SHEILA-D

(Submerged Hydro-dynamically propelled Explorer Implementation: Los Angeles - Demonstrator)

100% + 5%
Excellent!!

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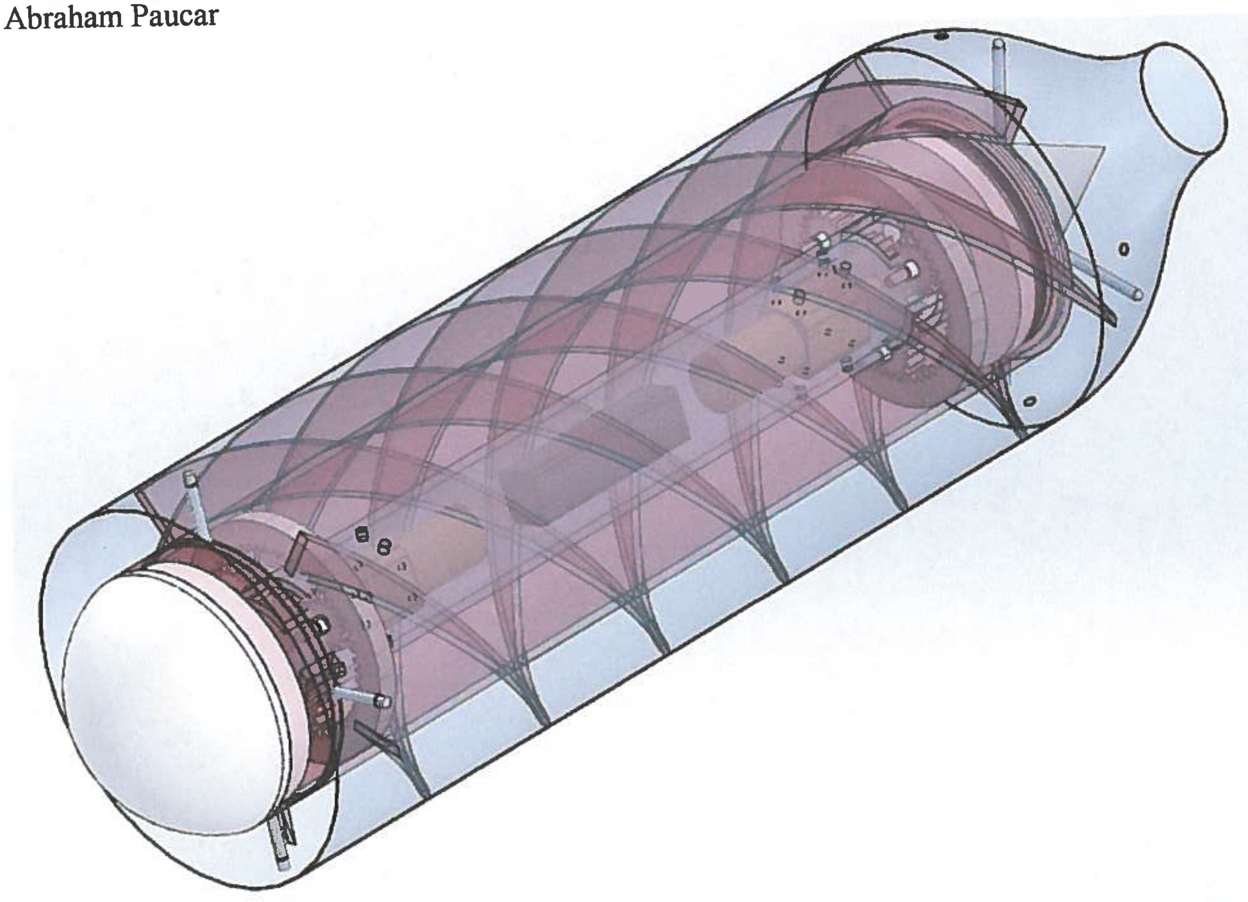

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Introduction

The objective of this project was to make a demonstrator for a new, powerful, and innovative propulsion system for an underwater vehicle. As it stands currently, there are numerous different types of underwater research and transportation vehicles all using the same propeller propulsion system. Propeller propulsion, although proven effective, has the risk of blades breaking off, cavitation issues, choppy unreliable thrust, and many other problems. This newly designed system corrects these issues and improves in many other areas. The design exchanges propeller blades for a set of winding extruded vanes pushing fluid through a nozzle-end producing a reliable, steady stream of flow that can be controlled and directed. Inside the propulsion system, SHEILA-D, are a pair of motors each connected to high ratio gear boxes; both of which are powered by a user controlled, rechargeable power source. From the gearboxes a pinion rotates 3 idling planetary gears connected to an internal gear which turns the outside cylinder that has the vanes attached. A majority of materials chosen for the design are corrosion proof polymers to reduce the weight but future designs will be of higher quality to ensure better strength and final finish. Having stated this, it should also be noted that for this demonstration phase of the design, in order to cut down production costs, many less than optimal materials were used (e.g. the sheet steel outer cylinder, and a few areas in which different metallic materials are in contact producing galvanic cells). SHEILA-D is a new, compact, safer propulsion system that provides reliable thrust to an underwater vehicle which vastly improves on what propellers and similar systems are currently capable of.

