Midterm 1 Report

Super Seal: Adding a 2nd Stage Sealing Device To Recapture Oil From Seal Leaks

Team 1 Members:

Project Lead: Christian Milione – cam11w@my.fsu.edu

Lead ME: Kyle Brooks – kwb13c@my.fsu.edu

Financial Advisor: Jonathan Strickland – jbs12d@my.fsu.edu

Research/Chronicler: Olaniyi Ogunbanwo – olaniyi1.ogunbanwo@famu.edu

Faculty Advisor

Dr. William Oates

Sponsor

Terry Shaw, Cummins Technical Center

Instructor

Dr. Nikhil Gupta

Date Submitted:

October 21, 2016

Table of Contents

Table of Figures

ABSTRACT

The goal of the Cummins Inc. sponsored project is to design a device that will capture oil that leaks out of a crankshaft seal. The collected oil is to be transferred to a holding reservoir enabling future reinsertion into the crankcase. This objective will be obtained through the application of various engineering design methods, coupled with collaboration between the college and industry appointed sponsors. The overall effectiveness of this device will be assessed through a 24-hour simulation across various operating regimes proposed by Cummins, and carried out through a custom test rig for which the group must also design and fabricate. Team 1 has done preliminary research into the problem to better understand the gap between the current and ideal situation, as well as the concept of the product needed to achieve success with the objective.

ACKNOWLEDGEMENTS

The group would like to make a point to thank Dr. Shih for his hard work in gathering the available projects for the entire semester. In addition, an acknowledgement must be made for the course instructor, Dr. Gupta, seeing as how he has been (and will continue to be) the main source of information pertinent to the course. Furthermore, a note of gratitude would like to be made for the sponsor engineer, Terry Shaw, for his time and effort spent with the team thus far.

1. Introduction

Due to the ever-increasing demand of goods and services, modern powertrain systems used for their conveyance are being repeatedly held to higher and higher operational standards. As such, the schedule for routine engine overhauls are being delayed to the point where the industry is beginning to push the boundaries of what is currently possible. To illustrate this, current diesel powered semi-trucks being used to deliver automotive parts are expected to adhere to a life of 30,000 hours of operation before a rebuild is necessary. [1] If that 30,000 hours target is assumed to be nonstop, that is just under 3.5 years of continuous function. It is important to note that performance for that duration without any stopping is highly unrealistic in this application, but rather was used as to help gauge just how high the target for modern machinery is. To meet these inflating targets, innovations in design and manufacturing are, and will continue to be, necessary.

As with everything, the primary driving factor is cost. Engine overhauls across the wide range of models in current production yield extremely high numbers when it comes to repairs. For example, the replacement of a rear crank seal for the Cummins 95 liter locomotive engine is $21,000. [2] This is due to the extreme amount of labor involved; these types of engines have such little clearance between parts, that to access anything that is not located directly on the exterior requires a large amount of component disassembly merely to access what needs replacing. If this repair cost is then coupled with the opportunity cost of lost productivity, the expenses for this seal failure begin to accumulate rapidly. This notion can be extrapolated to most large-scale power generation systems in use today. Ultimately, fixing issues as they fail is not an option for the companies that operate them.

With this in mind, the prevention of premature component failure is highly important. Not to mention, the concept of what is considered to be a “failure” has evolved over the years. In modern times, customers are becoming more sensitive to what is perceived as a failure. A large segment of this sensitivity increase concerns leaks. Historically, leaks were tracked through dripping, spraying, or fluid actively running from whichever aspect of the motor it was contained in.

However, current customers perceive a leak as an instance when a component is not “dusty dry”. [2] Interestingly enough, the customer is technically correct with this assertion.

Therefore, to continue to align themselves with increasing customer expectations for both engine performance and evolving classifications of failure, Cummins Inc. (Cummins) is looking for a method to recapture oil that has managed to work its way past the rear seal of a crankshaft. This oil is to be collected as thoroughly as possible and deposited into an accessible reservoir where it will remain until a powertrain technician can reinsert the oil back into the crankcase.

2. Background

2.1 The Role Of Engine Oil

Simply put, engine oil is the lifeblood of a motor. Most notably, engine oil serves the main function of providing lubrication between components of the motor as it performs under high pressures and temperatures. By reducing friction, and therefore wear, motor oil aids in prolonging the life of vital parts.

However, advances in technology have provided the opportunity for oil to do more than just lubricate parts through their additives. Modern motor oil removes dirt and debris buildup within the engine and suspends it within the fluid until it can be removed during the next oil change; effectively working to clean your engine. Furthermore, todays oil supports increased fuel economy, protects the vehicles emissions system, works to assist with hydraulic operations for variable valve timing (VVT), and maintains components at cooler temperatures, all while possessing the ability to perform these task across a extensive range of temperature spectrums. [3]

2.1.1 Why is Engine Oil Especially Important Today?

Consider the trend of engine design in today's market. To meet rising demand for better fuel economy, original equipment manufacturers (OEM) are transitioning into motors with smaller and smaller volumetric output. However, this typically comes with the sacrifice of power. To offset that sacrifice, turbochargers are commonly installed. The shaft inside of said turbocharger can rotate at speeds of up to 200,000 revolutions per minute. As a result, it is imperative that oil can reach these components quickly, while providing an effective form of lubrication and cleaning. Otherwise, the entire system fails, producing an inability to reach OEM targets and leaving the customer with costly repairs. [4] Figure 1 displays a photograph of a turbocharger during operation.

Figure 1: Depiction of a turbocharger during operation. The intense heat generated can be seen by the color change.

2.2 Oil Leaks

The “go to” cause of an oil leak is the result of a failed gasket or seal, most commonly being an issue with the crankshaft seal. [5] Depending upon the operating conditions of the vehicle, the seal can dry rot, crack, break, tear, or some combination of them all. However, there are also instances in which oil can escape past seal that is in perfect working condition. For instance, during normal operation the oil within the crankcase becomes heated. While heated, no only can the oil convert into a mist, but the increased temperatures cause thermal expansion within the seal. After operation, the subsequent cooling process allows oil mist to discharge past the crankshaft seal as a result of differences in cooling rates between the two, often collecting along the underside of the vehicle. In either case (faulty or non-faulty seal), this is still considered to be a component failure due to the fact that the oil is no longer completely contained within the crankcase.

2.3 Current Recapture Methods

Research conducted indicates there is no widespread device with the sole purpose of recapturing oil that has leaked out of a vehicle. This statement (and research) applies to any mechanical device or piece of hardware that is fitted to a vehicle, either in the OEM or aftermarket segments. Rather, the main option is to use a drip pan to catch large oil leaks while the vehicle is stagnant for large periods of time (where possible).

Instead, the aim of current line of products is to prevent the oil from continuing to leak through the use of a “stop leak” additive/sealant. These products are design to be mixed with your engine oil and circulate throughout the motor. During their circulation, these additives seep into seals to try and recondition them back to their original characteristics: softer, more flexible, and proper shape. [6]

Ultimately, this means that there is a gap in the current market. If successful, this project could have a large impact to the current design, manufacturing, assembly, and repair processes. Additionally, it could save customers and technicians both time and money.

3. Project Scope

3.1 Need Statement

A current industry setback is that engine oil commonly leaks past the rear crankshaft seal. This particular failure is becoming more paramount in the eyes of the customer.

3.2 Goal Statement

“Design a device to capture leaking oil from a rotating test crankshaft and deposit it in a reservoir so that it can later be reintroduced to the crankcase. Additionally, a test rig for the device must also be fabricated in order to assess the functionality of the design.”

3.3 Objectives

- Design a capturing device to collect oil that has leaked from a rear crankshaft seal.
- Design a rig that can be used to test the recapture device in order to determine its overall effectiveness.
- Determine feasibility of each design with technical proof (calculations, drawings, etc.).
- Order, obtain, or manufacture components for each design.
- Construct the oil recapture device.
- Construct the test rig; ensuring it can adhere to the operating regimes given by Cummins.
- Perform the 24-hour trial, and assess overall project success.

3.4 Constraints

The project must adhere to multiple constraints. They include, but are not limited to:

- A budget of $2,000 is not to be exceeded.
- The oil recapture device must work with a Cummins OEM 165mm rear crank seal.
- The oil recapture device must be able to hold oil that has been heated to 125°C.
- The oil collected must be deposited in a reservoir that can be emptied by a technician.
- The oil recapture device must fit in the space between the rear crank seal and flywheel housing.

In addition to the constraints of the recapture device, the rig with which it will be tested be designed around some constraints. These include but are not limited to:

- Rig must be able to operate continuously at variable speeds between 500 and 2000 RPM.
- Rig must rotate the appropriate size “crankshaft” across the provided operating regimes.
- Test fixture must be enclosed and properly grounded to ensure safe operation.

- Rig must be able to shutdown from full operation without any issues.

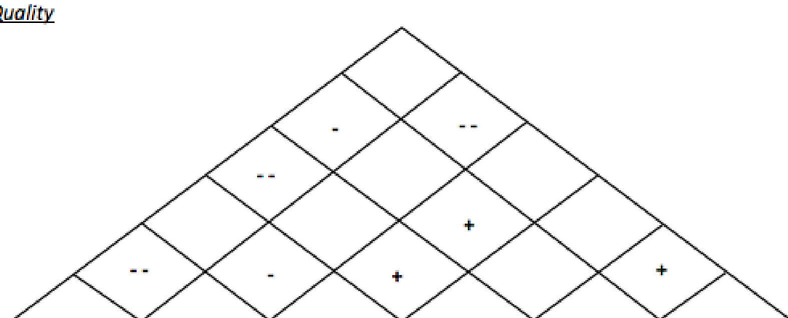
3.5 House of Quality

To help address the need statement, and fulfill the goals and objectives set forth by the team, a house of quality was constructed as a basic tool to help guide the team towards important engineering characteristics that will be vital to the design of a solution.

Please reference Figure 2 below for the House of Quality that has been generated for this project.

Super Seal: House of Quality

Roof Correlations	
+ +	Strong Positive
+	Positive
-	Negative
- -	Strong Negative



Customer Requirements	C.I. (1-5)	Size	Cost	Durability	Weight	Versitility	Efficiency
Device/Rig Operate Safely	5				3		1
Device Fits Size Constraint	4	5	3		3	3	3
Device Collects More Oil Than Surroundings	5	2		3		2	5
Device Operates in "Normal" Regimes	3		3	4		3	3
Design/Fabrication of Test Rig	5	3	4	2		3	
Collected Oil Can be Returned Into Engine	3	4	3	3		3	4
		1 - Low Correlation			*5 - High Correlation*		
Calculations	**Score**	57	50	46	27	55	63
	Rank	**2**	**4**	**5**	**6**	**3**	**1**

Figure 2: House of Quality for the project.

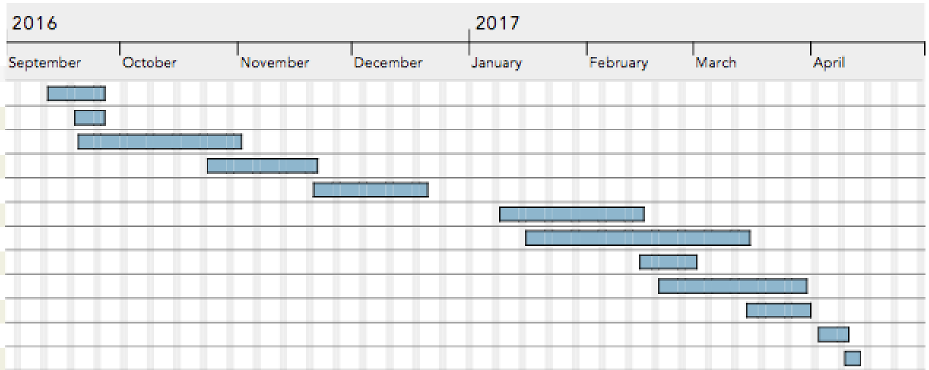
4. Methodology

For this project, there are a few different distinct phases. Each stage serves its own purpose in order to develop a quality product that will meet all of Cummins requirements. The first stage of this project required continued research into the problem that helped us gain a better understanding of how the gap between the current and ideal condition arises. In this stage it was important to work directly with the sponsor engineer, Terry Shaw, to learn from his experience and expertise regarding this system that the group was asked to address. Furthermore, benchmarking proved to be a key element in order to help ensure that Team 1 measured up well to other competitors, as well as ensuring that no time was lost on a portion of the project that someone else has already solved. Once sufficient knowledge was gained on the problem and system/components, the secondary phase was to then begin designing the two vital components to this entire project: the oil recapture device that was to be the solution of the problem, as well as a rig that was used to test the solutions ability to perform across a range of operating regimes provided by Cummins. It was important to keep a keen eye on time management for both aspects of the project in order to deliver a quality product in a timely manner. In addition, with the project sponsor being located so far away, effective information sharing and various communication between the group and sponsor was vital, and was performed as efficiently as possible. This will help the project flow more efficiently throughout the year. The design phase of the project proved to be the most challenging, yet rewarding portion of the project. This is where the background research and knowledge gained from the sponsor are put into motion. An important note to make is that the design of our solution and test rig should be developed using concurrent engineering, in an effort to minimize any lack of correlation between the two. Once viable designs were selected, the necessary parts were ordered as soon as possible to give the supplier a reasonable amount of time to process the request (and make any adjustments if need be). A close watch was kept during this phase in order to ensure that the budget of $2,000 was not exceeded. Once all parts were ordered or machined, the actual construction of the project begin. It will be extremely important that this stage not be taken lightly, as any shortcoming or overlooked detail could result in a failure of the entire system. Once the construction phase was complete, the final stage of the project is to test the device under operating conditions to which Cummins has provided so that the overall effectiveness of the design can be analyzed.

4.1 Project Schedule

Figure 3 below represents the schedule planned for the project.

Name	Begin date	End date
General Research	9/12/16	9/26/16
Needs Assessment	9/19/16	9/26/16
Conceptual Design	9/20/16	11/1/16
CAD Renderings	10/24/16	11/21/16
Finalize concepts	11/21/16	12/20/16
Finalize BOM	1/9/17	2/15/17
Order Parts	1/16/17	3/15/17
Air flow/Seal DFM	2/15/17	3/1/17
Assemble Test Rig	2/20/17	3/30/17
Performance Trials Test ...	3/15/17	3/31/17
Test Sealing Solution	4/3/17	4/10/17
Analyze results/ Adjustm...	4/10/17	4/13/17



The background research began in early September. As team one dove further into researching the problem a need statement was conceived. This was the spring board for the team to launch the conceptual design of both the test rig and the oil recapture system. Throughout the fall many conceptual designs were developed and improved. Through this iterative process a test rig design was completed. The sealing system conceptual design was not completed in the allotted amount of time. Throughout the spring the sealing system design was continually redesigned. Team 1 encounter substantial difficulty in finding available materials at low quantities that could be machined properly. It was not until march that the final design for the system due to these challenges. The bulk of the bill of materials was completed by mid-February. Again several components required significant machining. The materials needed for the custom components were difficult to find which added more time to the process. The materials for the seal was found and purchased in late March. The team began assembling the test rig as the shop finished machining the parts. This process began in the month of March and continued until mid-April. The testing and analysis of results transpired in the third week of April.

5. Conceptual Design

With an overall idea of the problem at hand, as well as the established goal and objectives to address it, initial brainstorming activity commenced. The objective of this session was to begin to piece together potential solutions through the use of the engineering characteristics dictated to be the most crucial through the house of quality: efficiency, size, and durability.

