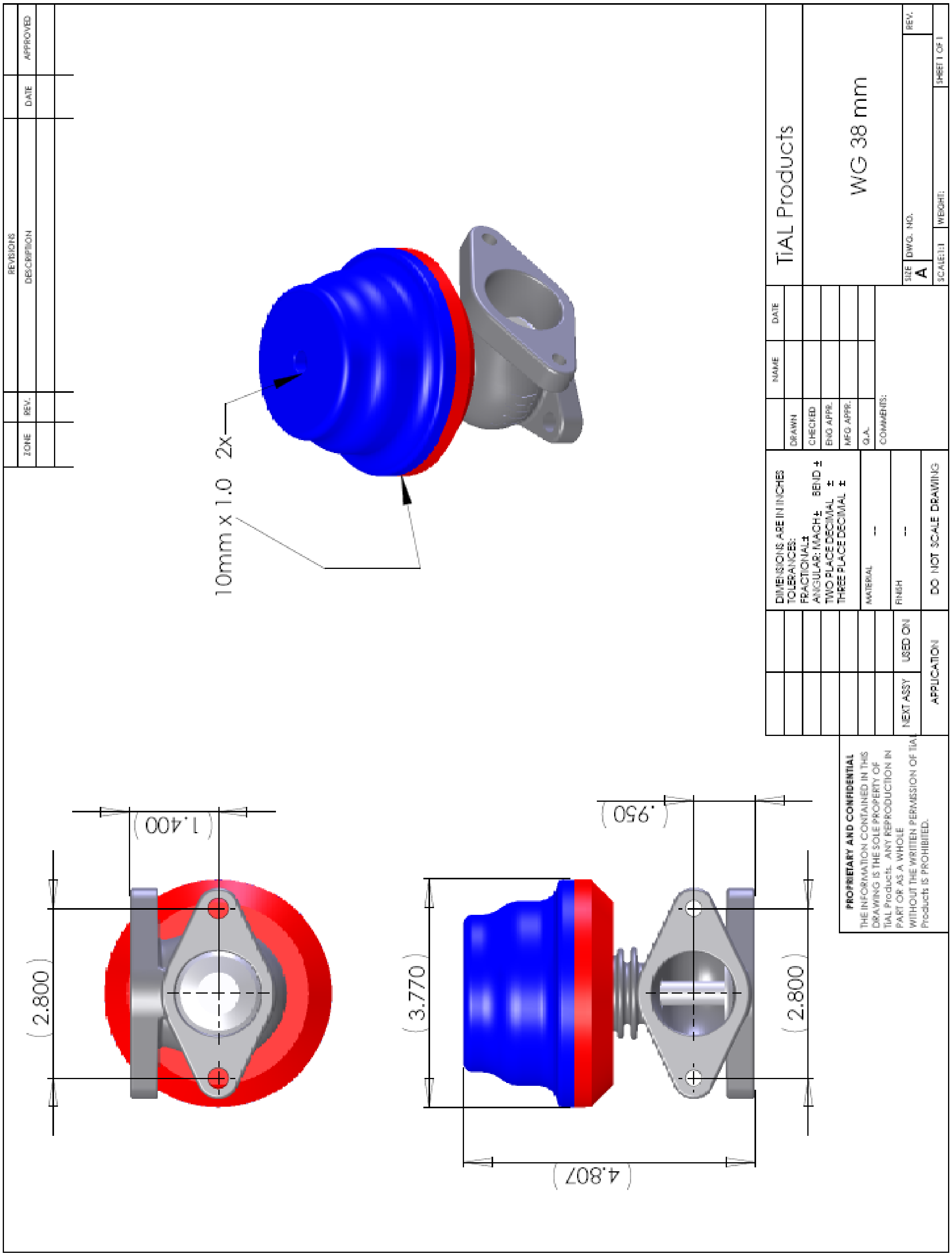
$$\langle R_{u,f} \rangle = \frac{1}{\dot{M} C_p \varepsilon_{he}} = 8.14 \times 10^{-3} \frac{K}{W}$$

$$\langle Q_u \rangle = \frac{T_6 - T_{coolant}}{\dot{M} C_p \varepsilon_{he}} = 51KW$$