Background

The world of marine propulsion has been ruled by the use of bladed propellers for hundreds of years ever since the first deviation from paddle wheel propulsion by the US Navy in the 1840s. Today, bladed propellers are by far the most commonplace means of propelling water craft, whether above or below the surface.

While bladed propellers have the advantages of being tried and true, relatively inexpensive to manufacture, and available in a wide variety of sizes and geometries, they also exhibit numerous inefficiencies and disadvantages. Among these are the facts that they create uneven, buffeting flow, cavitate easily (leading to accelerated degradation as a result of surface pitting), produce a significant amount of noise (mainly as a result of cavitation), must be made of heavy, metallic materials (in order to mitigate the effects of cavitation damage) and are frequently exposed to foreign object damage. These shortcomings can become greatly exaggerated when bladed propellers are used for the propulsion of fully submerged vehicles; this is especially true for applications in which stealth is key, such as in military submarines, special operator deployment craft, and unmanned undersea drones such as remotely operated vehicles (ROVs), unmanned underwater vehicles (UUVs), and autonomous underwater vehicles (AUVs). For these reasons and more, an innovative approach to the development of a submersible propulsion system based on unconventional mechanics may be warranted. Enter SHEILA-D.

Submerged Hydro-dynamically propelled Explorer, Implementation: Los Angeles-Demonstrator (SHEILA-D, or SHEILA for short), is a proof of concept demonstrator that has been developed in order to explore the capabilities associated with using a ducted vane screw for submersive propulsion. SHEILA is based off of a concept that is in its fifth year of maturation; however, prior to April this concept was nothing more than an abstract dream. Currently, this concept is being brought to fruition by a highly capable, highly motivated five person team. Upon completion of this project, based off of the results realized and the performance data collected during testing, the propulsion system will be either directly utilized as a subsystem in a planned underwater vehicle senior project starting in the Fall 2014 quarter, partially adapted for the senior project, or (if unsuccessful or determined to be unfeasible) used as a jumping off point for research into other new propulsion systems.

By moving away from the traditional bladed propeller and towards the propulsion system detailed in the following pages, a wealth of advantages may be realized, including:

- Even, steady flow as produced by the entrainment of the working fluid, followed by merging of the annular water entrainment tracks prior in a convergent acceleration nozzle.
- Increased maneuverability as afforded by the newfound steady flow.
- Reduced threat of cavitation as a result of the lower rotational speeds required to produce the same amount of thrust.
- Reduced noise as a result of prevention of cavitation [STEALTH].
- Reduced fouling susceptibility due to the protective concentric vane enclosure

- Reduced weight due to the enabling of the use of non-metallic materials as a result of lower fouling exposure
- Reduced magnetic signature due to the enabling of the use of non-metallic materials as a result of lower fouling exposure [STEALTH]
- Reduced thermal signature due to the enabling of the use of non-metallic materials (and therefore insulating materials) as a result of lower fouling exposure [STEALTH]
- Reduced materials costs due to the enabling of the use of non-metallic materials as a result of lower fouling exposure
- Increased safety due to enclosure of the prime movers (motor/gear/vane assembly)
- Provision of increased speed capabilities without cavitation (also as a result of lower mass)

These benefits are but a small sampling of the plethora of advantages that can be attributed with the propulsion system demonstrated here.

While there are many areas in which a water propulsion system based off of this annular screw vane geometry can prove revolutionary, it should be noted that as with any good research and development, there are many obstacles that need to be overcome. Among the most prohibitive of these barriers are the facts that this type of system is not currently used to this effect for the propulsion of undersea vehicles and as a direct consequence of this fact, the research required for the development of this system has been, is, and will continue to be very challenging and the initial development costs have been, are, and will continue to be (at least for a while longer) quite high. As a result of these high costs, less than optimal materials have been used as most of the design has had to be based off of commercially available off the shelf components (COTS) and items which could be bought for low financial costs that, while are not at all meant for these purposes, could be adapted to serve the required/desired purposes.

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As mentioned before, the research required for the development of this propulsion system demonstrator is quite challenging by nature of the scope of this project. There exists virtually no literature concerning the use of screw-like mechanisms (other than bladed propellers, which are also adaptations of the mechanics set forth by the pioneering work laid down by Archimedes' screw) for the purpose of vehicle propulsion. As a result of this fact, the design work performed here has been largely based off of the extrapolation of known principles to the parameters of this design, as so much is unknown with respect to the specific principles used by this system. In order to define the equations necessary for the optimization of this system, an interdisciplinary approach has been adapted in which principles from fluid mechanics, classical mechanics, and machine design have been used in concert to develop approximating equations of motion which enable simplified determination of some of the performance characteristics/output variables of interest, including: the nozzle exit jet velocity, nozzle exit thrust, stall torque requirements, running torque, running time (based on suitable battery packs that meet the current/voltage and power density requirements), gearing parameters, and numerous other variables. For more on the actual design process, see the section titled "Design Process".