5.1 Current Assembly

The motor to which the design will be initially applied is the Cummins ISX 15; a 15-liter inline 6-cylinder diesel combustion engine capable of producing up to 600 horsepower. With the aid of the liaison engineer from Cummins, technical drawings of this particular power plant were acquired to provide a visual for reference due to the lack of physical motor observe. Please reference Figure 4 and 5 below for these drawings. [1]

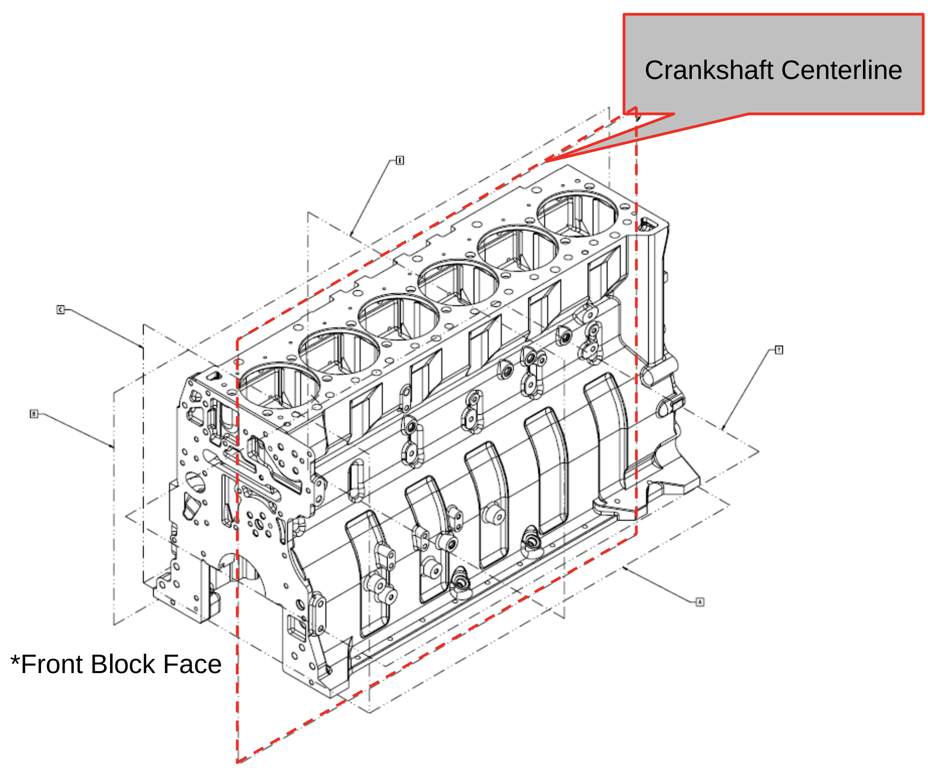


Figure 4: Cummins, Inc. ISX 15 engine block. [1]

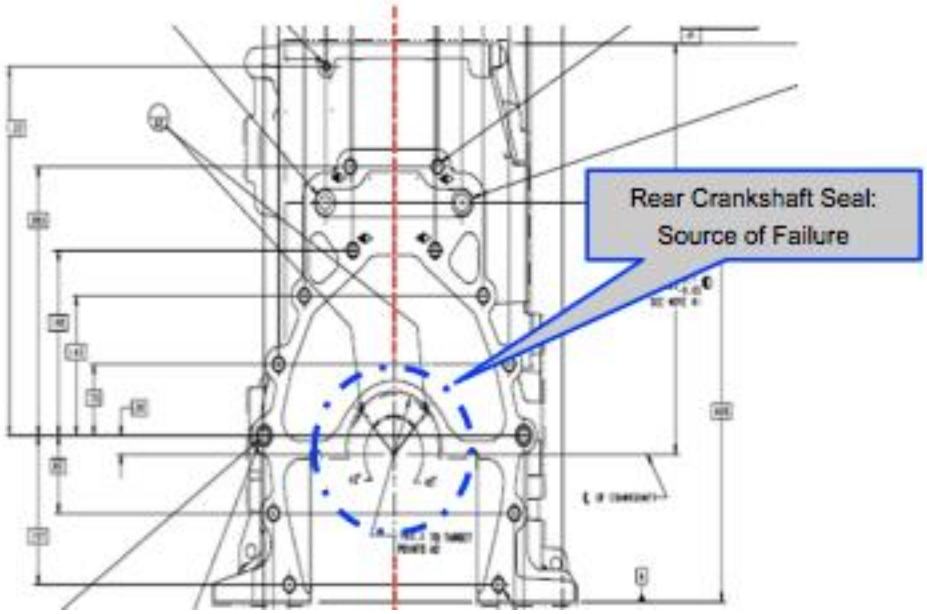


Figure 5: Cummins, Inc. ISX 15 engine block. Rear face showing where the crankshaft exist the engine block. [1]

Here, Figure 5 represents an orientation of the block showing the rear face where the crankshaft exits the block and bolts to the flywheel. The blue circle and callout represent the location of the current rear crankshaft seal (primary seal). This area also represents the area in which the secondary sealing device to be designed will reside. Diving deeper into the current assembly, a cross-section view of the primary seal assembly was also provided. This can be seen in Figure 6 below.

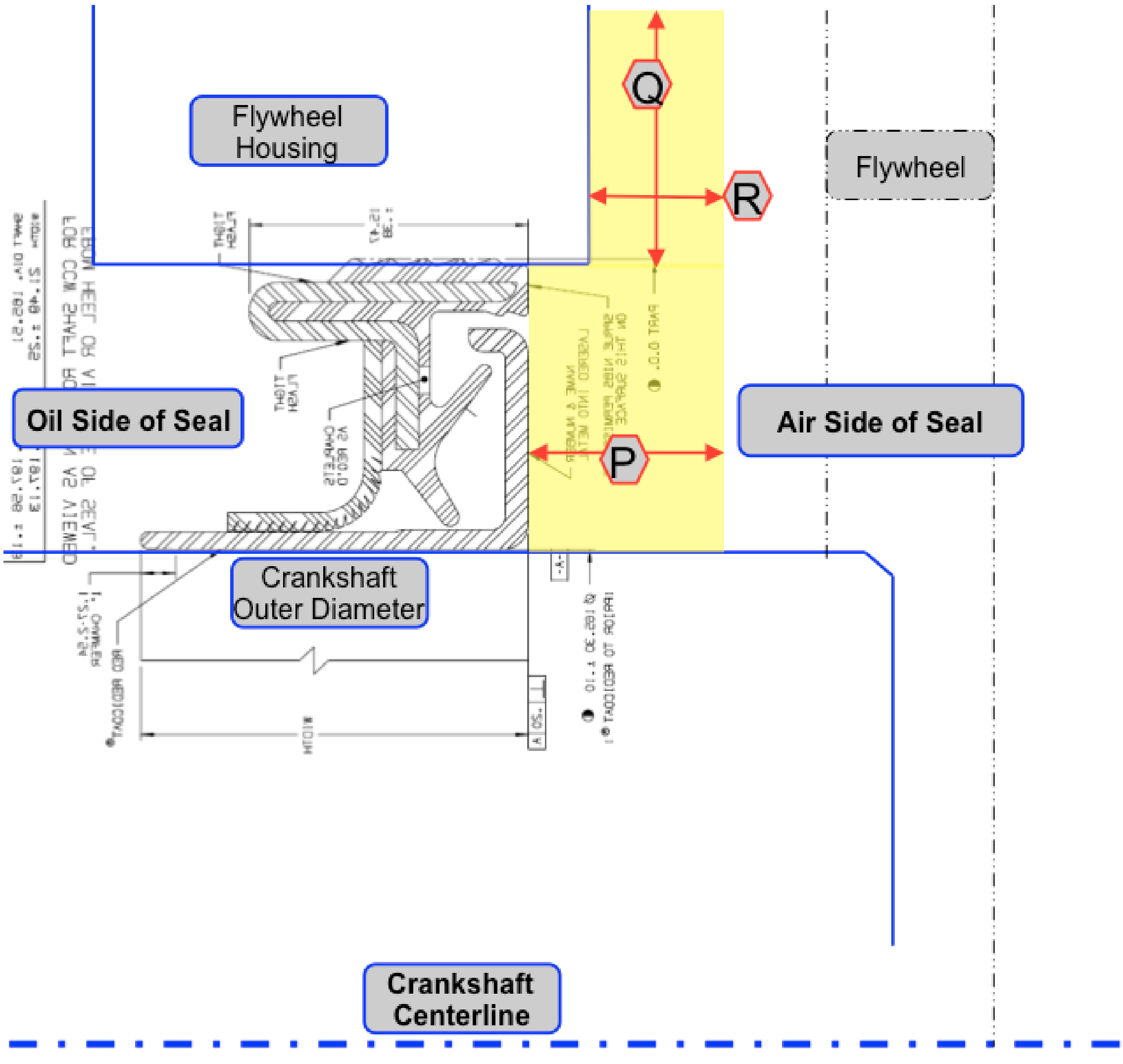


Figure 6: Schematic of rear crankshaft seal and its adjacent components. [2]

Figure 6 is especially helpful, not only for the visual it provides of the seal assembly, but particularly in the fact that it references the available dimensions between the crankshaft, seal, flywheel, and flywheel housing. Ultimately, this area is the vacant space discussed within the constraints in which the concept generated through this project will have to reside, and operate in. To assist in the understanding of these tight spatial constraints, the ‘Q’, ‘P’, and ‘R’ axes have a maximum dimension of 25mm, 25mm, and 15mm, respectively. To further challenge things, the project sponsor has requested that these parameters be minimized closer to that of 10mm, 15mm, and 10mm, respectively.

5.1.1 ISX 15 Component Materials

The materials used here are important in the sense that they will either need to be mimicked through the final project assembly, or improved upon if deemed beneficial to the overall goal. According to our liaison engineer, Cummins uses a gray cast iron for the engine block of the ISX 15, a crankshaft made of 4140 steel with an unhardened surface, and finally a seal composed of Teflon that is bonded to rubber surrounding the mild steel inner body. This seal also has ribs imbedded into it to aid in its sealing abilities. [1]

5.2 Macroscopic Ideation

Keeping in mind that the secondary sealing device generated must be proven through the fabrication of a test rig, the notion of concurrent engineering plays a large factor. If the two are developed hand-in-hand, the risks of future issues brought upon by any gaps between the two can be mitigated. From a macroscopic point of view, an idea of what the overall project will look like can be seen below in Figure 7.

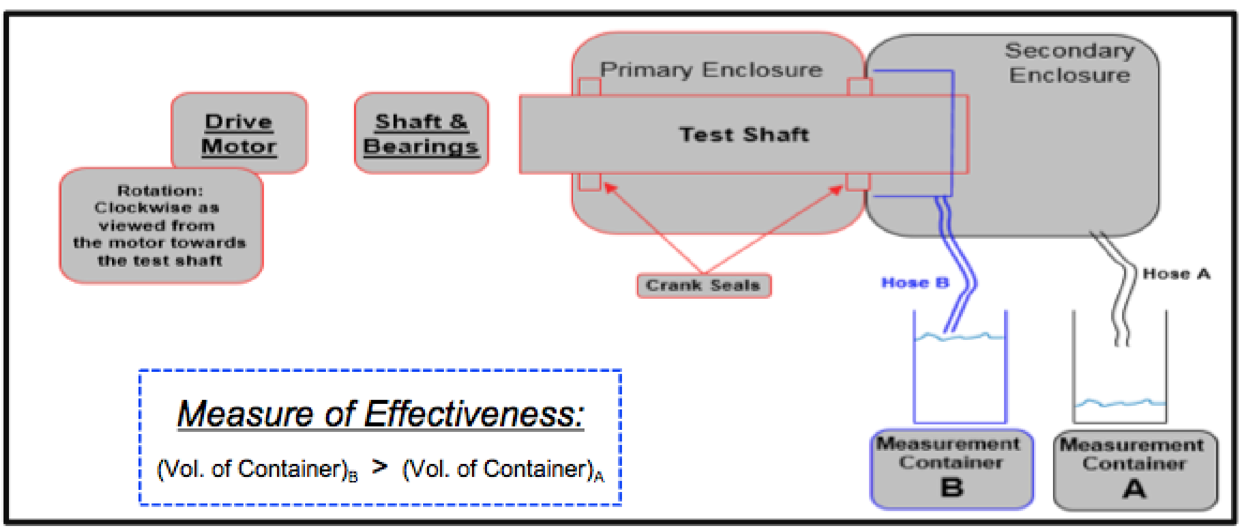


Figure 7: Pictorial representation of the overall project at this time. [2]

Referencing Figure 7, the red portions represent the test rig components of the final assembly. Initially, the drive motor was proposed as a small internal combustion engine. However, due to the challenge of determining exact revolutions per minute (rpm) required for testing purposes, a plug in electric motor was proposed. This provides the opportunity to adjust rpm more accurately and easily through the use of a potentiometer. [7] Technically, components such as the bearings will need to be theoretically calculated for their properties relative to cost, to ensure that the materials selected are both sufficiently strong and cost effective. In this instance, the primary enclosure represents the engine block for where the testing oil will be placed. The test shaft selected will be of the same material as the current production crankshaft in the ISX 15, 4140 steel with an unhardened surface. In addition, the test shaft will be the same diameter as the current production crankshaft at 165mm. Both of these parameters have been chosen to try and mimic the current production assembly as much as possible, effectively closing the gap between our design prototype and the current production assembly. If promising results are attained, this should make the transition from prototype to production component smoother. As such, the idea moving forward is that any opportunity within the project where reverse engineering is present to be taken advantage of to aid this transition.

Within Figure 7, the blue section is representative of the device that will be designed and fabricated. Per customer requirements, this device will be attached to some sort of drain line feeding into measurement container ‘B’ (the reservoir).

Moving farther right in the Figure, the black segment pertains to the secondary enclosure required to have an empirical control within the testing stages. This secondary enclosure will not only seal the device from the surrounding environment, but also provide a benchmark for the impact that the implemented secondary seal has on the system by capturing any oil that manages to leak past both the primary and secondary sealing methods. This escaped oil will also be drained into its own reservoir, measurement container ‘A’, for measure.

Ideally, the volume of measurement container ‘B’ is to be greater than the volume of measurement container ‘A’, with the ultimate goal of maximizing the amount of oil in container ‘B’ to as close to 100 percent as possible.

5.3 Design Work for Test Rig

After taking several factors into consideration including the necessity of shaft straightness within the test rig to avoid wallowing out the seals, the team conducted several brain storming sessions. The team began highlighting key features and components necessary for the sponsor given tests.

Figure 8 shows a depiction of the test rig with all the major components.

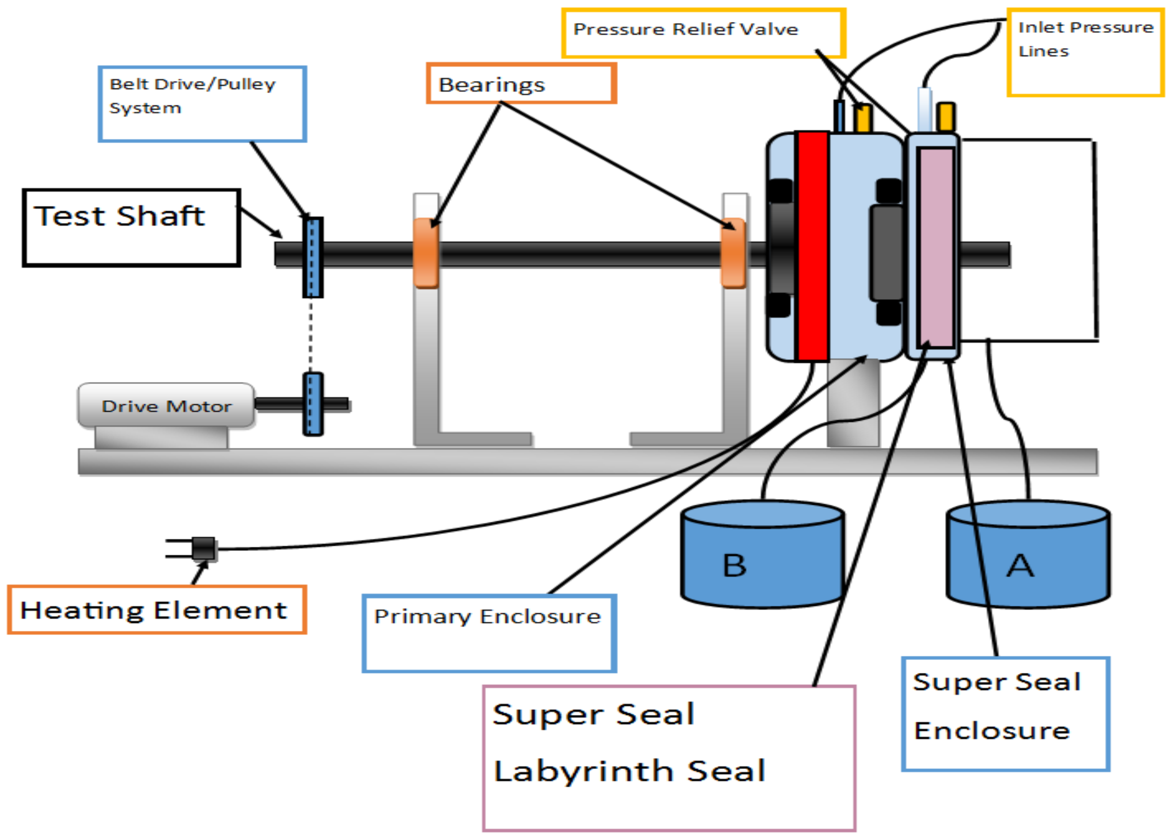


Figure 8. A macroscopic ideation of the test rig

A motor of would be fixed to a shaft by a pulley belt system sized to attain the desired speeds. This shaft would be supported by a set of pillow block bearings. The test shaft would press fit inside the front seal with inner diameter of 140mm as well as press fit into the larger 165mm rear seal supplied by Cummins Inc.. The test rig needed a mock “crankcase: as shown in figure 7 by the light blue region. The oil within the crankcase must be heated to 125 degrees Celsius. A heater was needed. This heater is depicted in the red area shown in figure 7. Beyond the crankcase a super seal enclosure was needed to recapture the oil that leaked past the rear crankshaft seal. Both the crankcase and the super seal enclosure required pressurization for purpose of realistic test simulation as well as functionality.

To power the test rig a 1/3 HP Dayton variable speed electric motor was selected as shown in figure 9.

Figure 9 Depiction of the Dayton motor employed during testing

This motor runs at 1475 RPM when connected to a voltage supply of 120 volts. A v-belt pulley system was utilized to achieve shaft speeds of 500 and 2000 RPM. A pair of 5200 series pillow block bearings were chosen to support and align our test shaft. Our test shaft was a custom component that required machining.