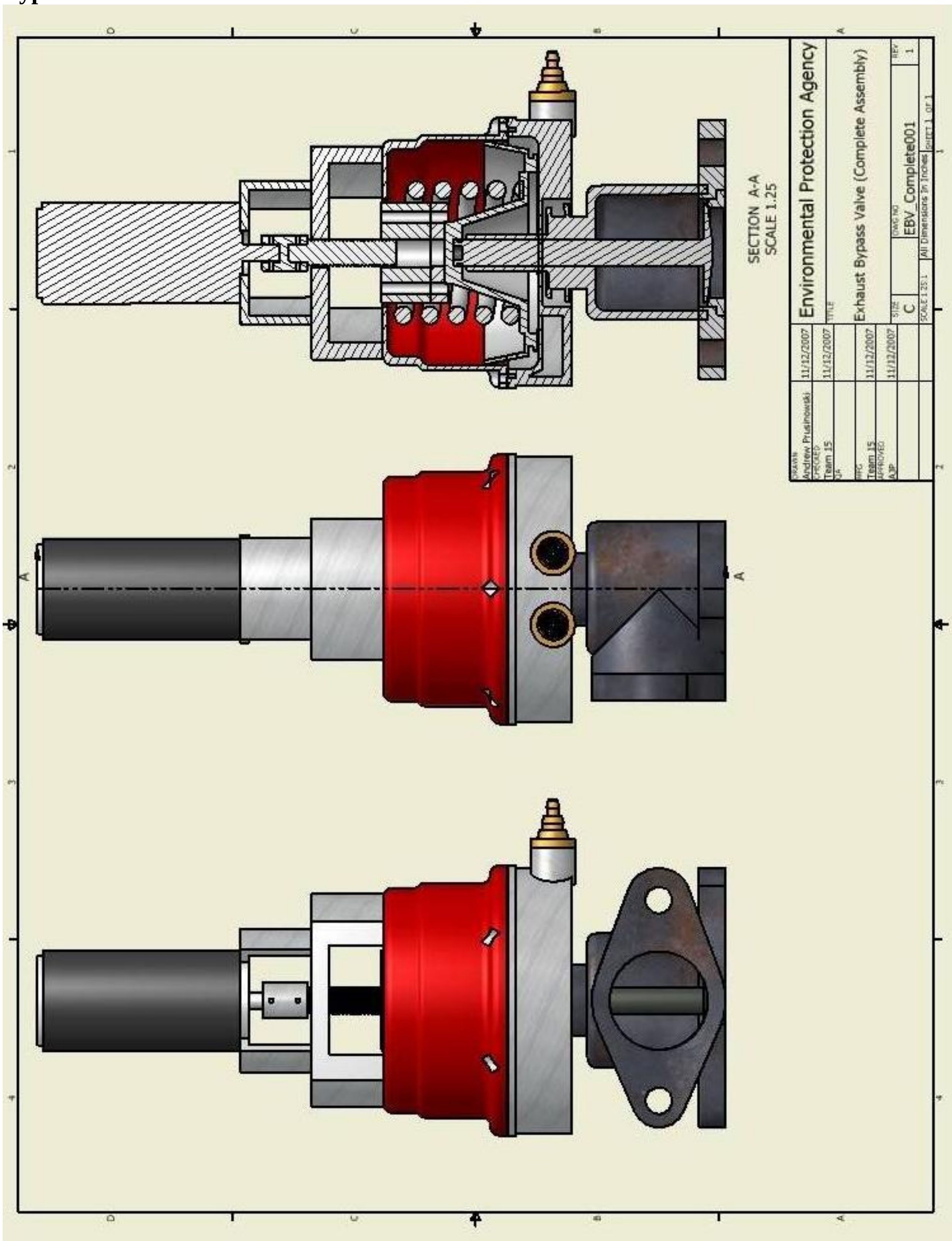
Conduction Heat Transfer:

$$\langle T_6 \rangle = T_4 + \langle Q_u \rangle \left(\frac{A_k k_{coolant}}{L}\right) = 103°C$$

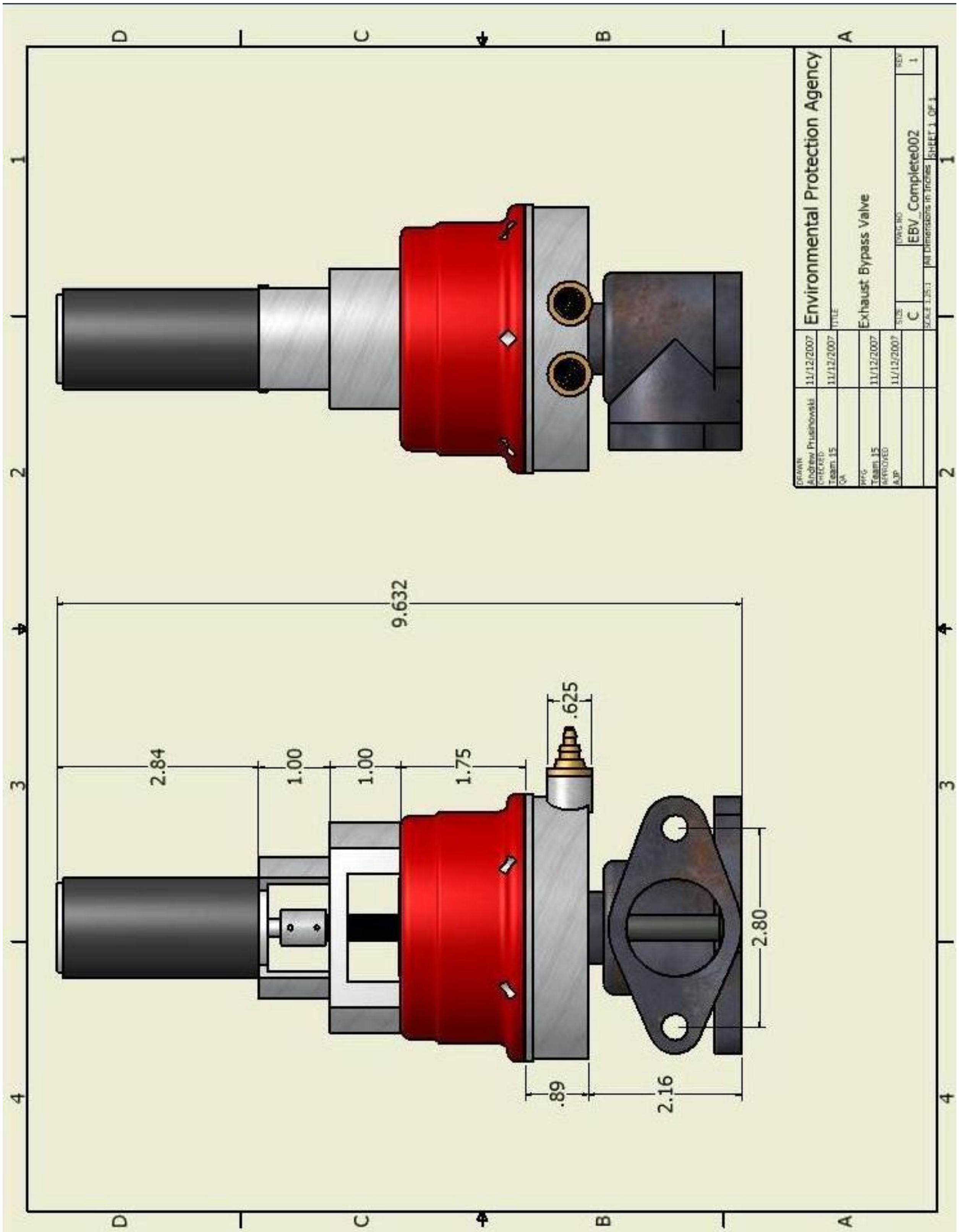
Appendix I: Tial 38 mm Wastegate Technical Drawing [10]