Great Section!

Competition

Commercially available, under water vehicles (UV's) don't come in all shapes and sizes. The idea of underwater exploration is not new and most UV's for public consumption follow the same basic principles: pvc tubing frame, modified water pumps with propeller attachments, and a large price tag. SHEILA-D is designed to take the idea of underwater exploration and point it towards the future. Gone are the high drag inducing shapes and propellers, and in their place, well thought out design and propulsion systems. There are a few companies out there that are direct competition but with a slightly different goal in mind.

OpenROV is a company which makes remotely operated vehicle (ROV) kits with an emphasis on open source development. They want to build a community of explorers of all backgrounds, from engineers to children, interested in UV's. A kit of off the shelf components is sold by OpenROV and assembled by the customer. The finished product, called OpenROV v2.6, captures video and transmits the feed to a computer. The customer is able to control the ROV from a remote control and has the option of adding functionality to the ROV at any time. The customer can freely share any findings or product enhancements to the OpenROV community. The strongest features of this design are the use of LED's to flood areas of exploration with light and the ability to control direction. Since SHEILA-D is purely a proof of concept in the area of propulsion, lighting and control features to rival OpenROV will be incorporated at a later time. The idea behind this company is unique but falls short to the same problems associated with UV's of the past. Their design, although easy to assemble, is boxy and prone to getting stuck in kelp beds. SHEILA-D is designed to be hydrodynamic and to be able to navigate through various underwater environments with ease. Since the vanes are enclosed in an exterior shell and direct driven, SHEILA-D is more able to overcome troubling environments.

Other UV's available for purchase such as the Apogee Kits Mini ROV have far less features than SHEILA-D. Most of these smaller and more affordable vehicles are prone to buoyancy issues and high drag. They are designed to be low speed explorers with most of the focus being on image capturing. The modified pumps used for creating thrust for these smaller UV's are underpowered and easily stalled when tangled in kelp beds. Other issues with common UV's are leakage and noise. These issues will have to be addressed by Team UV as well. Epoxy, waterproof duct tape, and silicone sealer will be used to protect electronics and prevent water from reaching critical areas. For a phase 1 project, this approach is adequate and common practice when building any underwater project. For later phases a more sophisticated approach can be designed. As mentioned before, noise is a common issue for underwater vehicles. Most UV designs have high operating noise from the motors and propellers. SHEILA has deleted propellers which make the vehicle run more quietly and can be a future candidate for military applications and stealth operations.

Finally, SHEILA is designed to be more affordable with more features than other underwater vehicles. The proposed total cost of this project is to be $800 for the propulsion system with later phases integrating controls and better manufacturing. So far, the project is coming in under budget with all parts adding to $700. Our direct rival, OpenROV, is retailing an unassembled kit for $850 designed for the hobby market. SHEILA is offering a better propulsion system with high speed capabilities and potential for more functionality in later phases.

Excellent.

Design Process

The design process contains a lot of detail which is broken into small sub-topics to better follow the production of SHEILA-D from the drawing board to the final water testing.

1. Design Criteria

The objective for this project was to provide a compact and innovative underwater propulsion system that differs from the traditional propeller in today's water vehicles. The underwater propulsion system, SHIELA-D, needed to be compact, about the size of an average scuba tank, yet still be able to produce enough thrust to travel at a rate of 1 knot. SHIELA-D would use a steady, constant laminar flow to produce thrust thus causing less wake when running. SHEILA-D will be a fully submersible system that will need to be able to withstand any drag and hydrostatic pressures underwater and still provide a satisfactory seal to run efficiently.

2. Initial Design/Setup

The aforementioned criteria dictated a compact design with minimal flow interferences and the capability to be autonomous. By removing the life sustaining space and power requirements, the initial design was reduced to a simple screw extending the length of the vehicle and driven by an external power source. To remain portable and easy to use, the design was restricted to approximately the size of a compressed air scuba tank. Many boats already have fittings for this shape of a package, especially in the market that this design is directed towards. With this size requirement in mind, the design was adapted to appear more cylindrical. Instead of driving the screw from externally mounted motors, a design involving three concentric cylinders was adopted. The innermost cylinder houses the motors, their power supply, and provides a space and support system for a future research or observation cell. This design focused mainly on the propulsion systems while providing space for future advancements. The middle cylinder, the vane shell, would serve as a mounting point for the threads of the screw, the vanes. Initially these vanes would only travel about 1/6th of the way around the cylinder over the entire length. The idea behind this was that the water would move faster along the length of the cylinder and therefore produce more thrust. The outermost cylinder contains the flow between itself and the vane shell. This was done to have more control over the flow and ensure that the energy produced by the rotating of the vanes went towards the propulsion of the device and not wasted as turbulence to the external body of water. The external shell also provides a mounting point for future control surfaces to stabilize and steer the device. The external shell terminates in a cone to increase the final speed of the flow accelerated by the vane shell. The increased speed of the exiting flow provides the thrust that propels the device through the water.