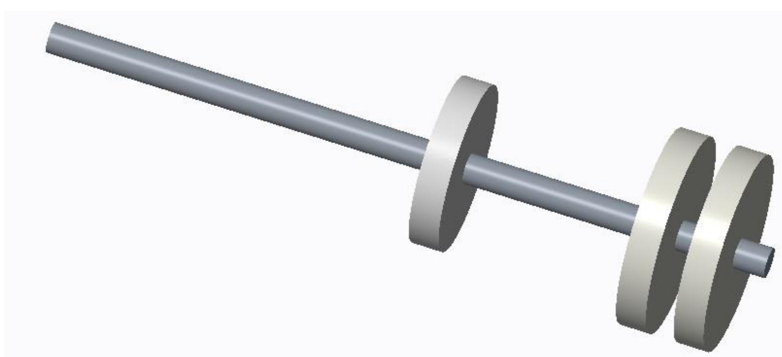


Figure 10. A cad depiction of the test shaft

The test shaft was a section of round bar with a diameter of 25 mm. Three steel disks were press fit and tack welded on to the shaft to form the inner surfaces for the front seal, rear seal and the super seal which will be further discuss in the following paragraphs. The steel flange the rear seal mounted over had an OD of 140 mm. The flange the rear seal and the super seal mounted over were both 165 mm OD. The crankcase was constructed from an 8 inch schedule 40 steel pipe. A set of caps were welded onto the ends of the section of pipe to have the seals press fit inside such that the rear, the front and the super seal will rest between the steel flanges and the caps. Figure 11 shows the crankcase with the caps welded on either side.

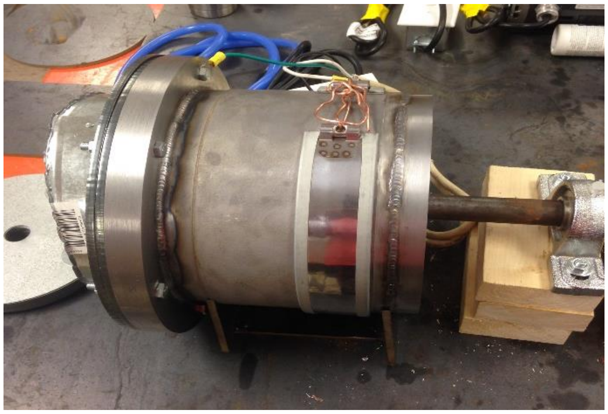
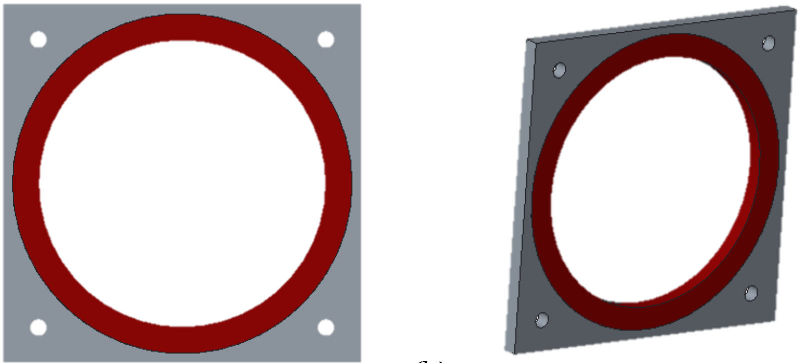


Figure 11. The crankcase with the caps welded on the ends

5.3.1 Preliminary Design Work for Sealing System

After some discussion and brainstorming, initial countermeasure designs began to take shape, each compounding off of one another through an evolutionary pattern to address concerns with the previous concept. For example, the initial concept generated can be seen below in Figure 8. This design is a 'bare bones' concept, providing the most basic platform to build off of and facilitate progress. It consists of a simple plate of extruded material, such as gray iron to match the material of the block (or aluminum for weight) that can be bolted to the rear face of the crankcase containing a secondary seal represented in red.



(a) (b)

Figure 12: Computer aided drawing of the preliminary design of a secondary sealing device showing (a) the front face and (b) a rotated orientation for 3D view.

This preliminary design is inherently flawed however, in the sense that any oil that may leak past the primary seal and into this cavity has the opportunity to accumulate in the corners, preventing recollection into the reservoir. To address this issue, the next design was created exploring a circular geometry in an effort to provide a single point of recollection. Figure 13 demonstrates this next stage of the concept generation.

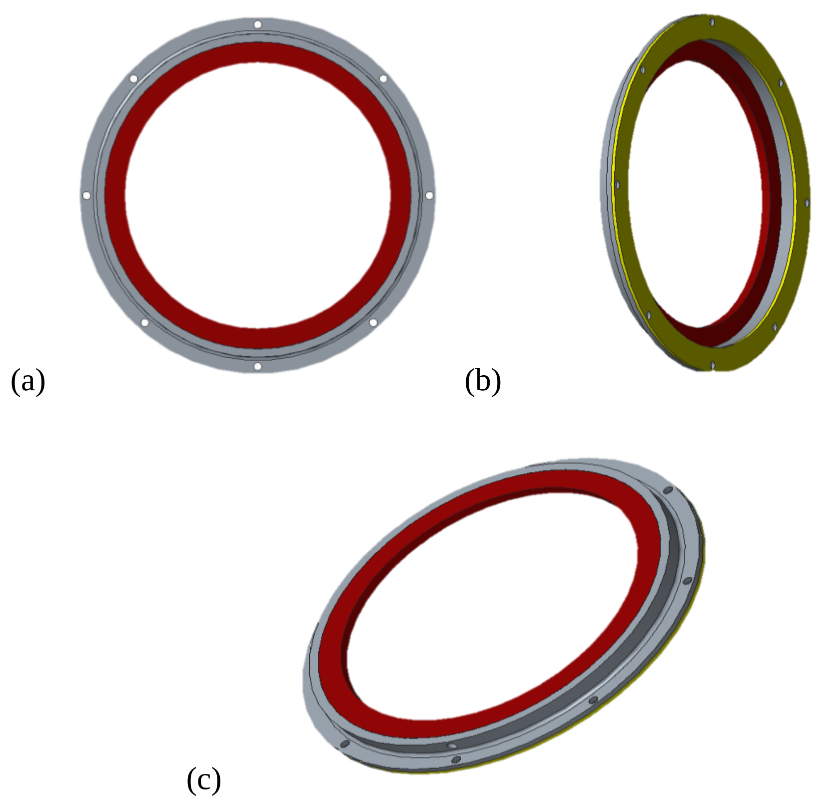


Figure 9: Computer aided drawing of the next concept of a secondary sealing device showing (a) the front face and (b) the rear face, and (c) a rotated orientation showing the drain hole at the base of the frame.

Not only does the circular geometry provide an increased opportunity for recollection, but through the addition of a drain hole at the very bottom a hose may now be attached that will provide the transportation medium between the seal and the reservoir.

5.3.2 Investigation In to Various Seals

Given the unique and extensive requirements that the 2^{nd} Stage seal must adhere to; it is imperative that the correct seal is chosen in order to optimize the robustness of the overall design. In order to do this, it is important to first know the functionality of seals, more importantly rotary shaft seals and how they operate. Once this knowledge foundation is set, the concepts can then be transferred into a viable solution for the second stage seal.

A shaft seal is used in components with rotating shafts, generally kept in lubrication with oil or grease contained in the system, while also keeping external debris such as dust and water out. The importance of these seals is crucial to the operation of the system. When a seal fails, oil leaks from the engine causing friction within the internal moving components. This increase in friction causes wear on the internal parts that can lead to premature engine failure. To accommodate this, the materials are carefully chosen with the coefficient of friction in mind, and specifically how to mitigate it with the aid of oil as a lubricant, to extend the overall life of the seal in use. Rotary or "shaft" seals are generally categorized in two types; contact seals and non-contact seals. Contact seals, as the name dictates are in contact with the rotating shaft at all times. These types of seals are your standard V or O-ring shaped seals that fit snugly around the shaft, causing a reduction in the life of these types of seals due to the extensive amount of friction and wear over time. These seals do work extremely well when a shaft rotates at low speeds or when the shaft is not rotating at all. An example of a contact seal is illustrated in Figure 10 below.

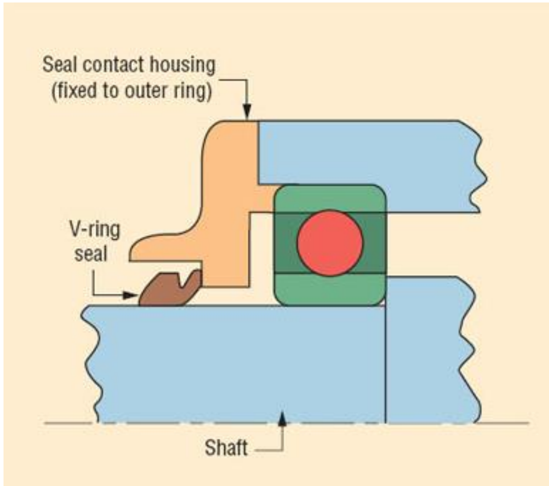


Figure 14: V-Ring contact seal illustration. [8]

Non-contact seals however, are not always in constant contact with the rotating shaft. These seals are more advanced and are typically used in long life situation. This is because the amount of friction and wear are significantly reduced as the shaft rotates. These seals can be used to seal shafts at that rotate at higher speeds. Though these are great attribute of non-contact seals, there are also some drawbacks. Being that the seal is not in constant contact with the rotating shaft, there are some instance where the oil can leak past the seal or where debris from the surrounding environment can enter in the controlled fluid cavity. The three main types of non-contact seals are Labyrinth, Hybrid Labyrinth, and Centrifugal Pressure seals. Although these seals are alike in the fact that they are non-contact, there are many differences in their actual functionality. An example of a Non-Contact Centrifugal Pressure Seal can be seen below in Figure 15.

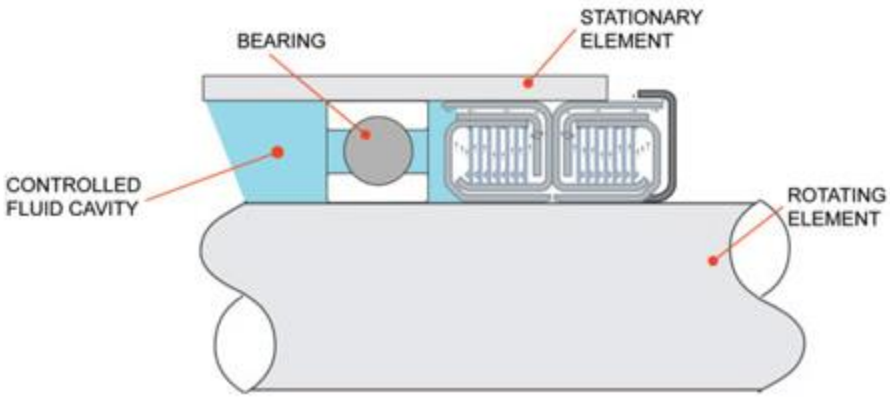


Figure 15: Non-contact centrifugal pressure Seal. [8]

The non-contact region is given by the series of channels that connect the controlled fluid cavity to the open environment. As the shaft rotates, a pressure difference within the channels of the seal produces the actual sealing effect. With this pressure difference, the controlled fluid is pushed back into the controlled fluid cavity while environment is also forced back in on its self. This makes this type of seal viable for certain situations.

In choosing a seal for the purpose of this project, it is important to realize that this seal will be dry for the majority of its life. This is due to the fact that this seal is placed behind the rear crankshaft seal, meaning the only time oil will be in this region is after the crankshaft seal has failed. If we are anticipating that the crankshaft seal will not fail until engine over hall is needed at 30,000 hours, the seal chosen must be either of a self-lubricating type or have the capability to run dry and withstand the overall wear and friction from the crankshaft. With this in mind, careful consideration and selection of the seal is of paramount importance.

5.3.3 Labyrinth Non-Contact Seal

A labyrinth seal is a type of non-contact seal that separates a controlled fluid cavity from its environment without closing off this passage way entirely. These types of seals use the fluids resistance to flow through a given passage of width and length; this is done by using a series of “mazes” or channels that the fluid can potentially flow through, but stops due to the long path of travel and the pressure drop across this path. This type of seal works extremely well when a shaft is rotating. However, when the shaft is stationary, the fluid is able to “weep” past this seal and into the environment. Also, given the unique nature of this device, an adequate seal is only generated at high rotational speeds. To further understand these types of non-contact seals, research was done at how they perform in the three stages of operation. These stages are non-rotating, constant speed, and accelerating/decelerating. In the constant speed stage, as long as the stationary and rotational elements stay axially aligned, the seal will hold and no fluid will leak into the environment. This also goes the same way when the shaft is accelerating or decelerating. As long as the shaft is rotating within the desired limits for the corresponding seal, there will be no failure. The failure with these types of seals comes during the last stage of operation: non-rotating. During this stage there is not a pressure difference between the seal and the fluid. This means that in this stage the fluid can potentially leak out or external environment can come into the controlled fluid cavity. Given this mode of failure, this type of seal may not be the best choice for our device. An example of a Labyrinth seal is shown below in Figure 16.

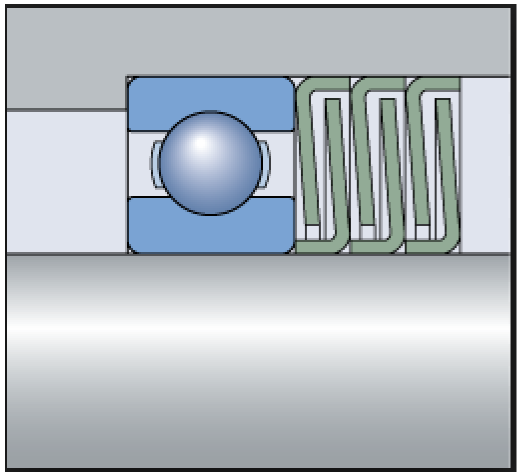


Figure 16: Labyrinth seal shown to the right of bearing. [9]

In Figure 16 above, the series of channels can be easily seen within the seal. These channels increase the fluids resistance to flow when the shaft is rotating. This is also the reason these types of seals fail with a non-rotating shaft.

5.3.3.2 Hybrid Labyrinth Non-Contact Seal

A hybrid labyrinth seal was chosen as a second option for a long life seal. These seals are much like a standard labyrinth seal in the fact that they use channels or “mazes” within the seal and use the fluids resistance to flow. However, the difference with these seals is they also incorporate aspects of a contact seal to help with the drawbacks of a standard labyrinth seal. These seals solve the problem of containments entering the controlled fluid cavity when the shaft is not rotating, but leakage may still occur, though this leakage is dependent on the amount of time the seal is stationary. These seals are closed to the external environment and therefore are a slightly more desirable design than a standard labyrinth seal. When the shaft is in its operating mode, the fluid surrounding the seal must stay within laminar flow. This restricts operation speeds with these types of seals. Also, when mounting these types of seals, alignment is crucial. If the seal is shifted in the axial direction at all, the seal will fail to perform as desired and with cause definite failure of leakage. An example of this type of seal is shown below in figure 17.

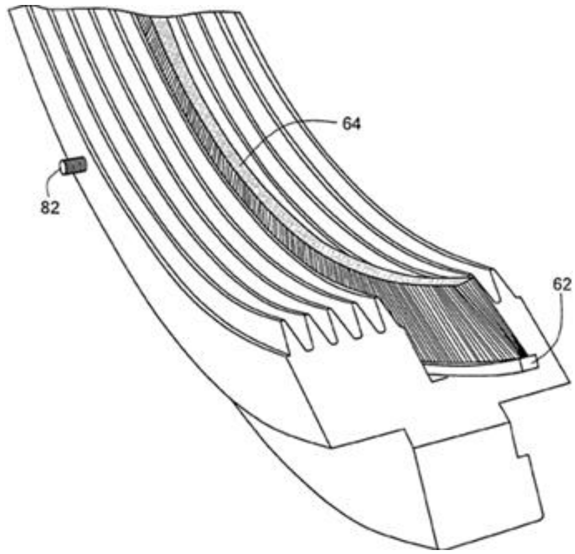


Figure 17: Illustration of a Hybrid Labyrinth Seal. [10]

A hybrid labyrinth seal is the preferred option for the super seal solution for team 1. However, they are difficult to machine and assemble as compared to a standard labyrinth seal and as such a standard labyrinth seal was designed and fabricated in order to demonstrate the effectiveness of labyrinth seals for this particular application. For further study in the future a hybrid labyrinth should be used for maximum effectiveness.

5.3.3.4 Centrifugal Pressure Seal

The next type of non-contact seal explored was a centrifugal pressure seal. This seals have an extremely long lifespan and can actually operate under more circumstances than the labyrinth type seals. This type of seal, as discussed early on creates a pressure difference within an open rotating cavity of the seal itself in order to pump the controlled fluid back into the controlled fluid cavity. Though this seal is partially open to the external environment when the shaft is not rotating. But when rotation occurs, the containments that have entered the seal are pumped out away from the controlled fluid cavity and back to the environment. This means this type of non-contact seal does not leak when the shaft is not rotating. This type of seal also corrects the alignment issue that comes with most standard non-contact seals. Centrifugal pressure seals can actual tolerate some vibration and misalignment and still perform up to the desired expectations. This in turn, makes for a viable option to our problem. An illustration of this type of seal is shown previously in this section.

5.3.3.5 Secondary Contact Seal

Another concept that was brought to attention was the implantation of a secondary crankshaft seal. This is a type contact seal that would in turn need lubrication throughout its life or suffer failure long before that of the primary crankshaft seal. Due to this situation, any progress with this concept further was deterred.