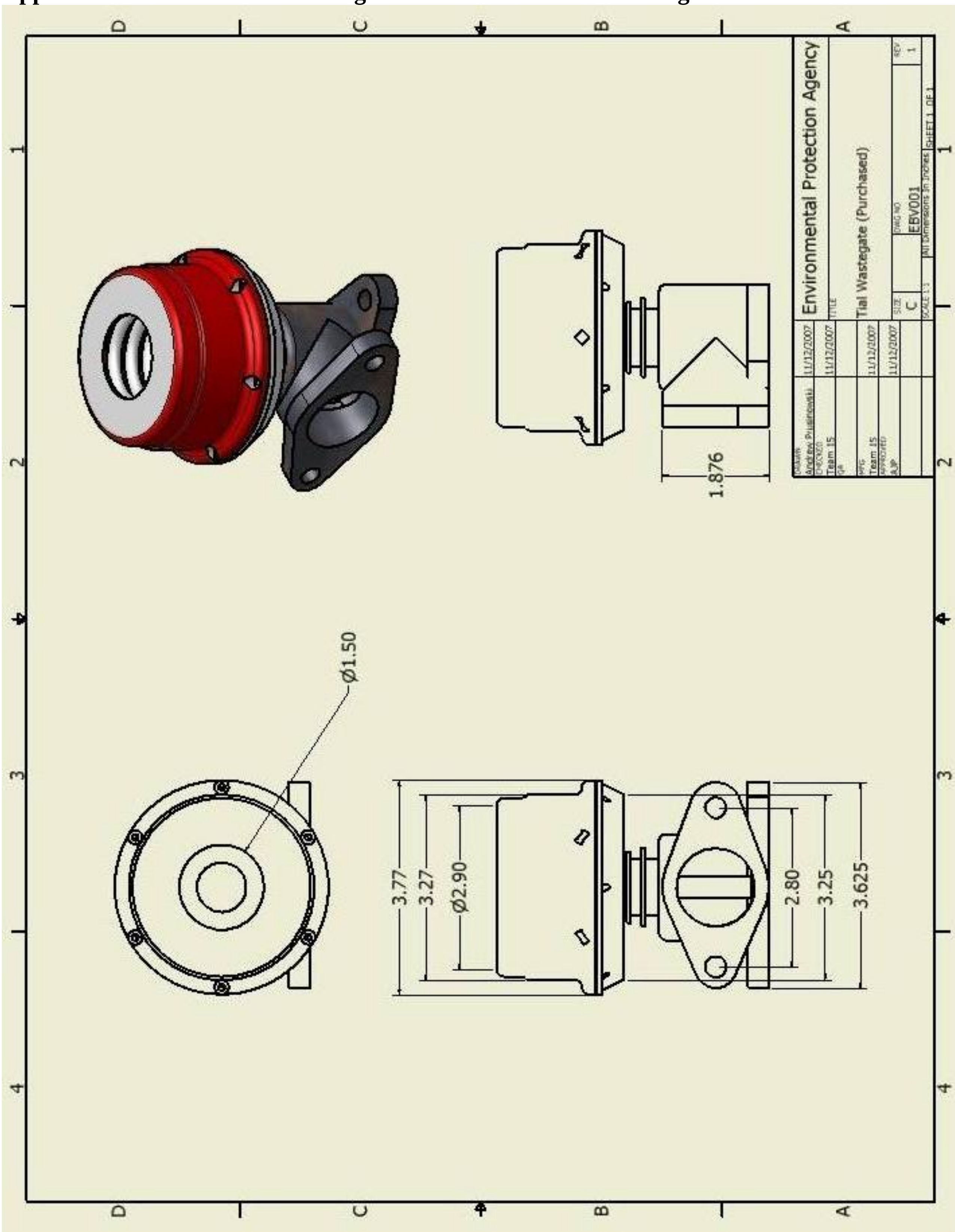
Appendix J: CAD Model/Cross Sectional View of Motor Controlled Poppet Exhaust Bypass Valve