3. Applicable Fluids Theory

The design considered very much resembled a turbine which has a constant flow of fluid coming into the system and becoming energized by the work put in by the motors and gears. One of the main loadings that needed to be overcome was the hydrostatic forces associated with starting the propulsion system underwater. Through the two motors and gears, the initial torque that was needed to overcome these initial static loads was applied and transmitted to the vanes which then input the energy into the working fluid. Considering SHIELA-D would be moving water through its length, there were many dynamic fluid considerations to implement into the design. The choice of the extruded vanes meant that there was drag components to the fluid moving along the length as well as along the walls of the external cylinder and surface of the rotating cylinder. To avoid mixing and unwanted turbulence and drag forces the design must control the flow in such a way where it stays laminar. Once the number of vanes and angle at which the vanes are attached are chosen, the friction factor can be determined to calculate all the associated drag forces. To avoid stagnation and to control the flow at entrance, a hemisphere is placed at the front for a more streamlined input of fluid into the device and a cone at the end for a more controlled output. This dome also acted to help accelerate the flow in that it creates a high (stagnation) pressure at the tip of the dome, forcing the moving fluid to accelerate around the dome into the lower pressure flow field within the vanes. Considering the choice of materials was going to be less than optimal, the drag forces on the outside of the external shell need to be overcome in addition to the weight of the device.

4. Design Refinement

As highlighted above, a large portion of the initial planning/conceptual development stage was spent in an attempt to think through the specific design challenges to be faced in the coming weeks. This was accomplished by thinking through such things as the shortcomings of the initial concept, some of the more remarkable/noteworthy features of the initial concept, how the theories as born from fluid mechanics could be adapted and further developed for the purposes of optimizing the design of SHEILA-D, and of course the many design criteria as laid out by the team, the problem statement, and the environment to which this demonstrator would be forced to immerse itself (both figuratively and literally). Thus, after having spent many hours discussing and analyzing some of the issues, some key alterations and upgrades were made to the additional design, a small sampling of which included the radial and axial location of the components (along with the required constraints) as well as the means by which the vane shell would be allowed to rotate (and thus produce thrust) without transmitting that rotary motion to the exterior shell and motor housing. Some of the actual physical innovative solutions that arose in this stage of the design process included the use of front support rods riding in a running track and through-rods (which also resist rotation of the motor housing and exterior shell) in the rear section to keep the components located radially (which the planetary gear sets also helped with) and the use of a thrust bearing (actually a rotary ring/"Lazy Susan" style bearing) for axial location. These are solely a small sample of the design decisions made at this stage; however, these are also some of the more lasting solutions from this stage, as the design of this demonstrator has been ever evolving and as will be seen in the following pages, was constantly updated as a result of iterations pertaining to the theory, calculations, component purchasing/availability, manufacturing difficulties, and testing parameters/constraints.

5. Fluids Calculations

In order to relate the torque from the motor to the final thrust of the device, the dynamics of the fluids between the vanes and through the final nozzle needed to be considered. The torque required to rotate the vanes through the fluid was calculated through the following procedures mathematically described in Appendix B. The first step was converting the angular velocity to an instantaneous tangential velocity at the average radius of the vane. Then Bernoulli's equation was used to find the pressure on the vanes due to their rotation. This pressure was converted to a force by multiplying it by the area of the vane multiplied by the cosine of the angle that the vane made with a line along the length of the cylinder. This force was then multiplied by the average radius to provide the relationship between the angular speed and the required torque. This relation went directly into the design of the gear train and the motor selection. Once the torque required by the motor was found the thrust provided by the motor needed to be calculated. This ended up being a simple conservation of linear momentum equation. The rotation of the screw through the water applies a force both along the length of the cylinder and along the circumference of the cylinder. The circumferential component is equal to the drag previously calculated and the lengthwise component is equal to the tangent of the angle that the vane makes with a line along the length of the cylinder multiplied by the circumferential component. The lengthwise component of the force was then divided by the mass of the water between the vanes to provide acceleration. The force on the water is assumed to be constant over the length of the cylinder and the velocity of the fluid is assumed to vary linearly meaning that drag is neglected for this calculation; it will be applied later. These assumptions allow the use of the equation $V^2 = a * x$ where V is the velocity, a is the previously calculated acceleration and x is the length of the vane. This provides an exit velocity, however head loss needed to be accounted for. This was done by taking the average velocity, fluid constants and the hydraulic diameter of a vane to find the Reynolds number for the flow. The Reynolds number was then used to find the pipe friction factor and finally the head loss. The head loss was converted into a velocity loss using Bernoulli's equation and then subtracted from the previously calculated exit velocity. This corrected exit velocity, along with the basic geometry of the device allowed for the completion of linear momentum calculations which resulted in the final thrust provided by the flow.

6. Dynamics Calculations

As stated previously, the calculations performed here within consisted of an interdisciplinary approach which focused on fluid mechanics (to figure out how to produce the desired flow), classical mechanics (to figure out what kind of machine kinetic requirements needed to be imposed to create the flow), and machine design (in order to figure out how to provide the necessary machine kinetics to transfer torque from our power source into torque at the vane shell, which would then be transmitted to the working fluid in the form of kinetic shat input energy). Thus, the dynamics calculations bridge the gap between the fluid mechanics requirements and the machine design requirements.