Figure 18: Standard Cummins Crankshaft Seal. [11]

As shown above, this seal is held in place by 12 screws that evenly place the seal around the crankshaft. The crankshaft slides through the middle of the seal and is in constant contact when either rotating or stationary. Consequently, this seal would wear overtime if not properly lubricated.

When looking for a solution to the seal problem, comparisons between the different options must be made. A Pugh matrix was constructed in order to help achieve this task. The different types of seals being compared were labyrinth, hybrid labyrinth, centrifugal pressure seal, and a secondary crankshaft seal. This options were compared again the essential engineering characteristics obtained from the House of Quality. The weighted scale for this Pugh matrix is from 0-2, 0 being there is not improvement and helps the system none, and 2 being there is much improvement to the design.

Engineering Characteristics	Sealing Options			
	Labyrinth	Hybrid Labyrinth	Centrifugal Pressure Seal	Secondary Crankshaft Seal
Efficiency	1	2	2	1
Durability	1	2	2	0
Size	1	1	1	1
Total	3	5	5	2

Figure 19: Pugh matrix comparison of seal types.

From the Pugh matrix, it was concluded that the hybrid labyrinth seal and a centrifugal pressure seal are the most viable option that will adhere to the extensive customer requirements. Although we feel that these are good sealing options, each type still theoretically fails some sort of customer requirement at some operation stages. For us, this means that the conceptual design phase may run longer than anticipated, until we feel we have a viable solution to this problem. Although the anticipated results have not been obtained yet, extensive knowledge on sealing applications thus far have gotten us closer to our goal of developing a long life seal with the ability to run dry.

6. Testing and Results

A series of tests were performed using the seals from Cummins Inc. as well as the labyrinth seal designed by team 1. Both of the tests were conducted over a 24-hour period using the conditions laid out previously by the sponsor. The first test was done using only the Cummins Inc. front and rear seal to see what the benchmark leak rate was before adding in the fabricated super seal. In order to demonstrate how the seals work during failure, our sponsor suggested slightly damaging the rear seal by adding scratches along the inner portion of the seal. The crank case was heated with the band heater to the specified 125°C which was measured with a laser temperature gun. Once the heater was to the 125°C team 1 allowed for a 45 minute waiting period to allow all of the oil to reach steady state temperature before pressurizing the crank case and then waited an additional 10 minutes for the pressurization and air flow to be steady state as well.

The first test was done with a pressurized crank case at about 1 psi. Without the positive pressure gradient of the secondary enclosure the oil was flowing steadily through the ‘damaged seal’. The first test resulted in a leak rate of 0.95 quarts over the whole test which is almost half of the oil that was in the crank case. During the test oil was flung radially but contained within the secondary enclosure such that it would flow down the walls of the container and then through to the drain hole. The oil effectively drained into a bucket below which was then transferred into a measuring cup to determine leak rate.

In order to prepare for the second round of testing the oil that was captured and measured was reintroduced to the crank case so that the amount of oil in the 1 run was approximately the same for the second run. The entire system was allowed to rest and cool down. The steady state parameters were achieved in the same manner as above with the addition of pressurizing the secondary enclosure to 1.5 psi. The second test was done after adding the super seal the leak was contained while running as the radial oil came into contact with the super seal. The flow of oil past the super seal was negligible compared to that contained between the two seals. The super seal, lacking a stator, was not able to effectively prevent flow of oil once the motor and pressurization was shut off, however, the oil was prevented from flowing out of the crank case while running due primarily to the positive pressure gradient.

7. Future Work

Test rig and Devices

There were several issues discovered throughout testing of the seals and test rig. The primary issues were with the motor selected for the project. The motor was effective in turning the shaft at the various speeds without the seals in place. However, the addition of the two press fit front and rear seals provided by Cummins Inc. added too much friction for the motor to achieve the full 2000 RPM specified. The testing was therefore completed at a lower testing speed. The motor was able to turn the shaft better as the testing progressed due to the wear of the seals over time but there is still room for improvement. Also, the motor would over heat every 1-1.3 hours and would have to cool off before continuing operation. In order for a more successful analysis of the effectiveness of the seals the motor should be more powerful, perhaps a 5 HP electric motor would be sufficient for the system to achieve the parameters desired.

Another issue team 1 was challenged with was that the flow rate of air through the fabricated labyrinth seal was far greater than expected for the initial test. Team 1 was initially using a standard home air compressor to maintain the pressure in the crank case which worked well. Upon attempting to pressurize the secondary enclosure as well as the crank case the air consumption was too great for the compressor to maintain and team 1 had to relocate to a fab shop in order to finish testing with a steady supply of shop air. As mentioned above, the super seal was not as effective at stopping the flow of oil with the system was stopped because of the lack of a stator part of the seal as well as a loss of the positive pressure gradient which forced the oil to stay within the crank case during operation. In order to mitigate these issues team 1 recommends adding a stator to the labyrinth seal. This would effectively control flow when the motor and compressed air are off. Also, by adding a stator the air flow consumption would be further restricted which would keep the consumption to the target of 2.25 L/s or less.

Material Selection

Throughout the course of this project Team 1 was primarily concerned with coming up with a cost effective design and as such made a few poor material choices. In order to come up with the best possible solution for future testing a few components should be re-fabricated. First and foremost, the seal was made from 24 gauge galvanized steel sheet, which proved to be unreasonably heavy and difficult to work with. The best option for an improved labyrinth seal would be to create a flange for the seal to rest on, much like the front and rear seals do from Cummins Inc. This would allow for better alignment of the seal and also allow for the seal to be much lighter. Additionally, the seal stator should be made from a high temp nylon or UHMW (Ultra-high-molecular-weight polyethylene) to withstand the temperatures but also provide a natural damping between the rotor and stator parts of the seal. The steel rotor should be changed to aluminum for weight reduction as well as ease of machining.

Future Testing

Team 1 recommends a new series of test in order to verify the different options for the sealing solution for the super seal. One such additional test would be to vary the pressure gradient from 0.5 up to 5 or even 10 psi to see how the oil responds even if the crank case is not pressurized itself. There should also be a test on varying types of seals. Different geometries of labyrinth seals such as interlocking teeth as well as stepped labyrinth seals should be constructed and tested as was the intent with the modular design of the test rig. Other seals such as hybrid labyrinth or bearing isolators should also be considered for the future of the project. A more intensive test should also be constructed to simulate the actual runtimes of these engines such as leaving them on for several days with a modular speed input such that the belts would not have to be changed periodically. These various tests would help to prove what the best possible option is for the secondary sealing device for the ISX15 motor.

8. Conclusion

The purpose of this project is to design a device that captures oil that has leaked from a rear crank-seal for a Cummins diesel motor. This leakage is considered to be a failure in the eyes of customers. In addition, Team 1 was tasked with the challenge of designing and building a test rig to determine the effectiveness of the oil capturing system. This custom test rig must be able to rotate a replicated crankshaft continuously at a steady speed of 500 RPM and 2000 RPM in order to gain the most knowledge on how the system works. The two tests lasted for 24 hours each, afterwards the oil in the capturing device was measured. The results indicated that the project was at least partially successful in that the addition of a positive pressure gradient aided in controlling the leakage of oil from the crankcase. With the addition of the aforementioned edits and changes to the test rig and seal as well as a more intensive test, Team 1 would have been able to successfully and completely demonstrate the full effectiveness of various seals under various conditions. With that in mind team 1 recommends continuing the project to see if a better sealing solution could be found.

References

1. Shaw, Terry. "Sponsor Meeting." Interview.

2. Shaw, Terry. *Project 1*. N.p.: Cummins Inc., n.d. PPT.

3. "Education: What Is Motor Oil?" Pennzoil, n.d. Web. 28 Sept. 2016.

4. "Synthetic vs. Conventional Oil." *Mobil Oil*. Mobil, n.d. Web. 28 Sept. 2016.

5. "Symptoms of a Bad or Failing Crankshaft Seal." *Your Mechanic*. N.p., n.d. Web. 28 Sept. 2016.

6. Pro, By BD Auto. "Best Oil Stop Leak & Oil Leak Sealant | BlueDevil Products." *BlueDevil Products Best Oil Stop Leak Comments*. N.p., 11 Apr. 2016. Web. 30 Sept. 2016.

7. Bweaver6. "How to Use a Potentiometer as a Voltage Divider." *Instructables.com*. N.p., 14 May 2015. Web. 21 Oct. 2016.

8. Jun 1, 2012 Michael E. Gamache President The Carlyle Johnson Machine Co. Bolton, Conn. | Motion Sy. "Engineering a Better Noncontact Seal." *Machine Design*. N.p., n.d. Web. 21 Oct. 2016.

9. "Non-contact Seals." *Non-contact Seals*. N.p., n.d. Web. 21 Oct. 2016.

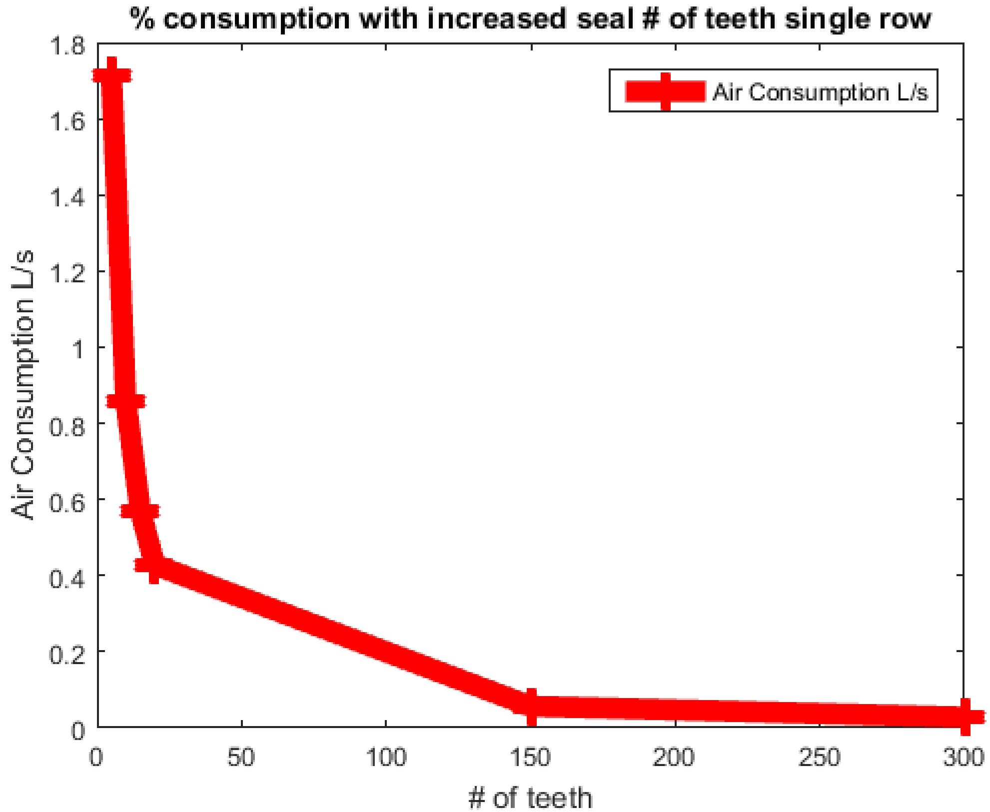
10. "Engineering Analysis for Gas Pocket Damper Seals." *Rotorlab.tamu.edu*. TAMU, n.d. Web. 21 Oct. 2016.

11. "Bearings and Seals for Cummins L10, M11, ISM, and QSM Engines." *Cummins Bearings and Seals*. N.p., n.d. Web. 21 Oct. 2016.

```
clc
clear all

%air consumption with teeth
%plot shows the air flow at each tooth
%final result indicates very small teeth

R = 190;                  % outer radius mm
r = (190-0.425);                    % inner radius mm
v20 = 15.1;               % fluid kinematic viscosity mm^2/s
rho = 1.2041e-9;          % density of fluid kg/mm^3
L = 25;               % overlap length mm
e = 0 ;                   % eccentricity, 0 if concentric
epsilon = e./(R-r);       % eccentricity ratio
p = 0.510342;              % pressure difference N/mm^2
Nt = 15;                  % Number of teeth on the seal
pdrop = p/Nt;             % pressure reduction after each tooth

for i = 1:Nt
q = ((pi*R*(R-r)^3)*(1+1.5*epsilon)*p/(6*v20*rho*L)); %flow rate mm^3/
s
p = p-pdrop;
flow_L_s(1,i) = q*(1e-6);
end
qend = flow_L_s(Nt);
% percent_cons = qend*100/225

p = 0.510342;
Nt2 = 10;
pdrop = p/Nt2;
for i = 1:Nt2
q2 = ((pi*R*(R-r)^3)*(1+1.5*epsilon)*p/(6*v20*rho*L)); %flow rate
 mm^3/s
p = p-pdrop;
flow_L_s2(1,i) = q2*(1e-6);
end

p = 0.510342;
Nt3 = 20;
pdrop = p/Nt3;
for i = 1:Nt3
q3 = ((pi*R*(R-r)^3)*(1+1.5*epsilon)*p/(6*v20*rho*L)); %flow rate
 mm^3/s
p = p-pdrop;
flow_L_s3(1,i) = q3*(1e-6);
end

p = 0.510342;
Nt4 = 5;
pdrop = p/Nt4;
for i = 1:Nt4
q4 = ((pi*R*(R-r)^3)*(1+1.5*epsilon)*p/(6*v20*rho*L)); %flow rate
 mm^3/s
```

```
p = p-pdrop;
flow_L_s4(1,i) = q4*(1e-6);
end

p = 0.510342;
Nt5 = 300;
pdrop = p/Nt5;
for i = 1:Nt5
q5 = ((pi*R*(R-r)^3)*(1+1.5*epsilon)*p/(6*v20*rho*L)); %flow rate
 mm^3/s
p = p-pdrop;
flow_L_s5(1,i) = q5*(1e-6);
end

p = 0.510342;
Nt6 = 150;
pdrop = p/Nt6;
for i = 1:Nt6
q6 = ((pi*R*(R-r)^3)*(1+1.5*epsilon)*p/(6*v20*rho*L)); %flow rate
 mm^3/s
p = p-pdrop;
flow_L_s6(1,i) = q6*(1e-6);
end


% figure(1)
% plot(1:Nt2, flow_L_s2)
% xlabel ('number of teeth')
% ylabel ('flow rate L/s')
% title ('Flow Rate vs. Teeth @ 0.85mm TIR tol & 10 teeth')
% x1=1;
% y1=1;
% str1='The consumption percentage: ';
% str2=num2str(flow_L_s2(Nt2));
% strtot=[str1 str2]
% ra=text(x1,y1,strtot,'FontSize',15)
%
%
% figure (2)
% plot(1:Nt, flow_L_s)
% xlabel ('number of teeth')
% ylabel ('flow rate L/s')
% title ('Flow Rate vs. Teeth @ 0.85mm TIR tol & 15 teeth')
% x1=1;
% y1=1;
% str1='The consumption percentage: ';
% str2=num2str(flow_L_s(Nt));
% strtot=[str1 str2]
% ra=text(x1,y1,strtot,'FontSize',15)
%
%
%
% figure (3)
% plot(1:Nt3, flow_L_s3)
```

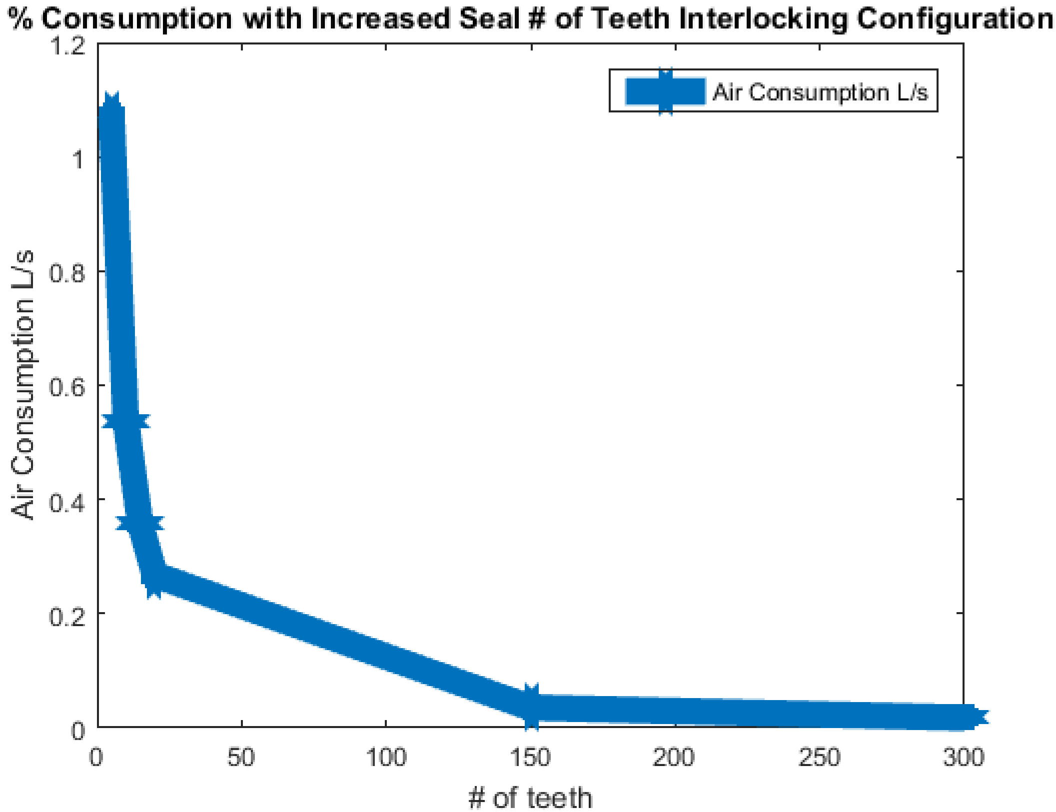
```
% xlabel ('number of teeth')
% ylabel ('flow rate L/s')
% title ('Flow Rate vs. Teeth @ 0.85mm TIR tol & 20 teeth')
% x1=1;
% y1=1;
% str1='The consumption percentage: ';
% str2=num2str(flow_L_s3(Nt3));
% strtot=[str1 str2]
% ra=text(x1,y1,strtot,'FontSize',15)
%
% figure (4)
% plot(1:Nt4, flow_L_s4)
% xlabel ('number of teeth')
% ylabel ('flow rate L/s')
% title ('Flow Rate vs. Teeth @ 0.85mm TIR tol & 9 teeth')
% x1=2;
% y1=2;
% str1='The consumption percentage: ';
% str2=num2str(flow_L_s4(Nt4));
% strtot=[str1 str2]
% ra=text(x1,y1,strtot,'FontSize',15)

sol = [flow_L_s4(Nt4),flow_L_s2(Nt2),flow_L_s(Nt),flow_L_s3(Nt3),
 flow_L_s6(Nt6),flow_L_s5(Nt5)]
xaxis = [5,10,15,20,150,300];
plot (xaxis,sol,'r-o','LineWidth',8)
xlabel '# of teeth'
ylabel 'Air Consumption L/s'
title '% consumption with increased seal # of teeth single row'
legend('Air Consumption L/s')


sol =

    1.7149    0.8574    0.5716    0.4287    0.0572    0.0286
```