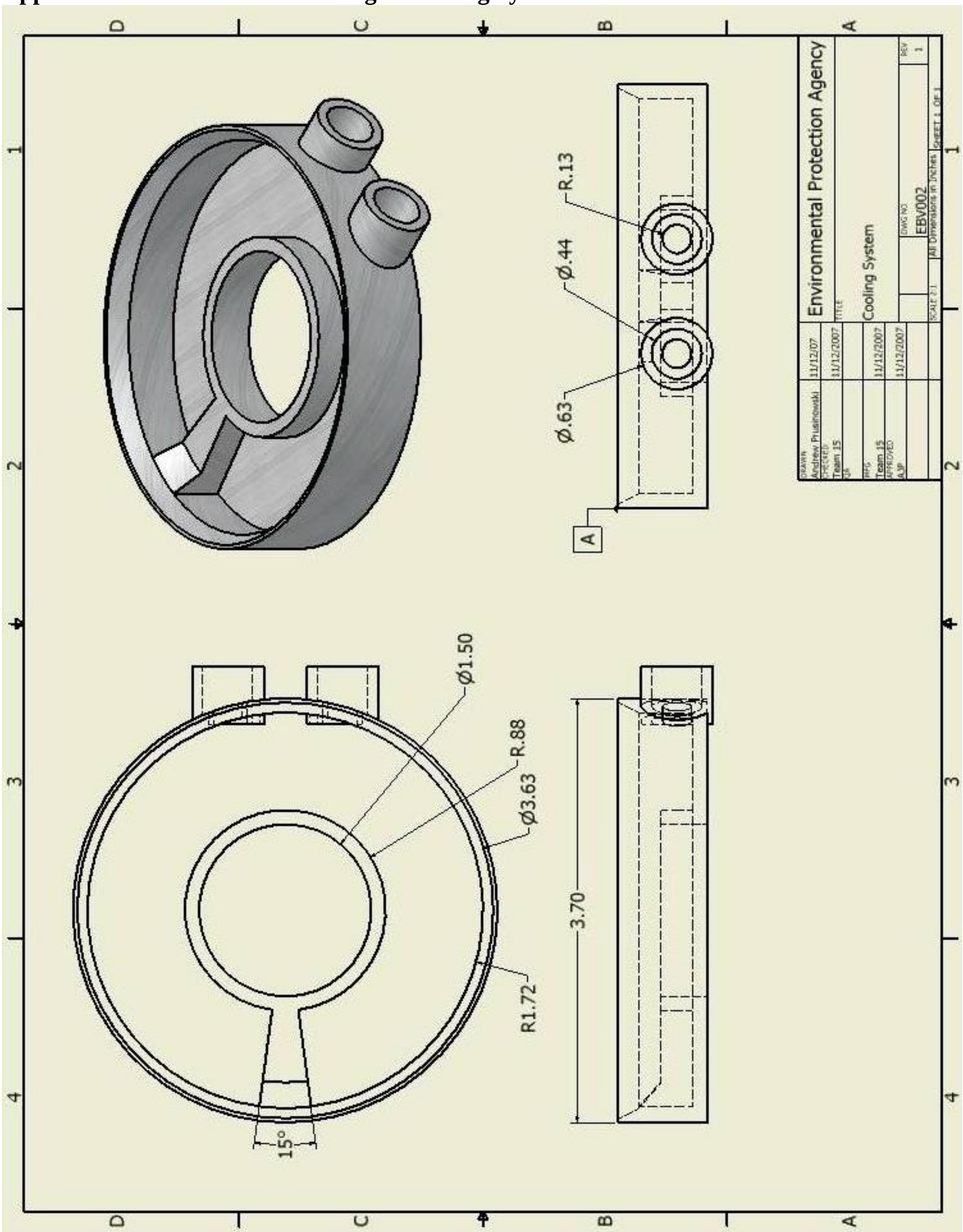
Appendix K: Dimensioned Drawing of Motor Controlled Poppet Exhaust Bypass Valve



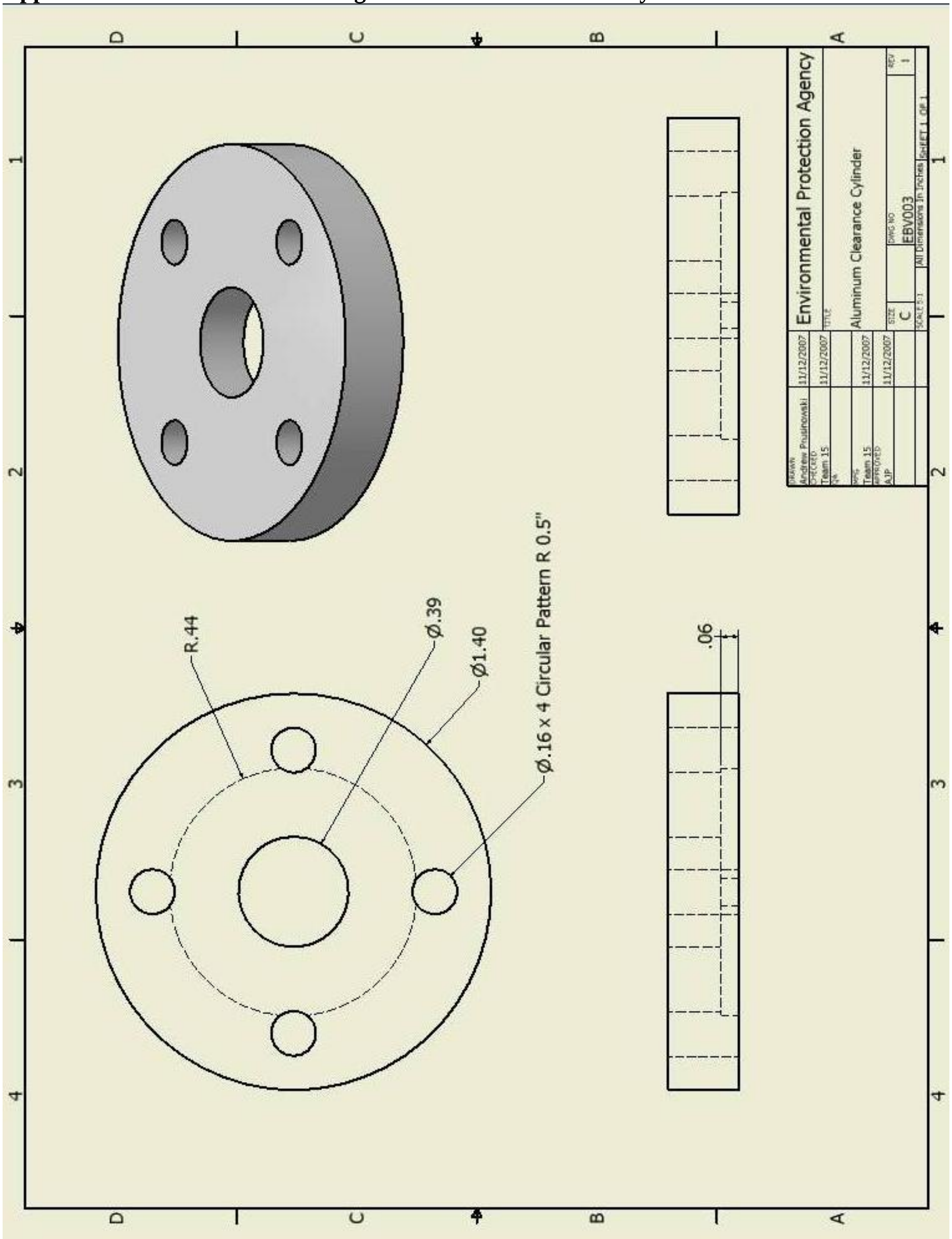
Appendix L: Dimensioned Drawing of Modified Tial 38mm Wastegate