As the inputs for the dynamics calcs, the vane geometry, geometry of the annular flow region, the vane shell length, the vane shell material properties, and the desired flow acceleration were implemented, with a desired output of the maximum (stall/starting) torque that the demonstrator would need to provide in order to provide movement of the fluid. This was accomplished by assuming that the demonstrator would be starting from stall, and therefore each vane would have to provide enough force (torque over the centroidal moment arm) at the vane-fluid interface in order to accelerate the mass of water in the annular regions between vanes at various accelerations while combating the maximum pressure exerted on the vanes by the fluid (hydrostatic, or stagnation pressure, which is the maximum pressure in the flow field for this incompressible working fluid). This was done by developing a dynamic free body diagram and deriving the equations of motion as presented in the sample calculations. A factor of safety of 1.5 was tacked onto the output total stall torque requirement to account for the rotational inertia of the vane shell cylinder itself. Once these mathematical relations had been derived, they were programmed into an excel spreadsheet just as the other calculations were, in order to facilitate more efficient design iterations down the road.

7. Motor/Gearbox Research and Selection

The previously discussed fluids calculations dictated the amount of torque per RPM required from the drive train. The challenge was to find a motor with minimal power consumption and a small package that fit these requirements. Stepper motors were initially considered because of their high torque and great control. These motors were eliminated from the selection process because they required complex control circuitry, take up a lot of power and space, and are not practical to rotate at high speeds. Off the shelf gear motors were also looked at but the majority of these motors have shafts that are off center thus making it difficult to mount them to the planetary gear set that is required to transmit the torque radially. The final option was to purchase a gear box and find a motor that mated to it. This provided more flexibility in gear ratios and package size. The planetary gear set that was designed spanned too great of a radial distance with too unusual of a diameter. This made it impossible to find off the shelf gears that would serve the purpose. The decision was made to 3D print the gears out of ABS plastic for custom sizing and less expense. To increase the strength of these gears they were designed to have large teeth and a large face width. These two decisions made it necessary for the majority of the speed reduction to be done by the gearbox attached to the motor while the plastic planetary gear set would mostly serve to transmit the load radially. The fluids calculations provided a relationship between the angular velocity of the vane shell and the torque required. The characteristics of the hobby motors that would mate to gearboxes were listed along with the gear ratios of the possible gear boxes and compared to the fluid requirements. An RS-555 motor was selected along with a mating 1:16 gearbox to satisfy the fluids requirements while remaining near the maximum efficiency of the motor.

8. Machine Design Calculations

A lot of assumptions had to be made for the machine design gear calculations. Using the overall gear ratio of 1:36 and using very basic design equations, the following parameters were calculated: number of teeth (N), pitch diameters (d), and diametral pitch (P) for the pinions, idler planetary gears, and internal gears. A lot of assumptions had to be made for the design calculation because the gears had to be designed and 3D printed from Acrylonitrile Butadiene Styrene (ABS) (imposing some serious stress concerns). It was assumed that an overload factor of 1 should be used, due to a power source and driven machine, the pinion, were both assumed uniform. Looking at the Shigley's text, it was also assumed that the gears were of the lowest quality of commercial gears, 3, to keep the design conservative. The teeth were also uncrowned making the C_{mc} factor 1 and the C_{pm} factor to also be 1 because of the lack of data for non-straddle mounted gears. In addition to this, it was decided to make the assumption of open gearing condition for the entire gear system. Since the design incorporates 2 sets of gears with two separate motors, the transmitted load was found to be divided by 2 making it only 36lb that runs through the gears. The Shigley's text only gives S_c and S_f factors for steels and other alloys and no other data to calculate the allowable stresses. Therefore, the S_c and S_f factors were assumed as our flexural yield that was taken from the ABS material data sheet.

9. Battery Selection

The power source came into careful consideration after the motor and gearbox selection was completed. Initially it was decided to use an external power source for the motors but this was dangerous as the system would be submerged underwater and thus it was decided to look for a rechargeable power source connected to a switch and potentiometer to control SHIELA-D from the surface and not worry about it running out into the water until the battery died. The battery needed to be able to power the motors for long enough to be able to collect accurate data when testing underwater. It was assumed that an hour of testing would also be unreasonable for run time (due to scarcity of high enough capacity batteries) and that 15 minutes of testing would be too short a run time before the battery packs would have to be recharged, due to the inconvenience posed by recharging. A number above 30 minutes was selected as a run time goal and the battery that was selected was the Lynxmotion 11.1V 28000mAh 3S 30C LiPo Battery which met all of the requirements. Once such met requirement was that the current draw it could handle (seeing as it is a 30C rated battery which means it can handle to release 30 times its rated 2.8A continuously for one hour). This also yielded about 50 minutes of testing time to collect data from SHIELA-D which met the imposed test run time requirement.