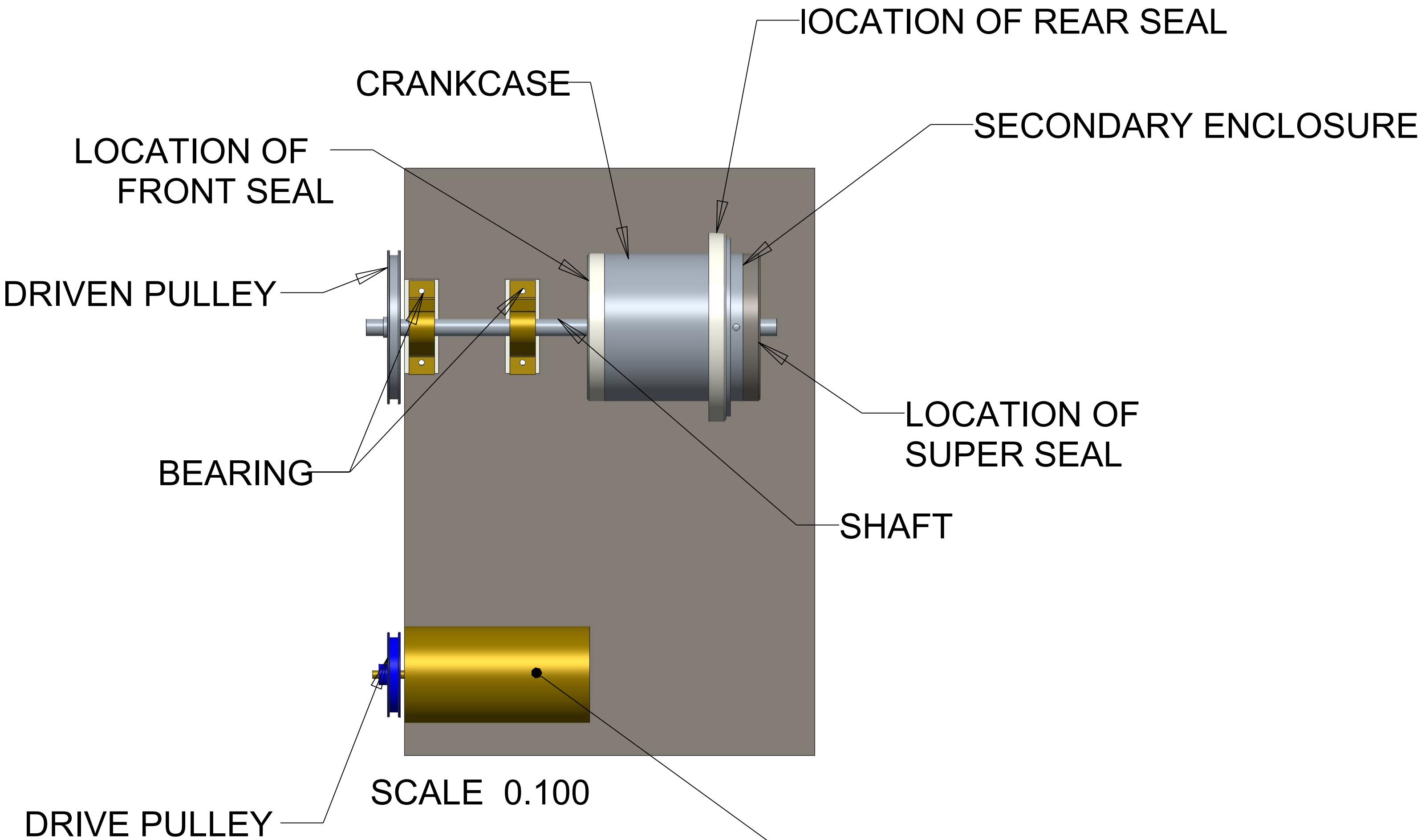
Published with MATLAB® R2015b

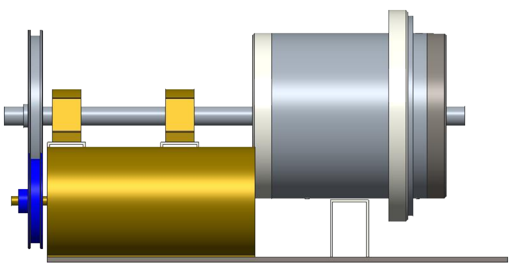
```
clc
clear all

%air consumption with teeth
%plot shows the air flow at each tooth
%final result indicates very small teeth

R = 190;                    % outer radius mm
r = (190-0.425);                      % inner radius mm
v20 = 15.1;                 % fluid kinematic viscosity mm^2/s
rho = 1.2041e-9;            % density of fluid kg/mm^3
L = 20;                 % overlap length mm
e = 0 ;                     % eccentricity, 0 if concentric
epsilon = e./(R-r);         % eccentricity ratio
p = 0.510342;                % pressure difference N/mm^2
Nt = 15;                    % Number of teeth on the seal
pdrop = p/Nt;               % pressure reduction after each tooth

for i = 1:Nt
q = ((0.5*pi*R*(R-r)^3)*(1+1.5*epsilon)*p/(6*v20*rho*L)); %flow rate
 mm^3/s
p = p-pdrop;
flow_L_s(1,i) = q*(1e-6);
end
qend = flow_L_s(Nt);
% percent_cons = qend*100/225

p = 0.510342;
Nt2 = 10;
pdrop = p/Nt2;
for i = 1:Nt2
q2 = ((0.5*pi*R*(R-r)^3)*(1+1.5*epsilon)*p/(6*v20*rho*L)); %flow rate
 mm^3/s
p = p-pdrop;
flow_L_s2(1,i) = q2*(1e-6);
end

p = 0.510342;
Nt3 = 20;
pdrop = p/Nt3;
for i = 1:Nt3
q3 = ((0.5*pi*R*(R-r)^3)*(1+1.5*epsilon)*p/(6*v20*rho*L)); %flow rate
 mm^3/s
p = p-pdrop;
flow_L_s3(1,i) = q3*(1e-6);
end

p = 0.510342;
Nt4 = 5;
pdrop = p/Nt4;
for i = 1:Nt4
q4 = ((0.5*pi*R*(R-r)^3)*(1+1.5*epsilon)*p/(6*v20*rho*L)); %flow rate
 mm^3/s
```

```
p = p-pdrop;
flow_L_s4(1,i) = q4*(1e-6);
end


p = 0.510342;
Nt5 = 150;
pdrop = p/Nt5;
for i = 1:Nt5
q5 = ((0.5*pi*R*(R-r)^3)*(1+1.5*epsilon)*p/(6*v20*rho*L)); %flow rate
 mm^3/s
p = p-pdrop;
flow_L_s5(1,i) = q5*(1e-6);
end

p = 0.510342;
Nt6 = 300;
pdrop = p/Nt6;
for i = 1:Nt6
q6 = ((0.5*pi*R*(R-r)^3)*(1+1.5*epsilon)*p/(6*v20*rho*L)); %flow rate
 mm^3/s
p = p-pdrop;
flow_L_s6(1,i) = q6*(1e-6);
end


sol = [flow_L_s4(Nt4),flow_L_s2(Nt2),flow_L_s(Nt),flow_L_s3(Nt3),
 flow_L_s5(Nt5),flow_L_s6(Nt6)]
xaxis = [5,10,15,20,150,300];
plot (xaxis,sol,'-*','lineWidth', 10)
legend('Air Consumption L/s')
xlabel '# of teeth'
ylabel 'Air Consumption L/s'
title '% Consumption with Increased Seal # of Teeth Interlocking
 Configuration'


sol =

    1.0718    0.5359    0.3573    0.2679    0.0357    0.0179
```

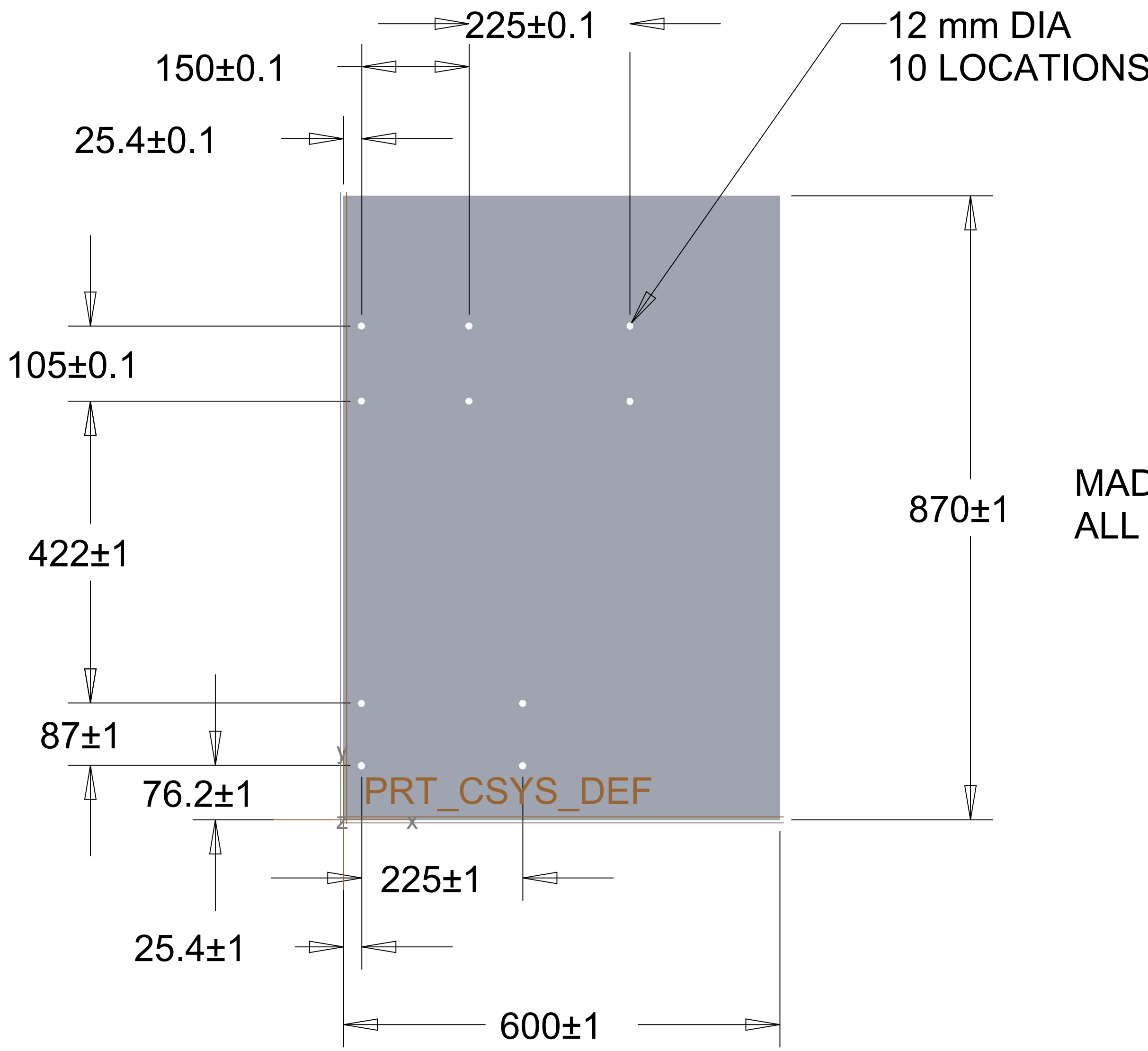


Published with MATLAB® R2015b





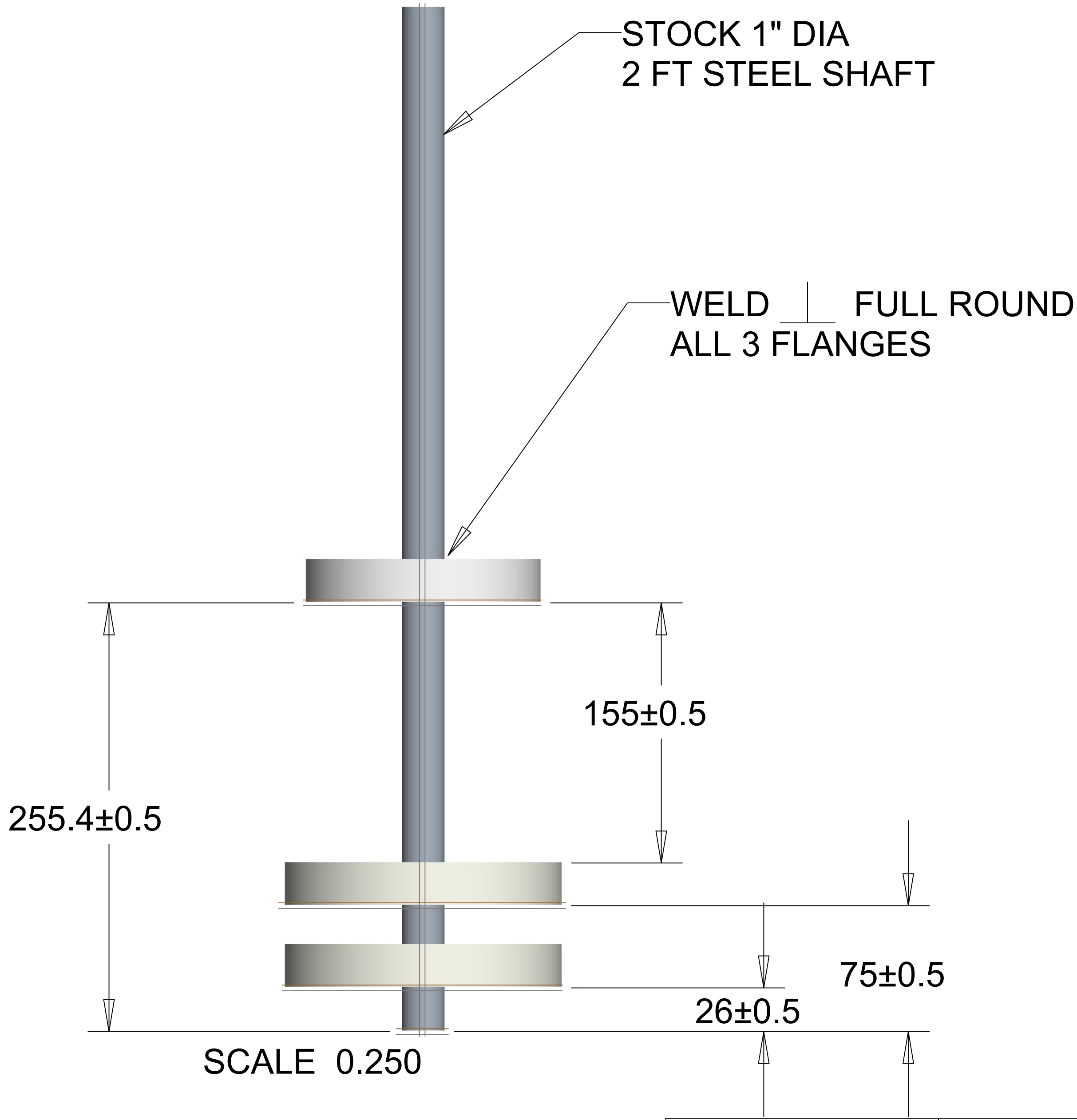
UNLESS OTHERWISE SPECIFIED	PROJECT		PART NAME	
DIMENSIONS IN mm	SENIOR DESIGN		TOTAL ASM. DRAFT	
TOLERANCES:	DRAWN BY	DATE	MATERIAL	
X.X ± 0.1	J STRICKLAND	MARCH 2017		
X.XX ± 0.01	SIZE / REV / SCALE		PART NUMBER	SHEET
X.XXX ± 0.003	A / 0 / 1.000			1 OF 1
ANGLES ± 0.5°				