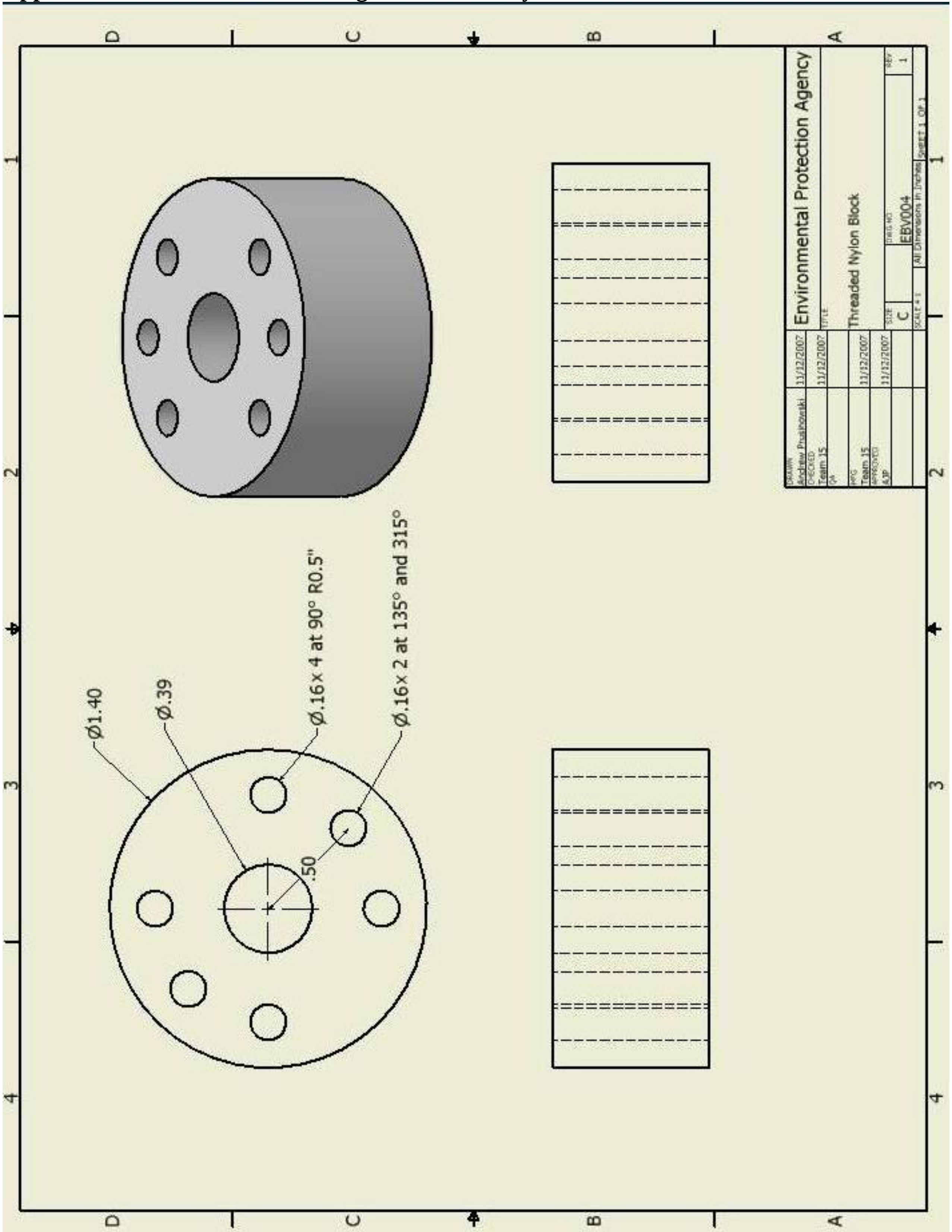
Appendix M: Dimensioned Drawing of Cooling System