10. Iterations (Optimal Setup)

Once the important electrical and mechanical items were decided (gearbox, motor, and battery) the first stage of calculations needed to be reiterated to reflect our updated design. The gear box provided a 16:1 ratio which provided great support in designing the planetary gearset which was designed to be a 2:1 ratio thus giving the gear system as a whole a 32:1 ratio. Calculations such as drag, speed, stresses, and overall thrust of the system needed to be reevaluated due to this change. A battery was chosen that would be powerful enough to power two motors in the core of SHIELA-D. This battery was sufficient to produce 30A in order to start both motors so 12 AWG wire was chosen since the nature of it's size would be able to withstand the current produced from the battery. However, since the battery would produce such high power at startup the system needed a slow starting mechanism so the gears would not undergo great initial force therefore a potentiometer was included in the wiring setup. There would be two switches wired in parallel, one that included a potentiometer that would be used for startup purposes only and the other just with a switch alone that would be activated once SHEILA-D was at full speed. Switches that were rated anywhere close to 30A were chosen as to make sure nothing would burn leaving the system stranded in the water.

11. Purchasing/Acquisition/Subassembly Refinement

In trying to hit the early presentation date, purchases were made at brick and mortar locations whenever possible. This allowed for in person inspection of quality and to ensure timely acquisition. In-person purchases were made at Home Depot and Orchard Supply Hardware for PVC components, exterior shell ducting, small hardware, and sealants. Specialty items such as electrical wire and small connectors were purchased at Pegasus Hobbies and Fry's Home electronics for their more refined selections. These locations were relatively near or along the way, to SHEILA's manufacturing site. As with any unique project, not all components can be purchased off the shelf or in person. More critical components such as the LiPO battery and electrical components had to be ordered online. Specialty online shops such as RobotShop.com and McMaster-Carr allowed access to parts that best fit the project requirements. These components needed to be ordered quickly to ensure timely delivery. Most deliveries were expedited by upgrading to priority or next day shipping. During manufacturing, quick runs had to be made to local hardware stores to pick up small parts. These small parts such as clamps and washers were not part of the initial design but essentially made SHEILA-D a possibility.

This phase of the project is heavily limited by the availability and cost of possible materials. Purchasing more effective and premium materials would have cost the team a tremendous amount of capital. Decisions to use less optimal components warranted timely construction and a reasonable budget. A spreadsheet of purchases and purchasers has been compiled to make sure that a detailed budget can be made. This is also a great way to make sure correct distribution of cost among all members.

12. Manufacturing/Troubleshooting/Further Subassembly Refinement

The first stage of manufacturing was to construct the vanes. The vanes needed to form along the rotating cylinder at a 45 degree angle leading to about a one and a half cycle per vein from top to bottom. The material for the vanes was acrylic and the rotating cylinder would have 5 vanes total. Each vane started from a rectangular shape that was 2 inches in width and roughly two feet in length. Several slits were cut about one and three-quarters of an inch in the acrylic so it could be formed around the rotating cylinder without breaking the material. A heat gun was used to soften the material at each slit so it could be formed around the cylinder then cooled with ice or water to quickly form the angle desired, this process was repeated for each of the vanes. Once the final shape was was obtained for each of the vanes they were epoxied to the rotating cylinder and waterproof tape was applied to both sides of the vanes to cover the spaces produced by the slits after forming the material around the cylinder. The tape was also to provide a solid constant surface area to move the fluid through the system.

The planetary gearset was made of ABS plastic and 3D printed. They were printed as one long extrusion and cut to length. It was found that the printed gears were not solid as expected, but consisted of a honeycomb shape throughout the internals of the gear. Since these gears are undergoing a considerable amount of torque along with them being plastic they were then filled with epoxy in order to provide a more rigid gear that could take the sufficient load. The pinion gears were press fit onto the gearbox and sealed with epoxy in order to ensure the gear would not strip from the gearbox shaft. Quarter-inch diameter rods were cut in lengths of roughly 3 inches that would hold the planetary gears connecting the pinion to the internal gear; these shafts sat inside nylon bushings that are mounted to the motor housing to provide stability during motion.

The motor housing (innermost cylinder) contained the power source, motor, gearbox, pinion gear, partial of the planetary gears, support rods and wiring. The motor housing was made of PVC and cut to a length of two feet. Several cuts were made to this such as holes for mounting the gearbox, slits to provide room for the planetary gears that connects the pinion to the internal gear, and holes for the support rods in the rear of the system. The rotating cylinder (2nd innermost cylinder) has a separate clear plastic hemisphere on the front of it where a camera would sit as well as provide a seal to protect the battery and the motors from any water. The inner diameter of the rotating cylinder was bored out so the globe to fit inside along with a gasket to provide sufficient sealing when underwater. A track was cut on the outside of the cylinder, roughly an inch from the font. This track would sit on nylon tubes to provide support for the front of the system from any flow induced vibrations and keep the rotating of the cylinder as concentric as possible.