MADE FROM 1/4"STOCK
ALL DIM IN MM

SCALE 0.100

PROJECT		PART NAME		
SENIOR DESIGN		BASE PLATE		
DRAWN BY	DATE	MATERIAL		
J STRICKLAND	FEB 10, 2017	STEEL		
SIZE	REV	SCALE	PART NUMBER	SHEET
A	0	0.100	1	1 OF 1



SCALE 0.250

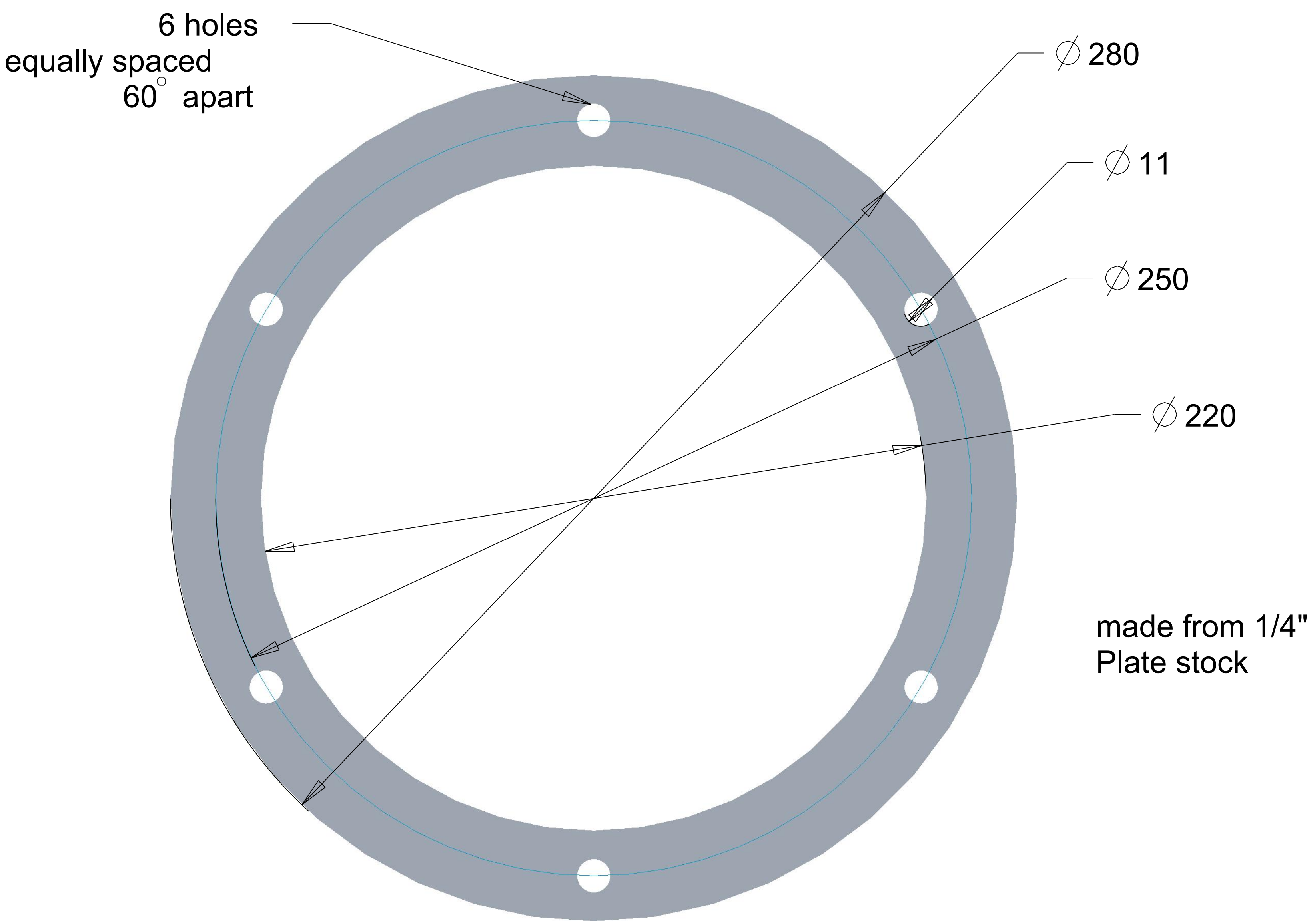
UNLESS OTHERWISE SPECIFIED

DIMENSIONS IN mm
TOLERANCES:
X.X ± 0.1
X.XX ± 0.01
X.XXX ± 0.003
ANGLES ± 0.5°

PROJECT	PART NAME
SENIOR DESIGN	DRIVES SHAFT ASM

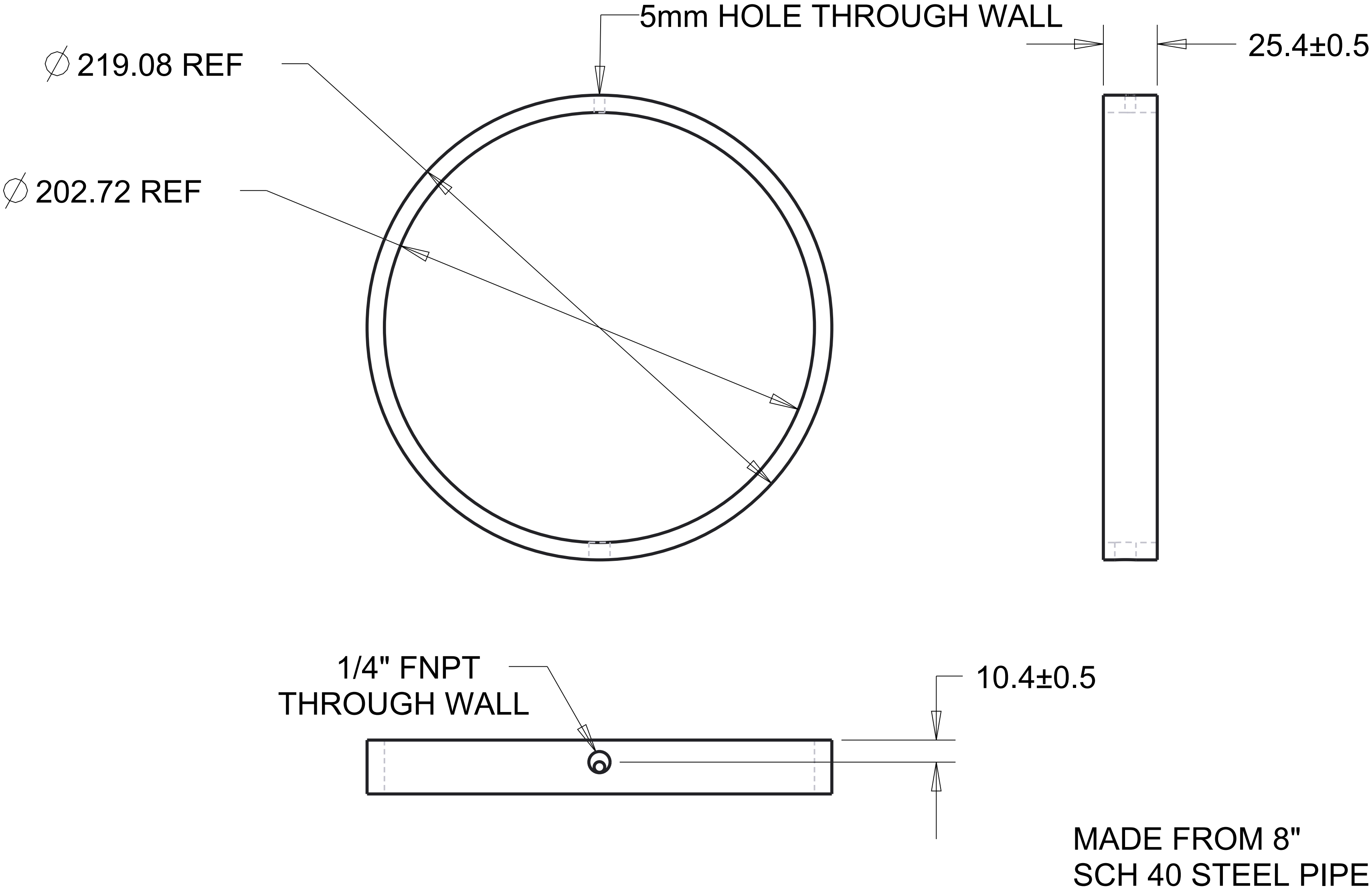
DRAWN BY	DATE	MATERIAL
J STRICKLAND	MARCH 22, 2015	STEEL

SIZE	REV	SCALE	PART NUMBER	SHEET
A	0	0.250		1 OF 1



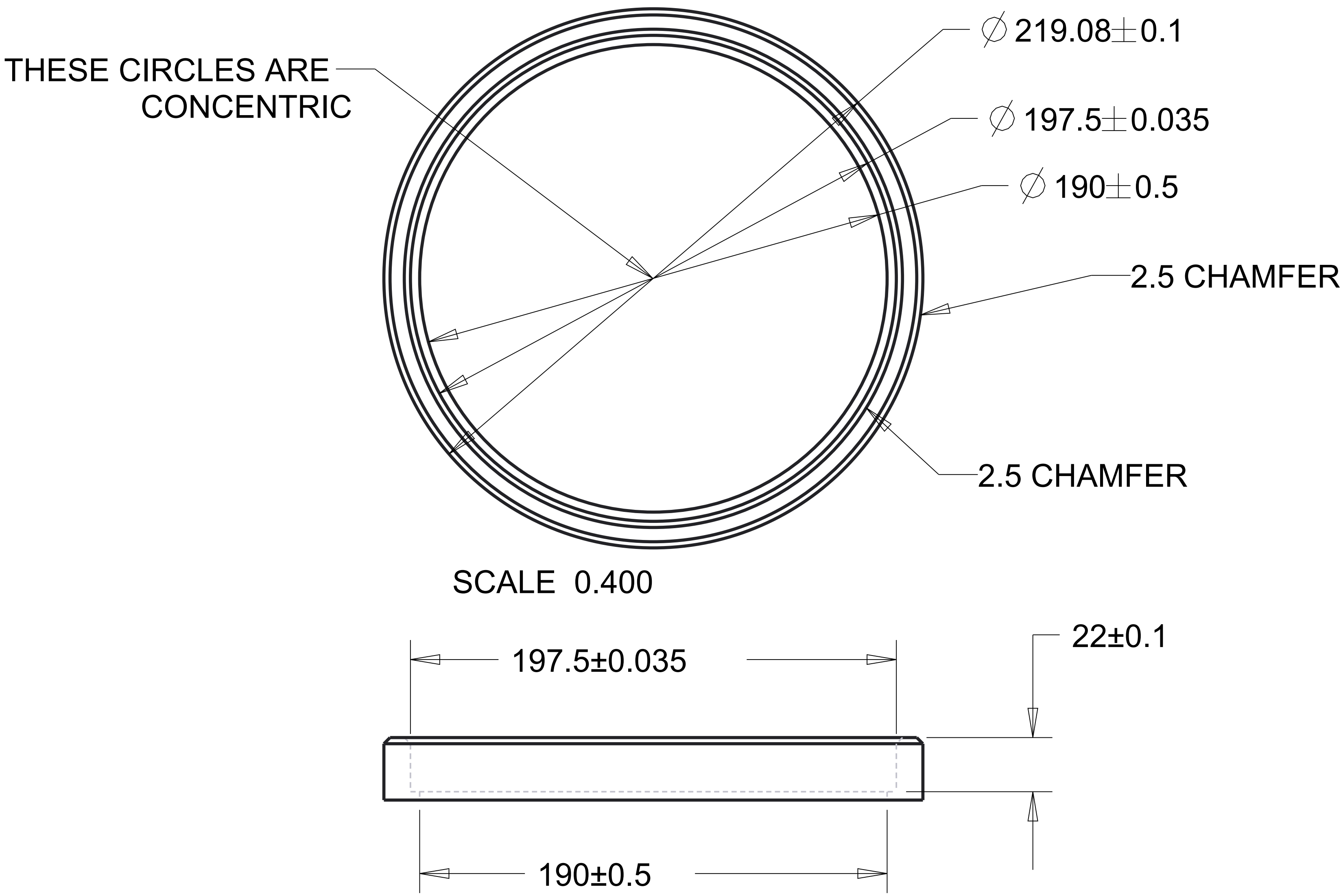
SCALE 0.500

UNLESS OTHERWISE SPECIFIED	PROJECT: Senior Design		PART NAME	
DIMENSIONS IN mm TOLERANCES: X.X ± 0.1 X.XX ± 0.01 X.XXX ± 0.003 ANGLES ± 0.5°	DRAWN BY	DATE	MATERIAL: steel	
	SIZE: A	REV: 0	SCALE	PART NUMBER	SHEET: 1 OF 1

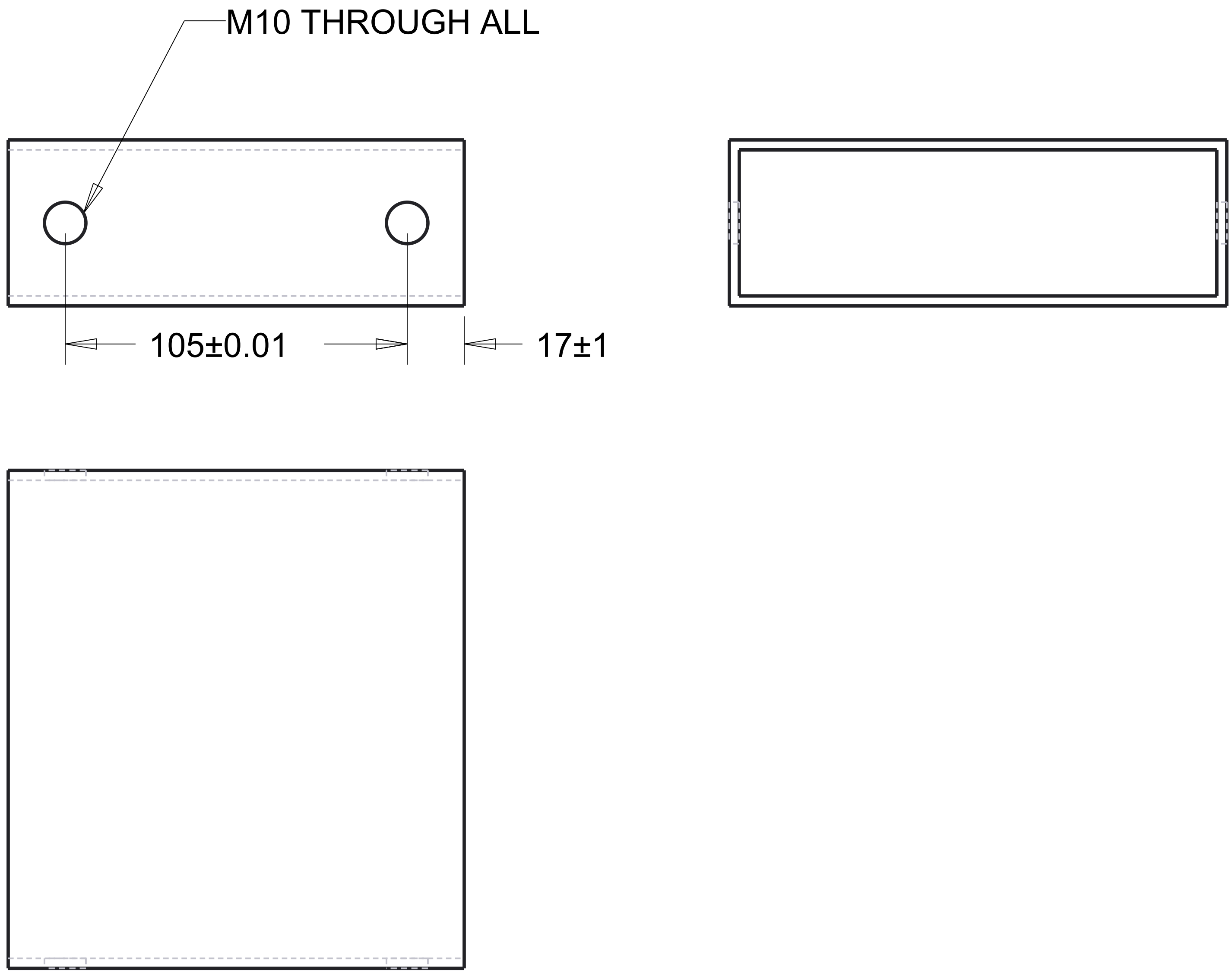


SCALE 0.350

UNLESS OTHERWISE SPECIFIED	PROJECT SENIOR DESIGN			PART NAME PRE-WELD SS ENC.	
DIMENSIONS IN mm TOLERANCES: X.X ± 0.1 X.XX ± 0.01 X.XXX ± 0.003 ANGLES ± 0.5°	DRAWN BY J STRICKLAND		DATE MARCH 2017		MATERIAL SCH 40 STEEL PIPE
	SIZE A	REV 0	SCALE 0.350	PART NUMBER	SHEET 1 OF 1