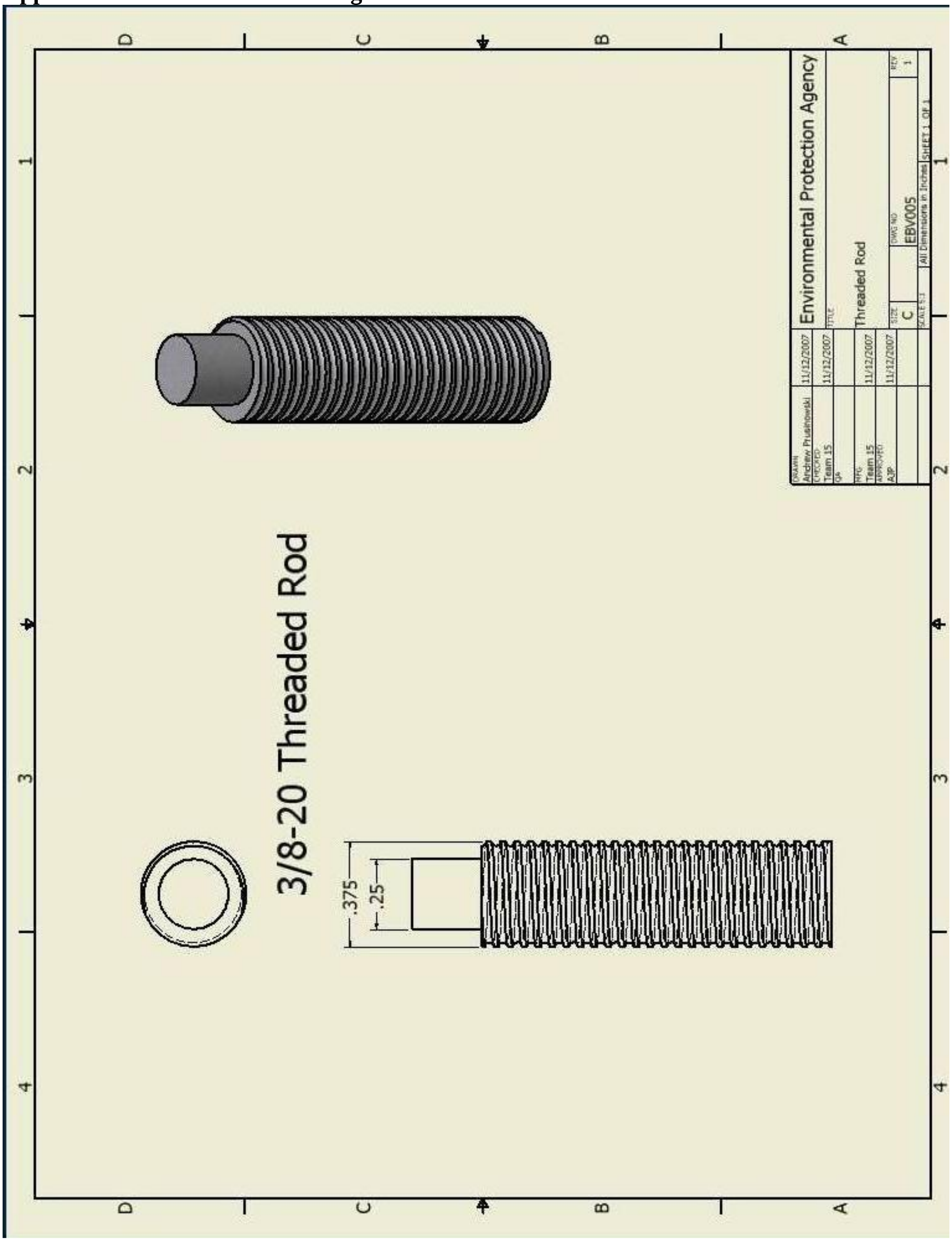
Appendix N: Dimensioned Drawing of Aluminum Clearance Cylinder



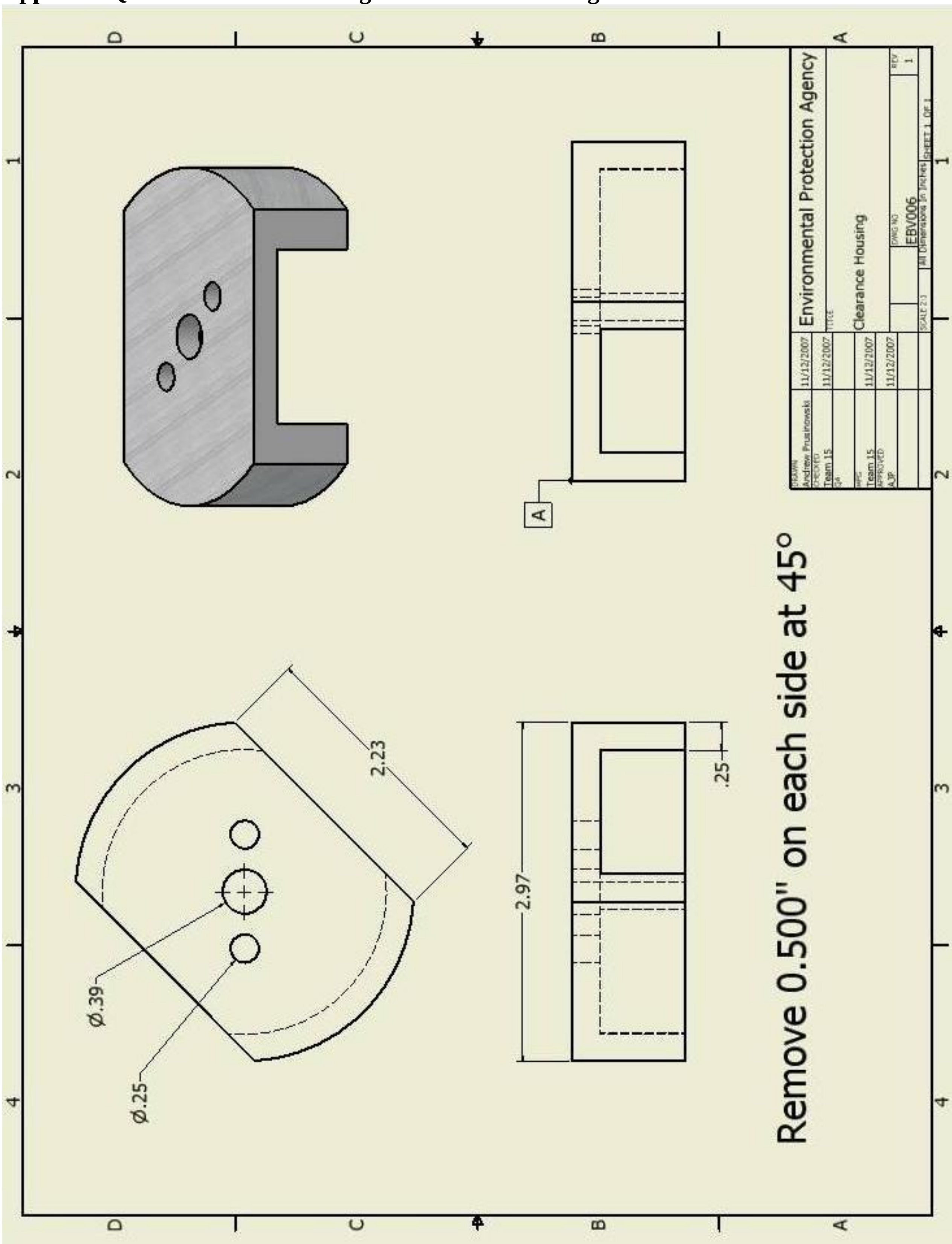
Appendix O: Dimensioned Drawing of Threaded Nylon Block