The length of the motor housing is longer than the rotating cylinder so in order to provide a seal in the rear a three in diameter hole was cut in an acrylic plate that would sit on the back of the rotating cylinder and the motor housing would stick out of the plate. A rubber shaft seal was placed on the motor housing that extended past the acrylic plate to create a seal as well as allowing the shaft to rotate. On the inside part of the rotating cylinder a plastic turntable, aka "Lazy Susan", was used to act as a thrust bearing. It was placed inside toward the rear end of the rotating cylinder that mounted on the motor housing and would provide support for any axial loads as well as keep a low friction force if it may come in contact with the plate that provided the seal in the rear. A cone made of thin aluminum was placed on the back of the motor housing to provide a smoother flow of the fluid and reduce any losses in the system before the water reaches the nozzle.

The external shell (outermost cylinder) consisted of thin sheet metal, like those used in air conditioner ducting, and a separate cone attached to the back to act as a nozzle. The cone was also material used in HVAC application and was modified to have a small exit area in order to provide a higher exiting velocity therefore increasing thrust. Holes for the support rods were drilled in the back to provide support between the two non-rotating cylinders (innermost and outermost). Nylon tubes were mounted inside on the front of the external shell that would sit on the track cut out on the rotating cylinder.

SHIELA-D consisted of two motors powered by one battery that lead to two switches connected in parallel. On one side of the parallel wiring was a switch along with a potentiometer and the other side of the parallel was a switch alone. In startup the side that contained the potentiometer and switch was used first in order to provide a slow startup then at full speed the other switch would be activated and thus be the official "on/off" switch.

13. Test Design/Location Scouting

In order to successfully complete this project, a few tests were developed to ensure functionality of the design. SHEILA-D would undergo go testing to determine: Run capability, air speed, water exit jet speed, water thrust, submerged top speed, stall acceleration and time, flow field, and battery discharge time. Four tests would ensure capturing of this data with the first one being an Air Test. This test would offer run capability and air speed out the back. To run this test an anemometer is needed to read air speed and testing supports would have to be constructed to constrain SHEILA-D.

The second planned test is an Open Water Test. This test will also determine run capability but provide submerged top speed, stall acceleration and time. In order to run this test, an apparatus called Rectilinear Motion Constraints (RMC) and a tow line will be deployed to control SHEILA-D. A testing location for test two was found at Frank G. Bonelli Park. Puddingstone Lake, found within Bonelli Park, is often used for recreation and project testing. After talking to another project team performing tests at Sail Boat Cove, a decision to test at the same location was finalized to meet all test requirements. Sail Boat Cove has a depth of 10ft and a length of 50ft in which to test SHEILA-D. Although the water clarity is poor and there can be some foot traffic, this location is perfect for easy and quick setup.

Test three is the Bath Tub Test which will be used to obtain water exit jet speed, water thrust, and flow field. This test will need a spring scale, dye, a waterproof anemometer, and use of the RMC. An inspection of SHEILA-D will have to be made prior to this test to ensure no damage was done during test two. The last test will be the Full Battery Discharge test. This will be performed right after test three as the same setup can be used. This test will provide battery run time and fade out.

14. Testing

Testing was first scheduled to begin on Monday, May 12th 2014 but a potentiometer failure delayed the testing schedule by a few days. Testing was conducted a few days later on Friday, May 16th 2014 at Puddingstone Lake. Team UV arrived at the test location at 0600 and promptly performed final sealing of the motor housing and camera location.

The RMC was positioned and after placing SHEILA-D in the water, the motors were powered on. The demonstrator began to move but power was quickly shut off after water was noticed to leak into the critical motor housing. Opening the motor housing through the observation dome allowed for draining of the water and quick drying. A second test was performed swiftly to obtain video footage for the presentation and to save the motors from frying. This test also failed as the vane shell stopped rotating due to high friction in the system.

An assessment day was scheduled for the following Saturday (5/17/2014) to improve the meshing of the gearset and to improve motor housing sealing. Filing of the gear faces to remove excess material and applying more epoxy to failed sealing areas provided an adequate solution to problems faced during the first test. Team UV also noticed that the vane shell was under a lot of friction. This was fixed by adjusting the location of the rear seal and thrust bearing. After air testing the improvements made to SHEILA-D, a planned water test was made for Monday, May 19th 2014.

On May 19th, 2014 SHEILA-D successfully completed an underwater test. Not only did the demonstrator move linearly in the water, it showed a striking free jet out the rear nozzle. Multiple runs were made to show the free jet and each run was video documented.

Additionally, testing of sub-assemblies was performed at various stages of the manufacturing timeline. Waterproof duct tape, which will be used to seal various components, underwent a submersed test to ensure sealing abilities. The conclusion was no decline in adhesion but water did penetrate the dry interior of the tube. Another critical test was the confirmation of motor power and rotation. The final electrical circuit to power the motors was wired and allowed to run for a few minutes to confirm desired operation. Meshing of the gears once located on the motor housing was also video documented.