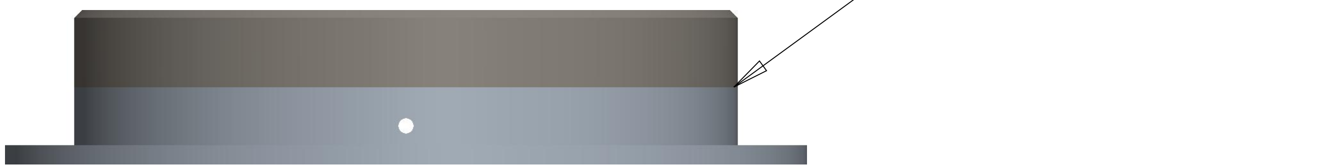


UNLESS OTHERWISE SPECIFIED	PROJECT SENIOR DESIGN			PART NAME SUPER SEAL CAP		
DIMENSIONS IN mm TOLERANCES: X.X ± 0.1 X.XX ± 0.01 X.XXX ± 0.003 ANGLES ± 0.5°	DRAWN BY J STRICKLAND		DATE MARCH 2017		MATERIAL STEEL	
	SIZE A	REV 0	SCALE 0.400	PART NUMBER	SHEET 1 OF 1	

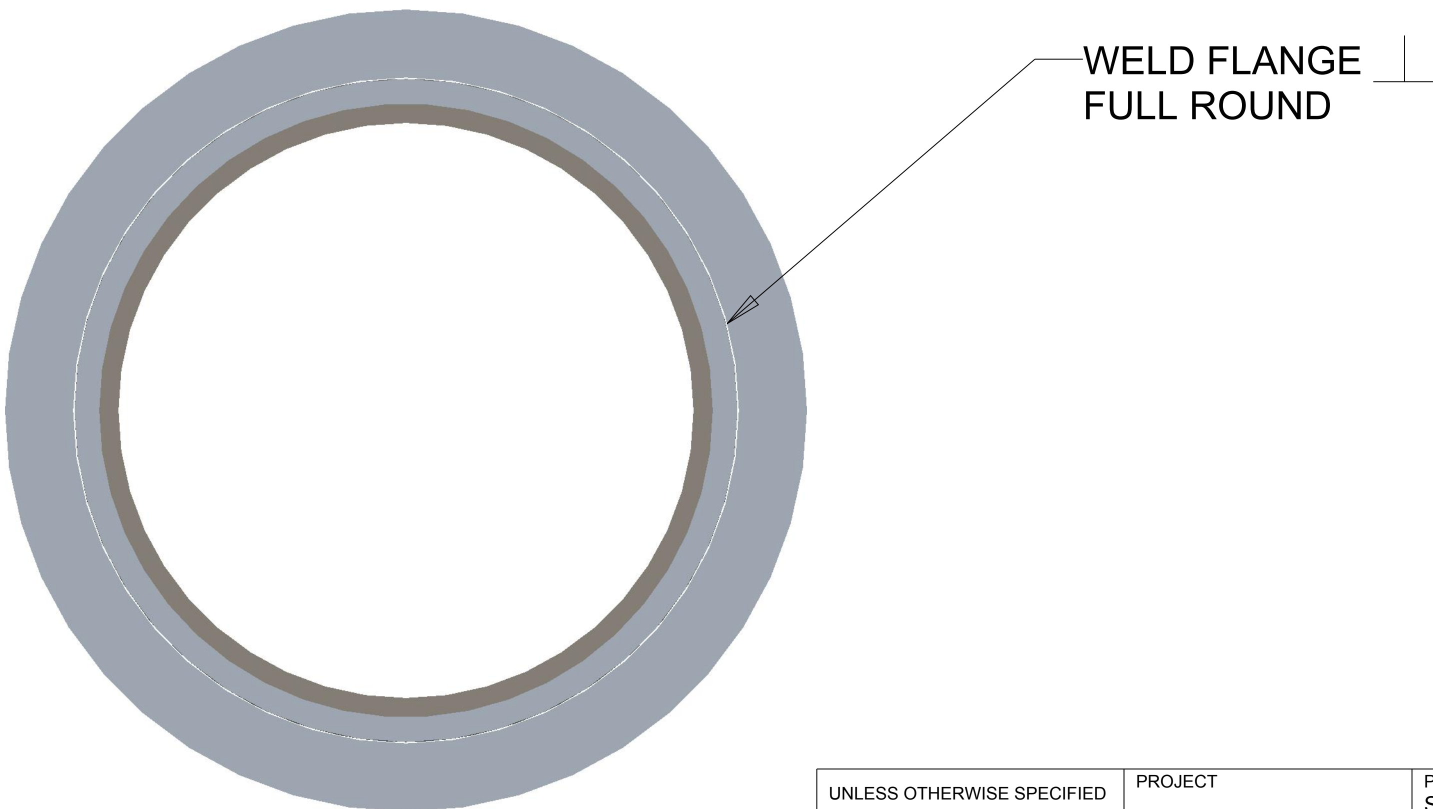


SCALE 0.500

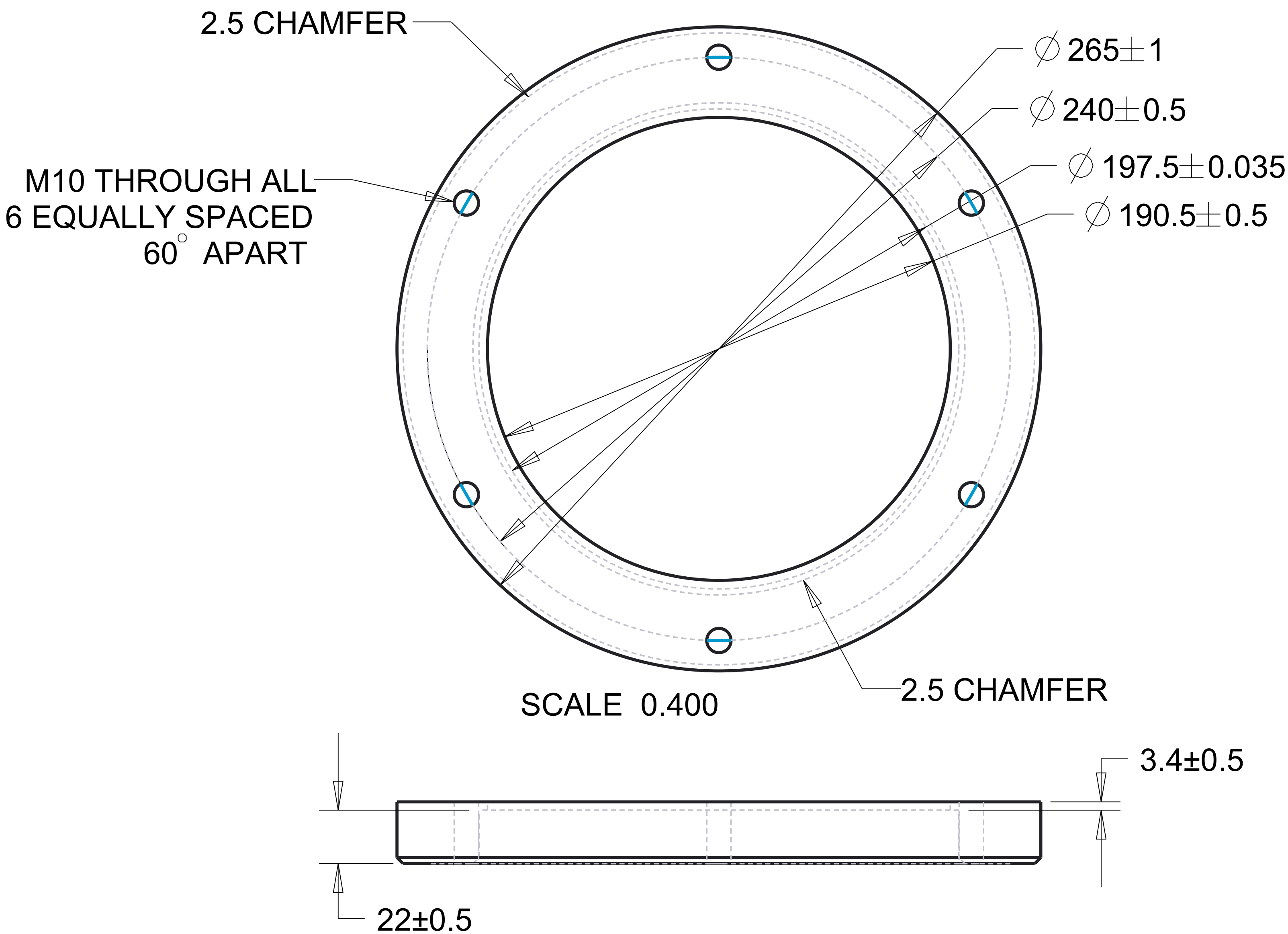
ALL DIM IN MM HOLES CONCENTRIC THROUGH BOTH SIDES STD 6"X2" REC TUBE	PROJECT SENIOR DESIGN			PART NAME SPACER BLOCKS	
	DRAWN BY J STRICKLAND		DATE MARCH 2017		MATERIAL STEEL
	SIZE A	REV	SCALE 0.500	PART NUMBER 6	SHEET 1 OF 1



SCALE 0.400



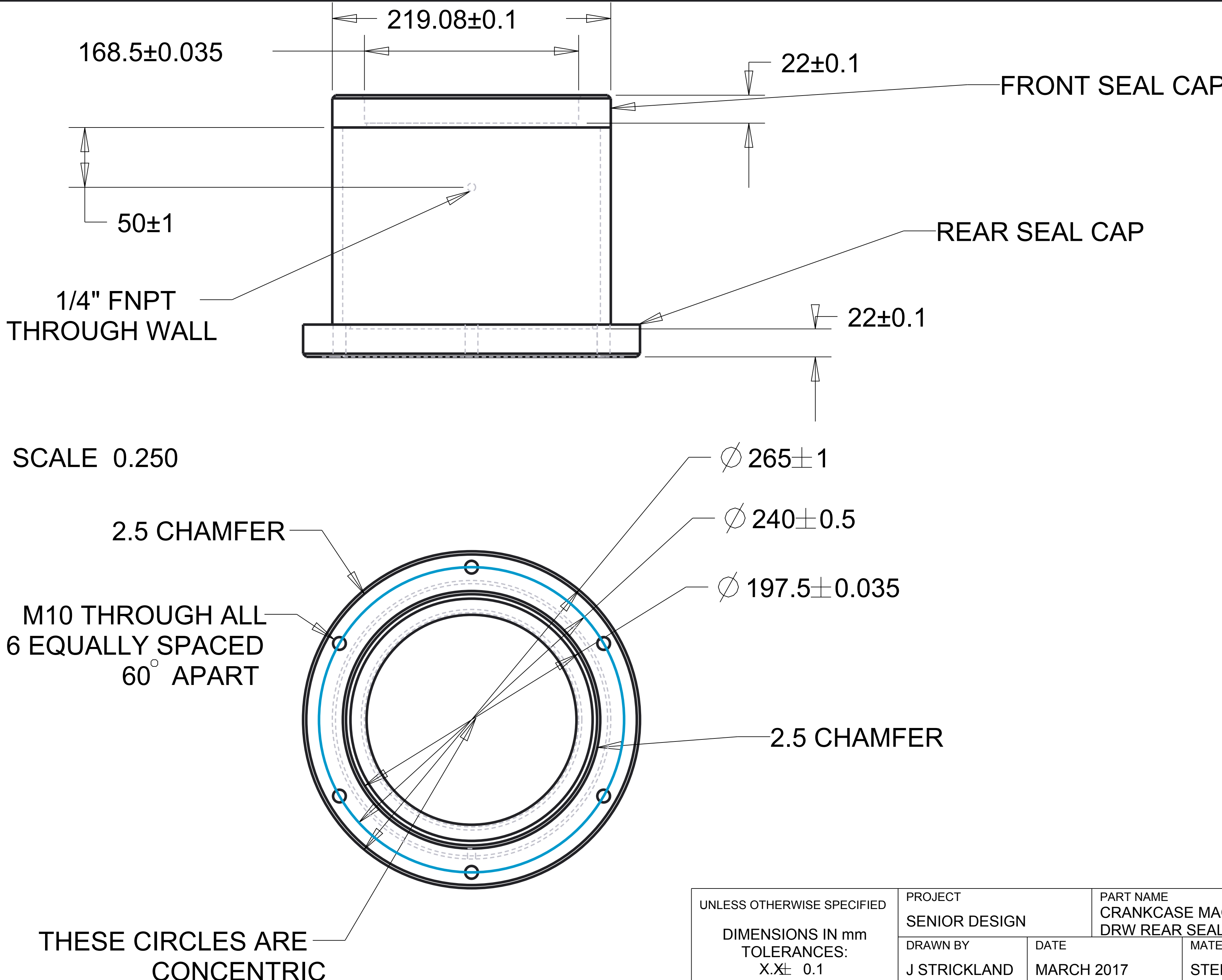
UNLESS OTHERWISE SPECIFIED	PROJECT		PART NAME	
DIMENSIONS IN mm	SENIOR DESIGN		SUPER SEAL ENCLOSURE WELD DRW	
TOLERANCES:	DRAWN BY	DATE	MATERIAL	
X.X ± 0.1	J STRICKLAND	MARCH 22, 2015	STEEL	
X.XX ± 0.01	SIZE	REV	SCALE	PART NUMBER	SHEET
X.XXX ± 0.003	A	0	0.400		1 OF 1
ANGLES ± 0.5°					



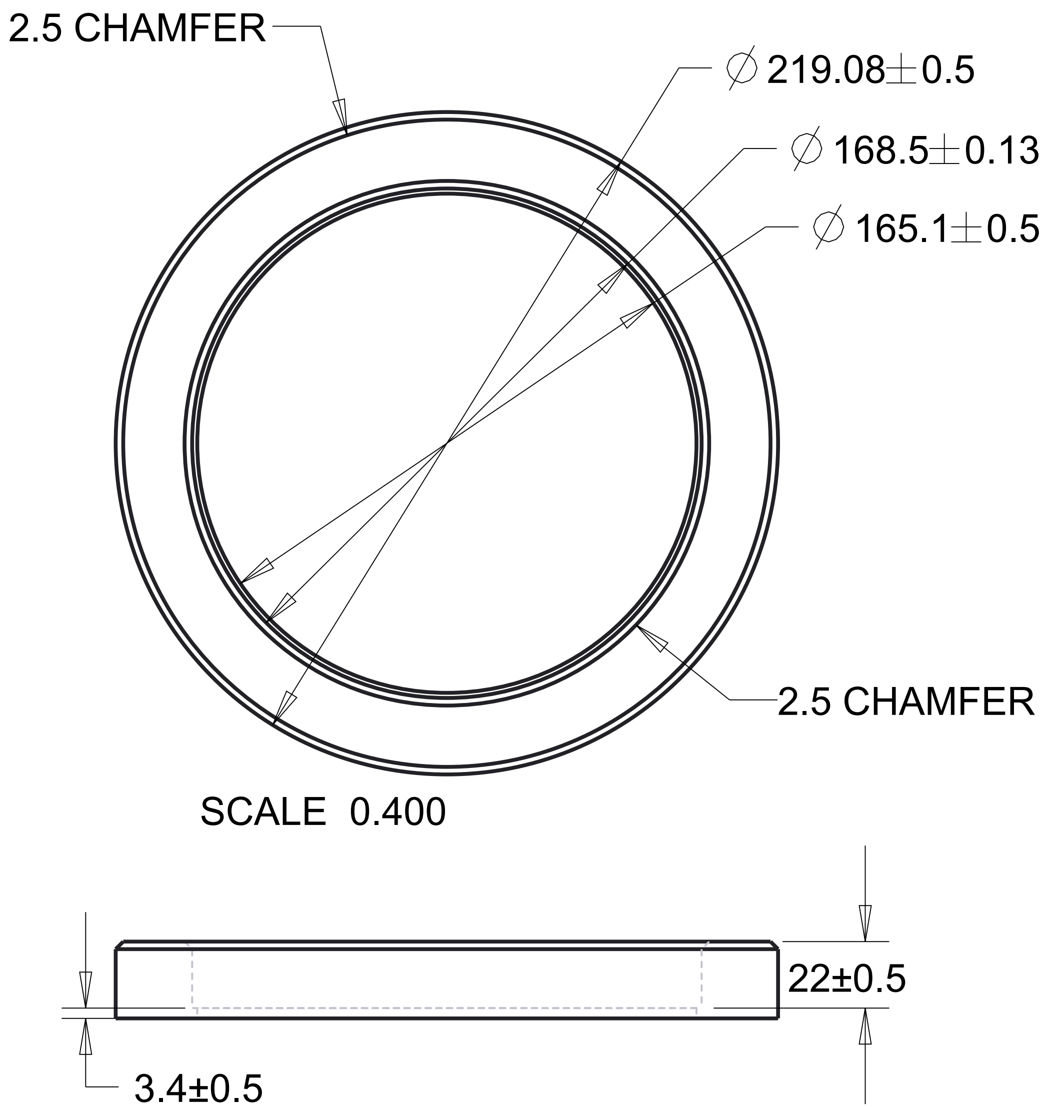
UNLESS OTHERWISE SPECIFIED
DIMENSIONS IN mm