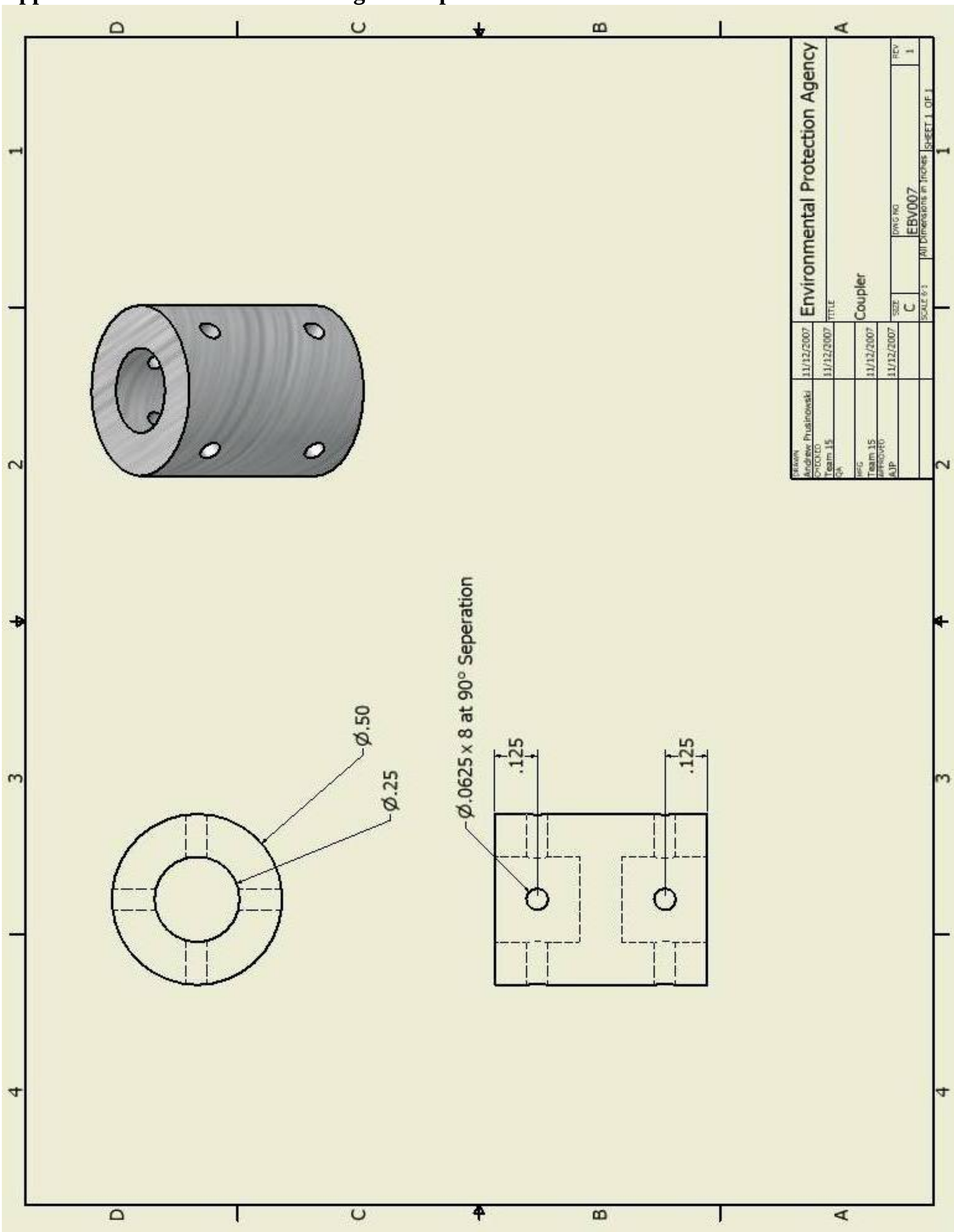
Appendix P: Dimensioned Drawing of Threaded Rod



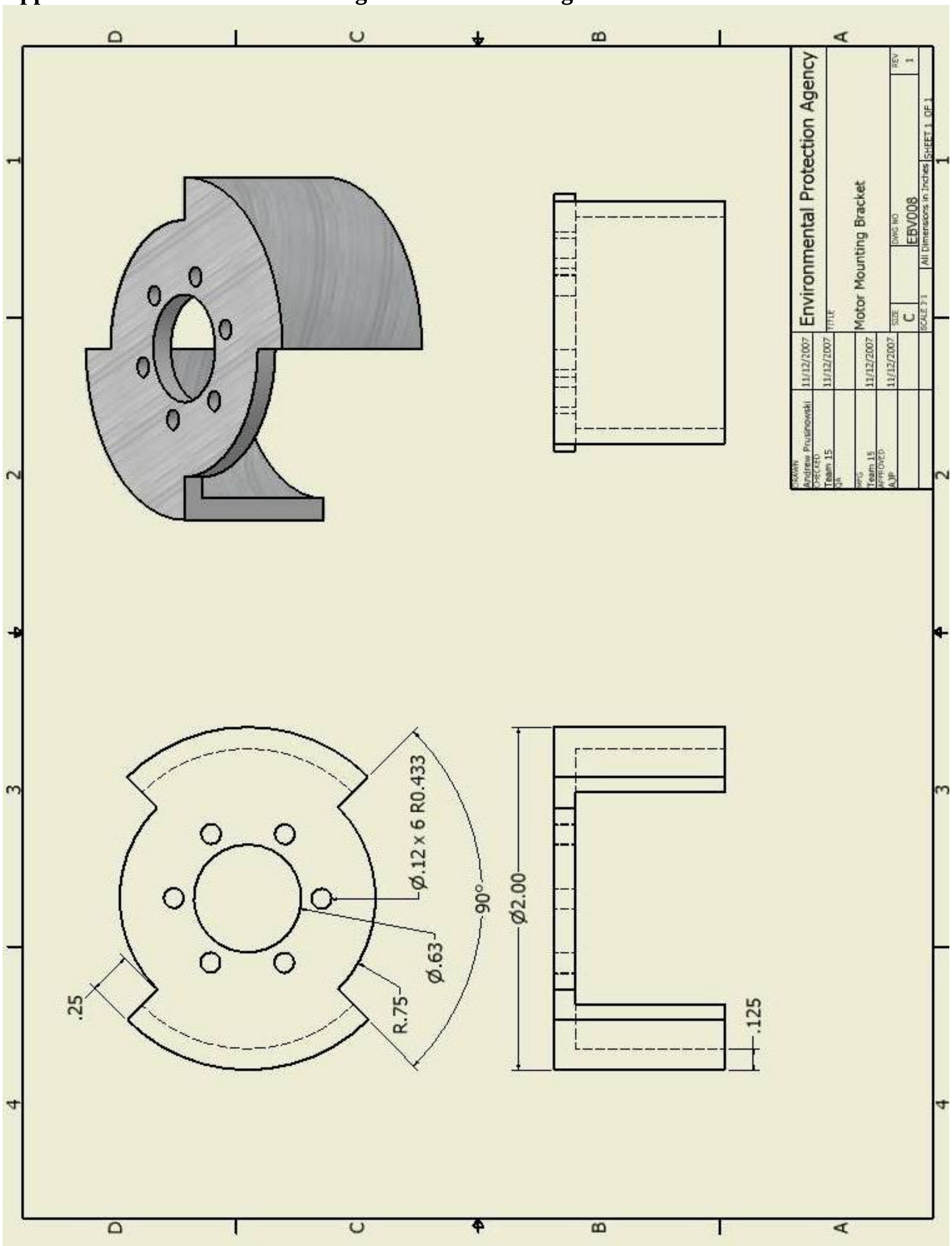
Appendix Q: Dimensioned Drawing of Clearance Housing



Appendix R: Dimensioned Drawing of Coupler



Appendix S: Dimensioned Drawing of Motor Mounting Bracket



Appendix T: Prototype Bill of Materials

Quantity	Part Description	Purchased From	Part Number	Unit Price	Total Price
1	Tial 38 mm Wastgate	Import Evolution		$219.00	$240.15
1	Maxon RE 40 DC Motor	Environmental Protection Agency		$0.00	$0.00
1	Maxon RE 40 DC Motor Encoder	Environmental Protection Agency		$0.00	$0.00
1	2" Diameter 2" Long Aluminum Round Stock	University of Michigan		$0.00	$0.00
1	1-1/2" Diameter 2 " Long Aluminum Round Stock	University of Michigan		$0.00	$0.00
1	1" Diameter 1" Long Aluminum Round Stock	University of Michigan		$0.00	$0.00
4	10-32 Stainless Steel Set Screws	University of Michigan		$0.00	$0.00
1	4 ft length 0.032" Stainless Safety wire	University of Michigan		$0.00	$0.00
1	12" by 12" by 0.01" Aluminum Sheet	University of Michigan		$0.00	$0.00
1	4"x 4"x 3/8" Steel Sheet	University of Michigan		$0.00	$0.00
1	1-1/2" Diameter 1ft Long Tubing	University of Michigan		$0.00	$0.00
1	1 oz. White Lithium Grease	University of Michigan		$0.00	$0.00
2	Three pole 15 amp switches	University of Michigan		$0.00	$0.00
1	3 ft. mulit strand wire 14 awg	University of Michigan		$0.00	$0.00
1	3/8"-24 Stainless Steel Threaded Rod	McMaster-Carr	95412A659	$5.00	$5.00
2	1/4" Diameter 1" Long Stainless Steel Shoulder Bolts	McMaster-Carr	90298A542	$1.73	$1.73
1	1-1/2" Diameter 1 ft Long Kevlar/Nylon Round Stock	McMaster-Carr	8502K48	$37.36	$37.36
4	10-32 Cap screws 1" Long	McMaster-Carr	90152A261	$7.25	$7.25
1	1/8" pipe fitting plug	McMaster-Carr	4464K561	$0.50	$0.50
2	Barbed Male 1/4" Pipe Brass Hose Fittings	McMaster-Carr	5346K18	$8.41	$16.82
6	M3 8mm Long Socket Cap Screws	McMaster-Carr	91290A113	$4.32	$4.32
				Total $	$313.13