TOLERANCES:
X.X ± 0.1
X.XX ± 0.01
X.XXX ± 0.003
ANGLES ± 0.5°

PROJECT		PART NAME		
SENIOR DESIGN		REAR SEAL CAP		
DRAWN BY	DATE	MATERIAL		
J STRICKLAND	MARCH 2017	STEEL		
SIZE	REV	SCALE	PART NUMBER	SHEET
A	0	0.400	7	1 OF 1

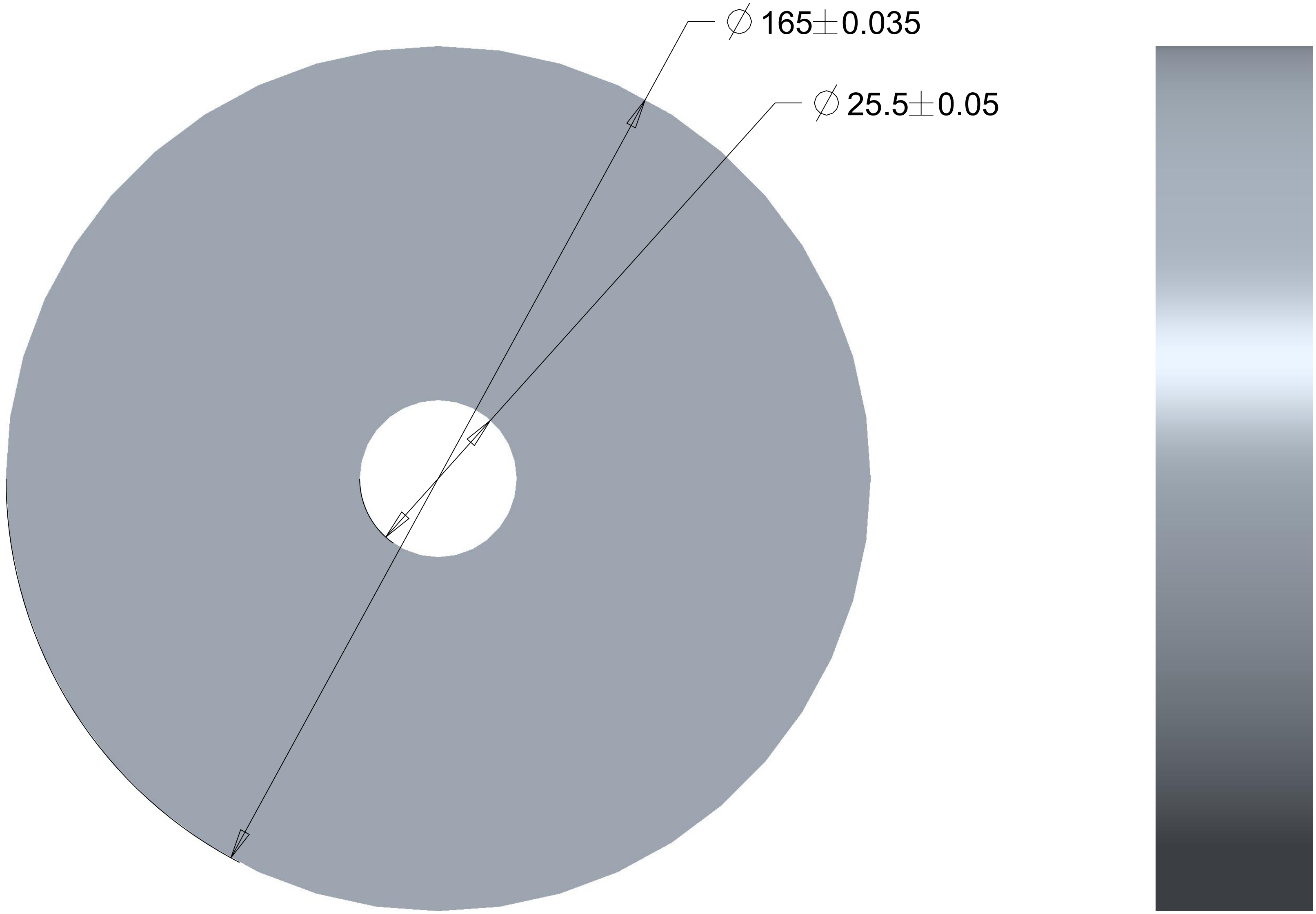


UNLESS OTHERWISE SPECIFIED DIMENSIONS IN mm TOLERANCES: X.X± 0.1 X.XX± 0.01 X.XXX± 0.003 ANGLES ± 0.5°	PROJECT SENIOR DESIGN				PART NAME CRANKCASE MACHINE DRW REAR SEAL	
	DRAWN BY J STRICKLAND		DATE MARCH 2017			MATERIAL STEEL
	SIZE A	REV 0	SCALE 0.250		PART NUMBER	SHEET 1 OF 1



UNLESS OTHERWISE SPECIFIED	PROJECT		PART NAME	
	SENIOR DESIGN		FRONT SEAL CAP	
DIMENSIONS IN mm TOLERANCES: X.X± 0.1 X.XX± 0.01 X.XXX± 0.003 ANGLES ± 0.5°	DRAWN BY	DATE		MATERIAL
	J STRICKLAND	MARCH 2017		STEEL

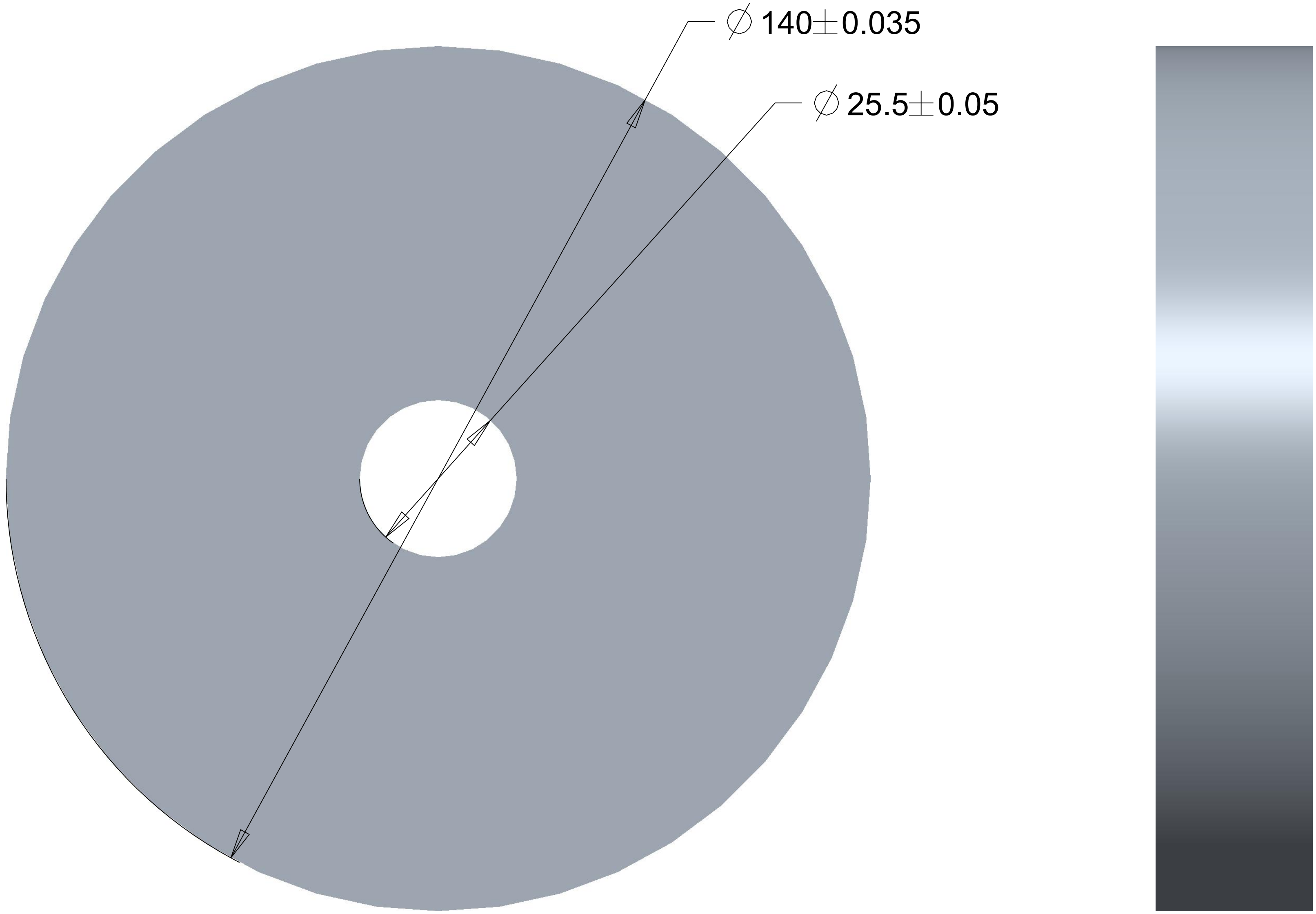
SIZE	REV	SCALE	PART NUMBER	SHEET
A	0	0.400	8	1 OF 1



SCALE 1.000

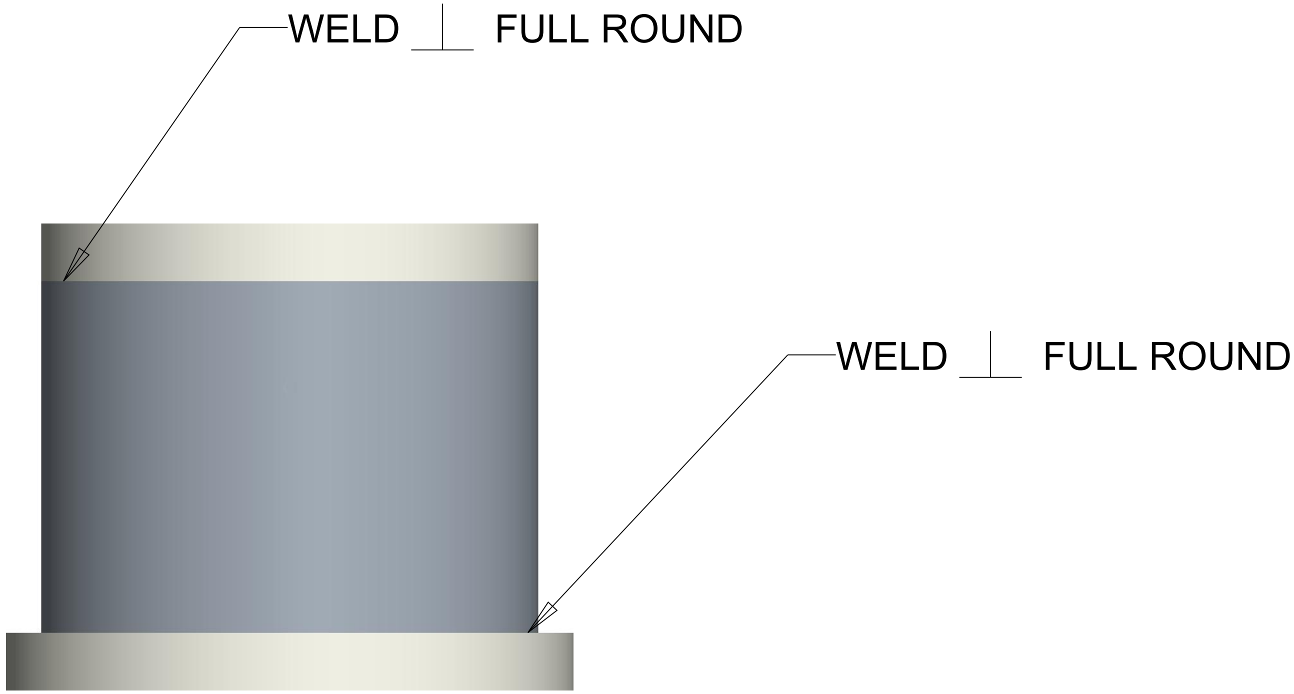
CIRCLES ARE CONCENTRIC

PROJECT	PART NAME			
SENIOR DESIGN	FLANGE REAR SEAL			
DRAWN BY	DATE	MATERIAL		
J STRICKLAND	MARCH 2017	STEEL		
SIZE	REV	SCALE	PART NUMBER	SHEET
A		1.000	3	1 OF 1



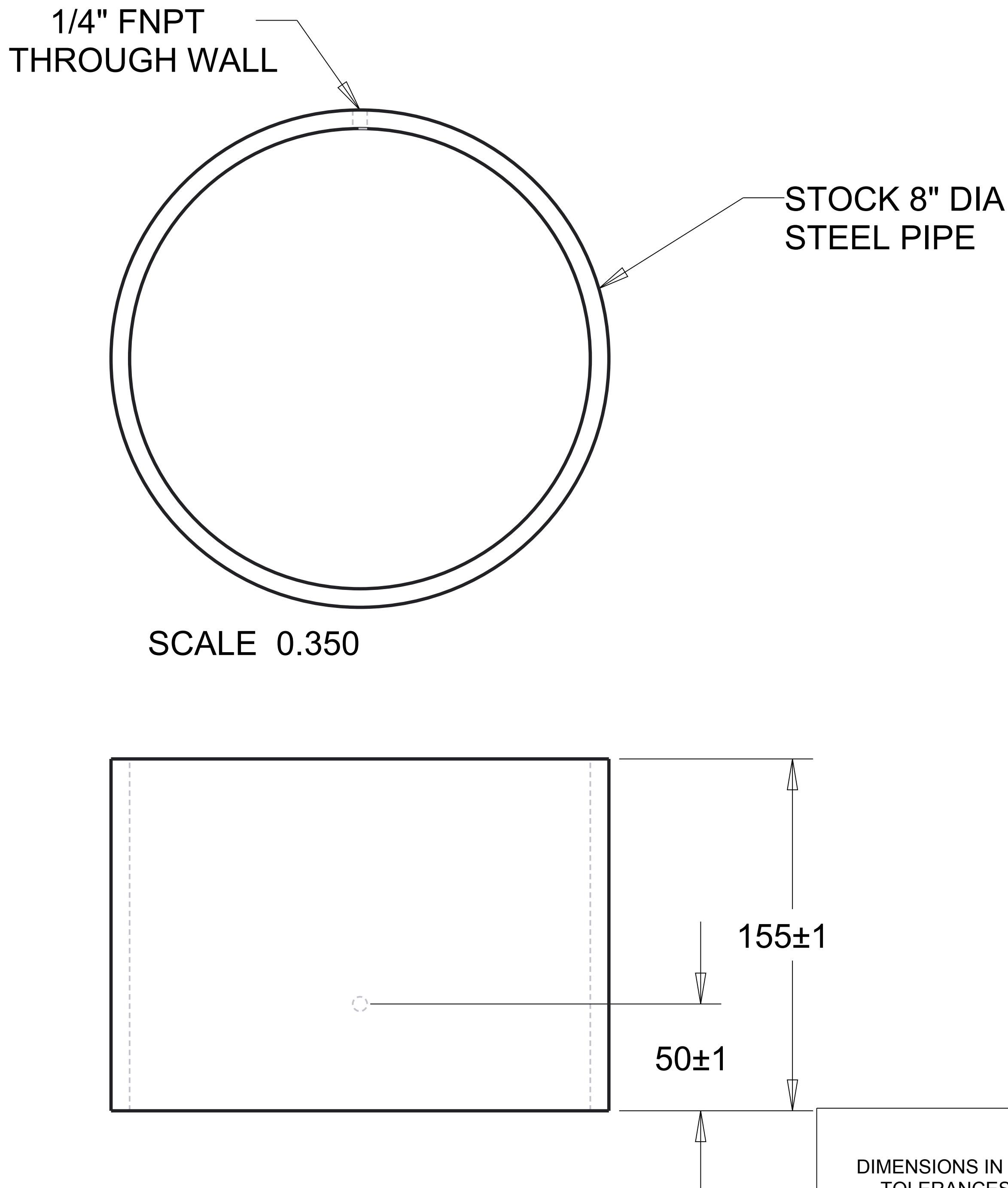
SCALE 1.000

CIRCLES ARE CONCENTRIC	PROJECT: SENIOR DESIGN		PART NAME: FLANGE FRONT SEAL		
	DRAWN BY: J STRICKLAND	DATE: MARCH 2017		MATERIAL: STEEL	
	SIZE: A	REV:	SCALE: 1.000	PART NUMBER: 3	SHEET: 1 OF 1



SCALE 0.300

UNLESS OTHERWISE SPECIFIED	PROJECT SENIOR DESIGN			PART NAME CRANKCASE WELD DRW	
DIMENSIONS IN INCHES TOLERANCES: X.X ± 0.1 X.XX ± 0.01 X.XXX ± 0.003 ANGLES ± 0.5°	DRAWN BY J STRICKLAND		DATE MARCH 22, 2015		MATERIAL ALUMINUM
	SIZE A	REV 0	SCALE 1.000	PART NUMBER SE-P-00X	SHEET 1 OF 1



DIMENSIONS IN mm
TOLERANCES:
X.X ± 0.1
X.XX ± 0.01
X.XXX ± 0.003
ANGLES ± 0.5°

PROJECT			PART NAME	
SENIOR DESIGN			CRANK CASE	
DRAWN BY	DATE		MATERIAL	
J STRICKLAND	MARCH 2017		STEEL	
SIZE	REV	SCALE	PART NUMBER	SHEET
A	0	0.35	9	1 OF 1