Future plans to test SHEILA-D have been made to collect performance data. Need more testing

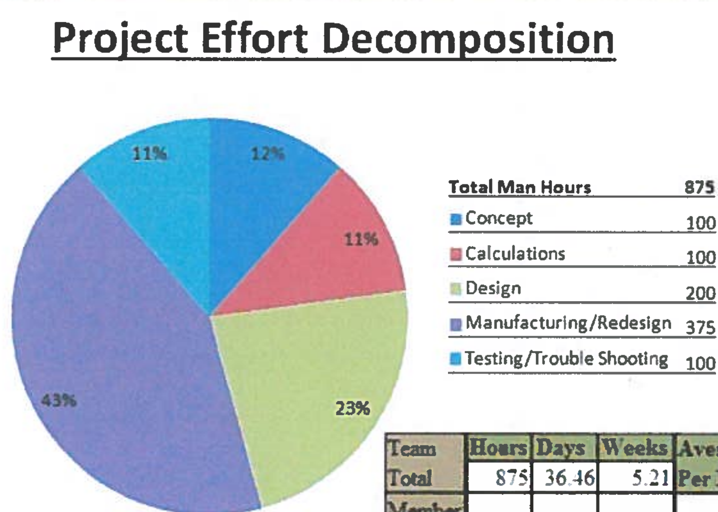
Great Section!

Results

Successful testing solidified the vision of SHEILA-D as an adaptable propulsion system demonstrator. Some very noteworthy results were realized through the second round of testing, including: successful production of a *initially laminar* freejet (and the thrust that is associated with this jet), minimal wake regions stemming off the front edges of the demonstrator, and steady flow out the rear of the convergent nozzle; the first two of these effects may be spotted by the trained eye in the flow field displayed in Figure 10. These positive results verify this type of propulsion as a viable option for the future, thus validating the intent of this project.

Having noted some of the good results associated with the successful testing, it is also necessary to note some of the pitfalls of the current design that were made more than apparent in testing: the current design is *not* hermetically sealed, in order to make efficient use of this type of propulsion (including reducing noise and thrust), much higher tolerances must be used in the manufacturing (and thus financial support must be obtained for the next phase of the project), and the buoyancy must be controlled (as is the demonstrator has a very high degree of positive buoyancy, thus causing SHEILA to float during testing). The upside to these negative results is that any bad things encountered in this phase will directly influence what will be learned from this phase (and thus what can be done in the next phase), none of these results were unforeseen (this is after all simply a propulsion *demonstrator*), and all of these results are things which can be fixed in the future; as a matter of fact, these are all things that currently have conceptual solutions in the planning stage for the next phase.

Lastly, it is important to comment on the time/effort required in this phase; this commenting will be done in reference to the figure below:



	Hours	Days	Weeks	Average Hours Per Day
Team Total	875	36.46	5.21	
Member Average	175	7.29	1.04	5

Moving down the list of subsections, the following may be noted:

- The concept phase was quite short in duration; this is in no way indicative of the level of thought that has gone into this project, this underwater vehicle (UV) concept is actually nearly five years old in planning (the initial concept was dreamt up in late 2009/early 2010).
- The calculations phase was lengthy, but not as long as it could have been; there does not exist a magical set of calculations that corresponds perfectly to this means of propulsion. This is to say that this concept is highly unconventional and many theories had to be applied, mixed, and adapted in order to do any calculations at all (and the ones done thus far in this phase have been simplified since there is much unknown about this type of flow or how to apply it).
- The design process was highly iterative and thus actually existed in some form or another throughout every stage of the project; the listed 200 hours is only considering meetings that were *purely* called with design-intent in mind.
- The manufacturing process was extremely difficult and time consuming as this really has not been done before, and thus even with having gone to great lengths to think the manufacturing through as much as possible prior to starting, it was simply impossible to foresee all the issues that the team came across. In addition to this, the design and manufacturing stages were highly intertwined as redesigns were constantly necessary to solve issues come across in manufacturing.
- Testing went off for the most part without a hitch in the end; initially there was a great deal of trouble with testing, and the demonstrator had to be torn down, modified, and built back up quite a few times, but eventually very highly promising results were obtained through testing.
- Lastly, it should be noted that the total man hours listed above is *highly* conservative; to give some sort of idea of the time involved in this project, it will be noted that the weekend of Friday May 9th to Sunday May 11th contained three back-to-back 15 hour days (that's 15 hours per member present per day). Weeks like this were quite typical and over the 35 day project period (April 15th-May 19th) many meetings ran to about 0400 the following day.

Roles and Responsibilities

	Role	Responsibilities
Brian Martin	Leader	Team leader for project. Designate tasks to group members, supervise design process, and assist in all aspects of the project. Create concept/preliminary drawings of system to build off and improve throughout the project. Report Roles: Background and Design Process
Ben Saletta	Member	Lead in design meeting. Coordinator for manufacturing methodology and material purchasing. Assist in drawing package, manufacturing, and design. Report Roles: Marketability and Design Process
Abraham Paucar	Member	Primary lead in stress calculations pertaining to gear strength, lead for manufacturing troubleshooting/innovation. Assist in manufacturing, and design. Report Roles: Introduction and Design Process
Ketton James	Member	Primary coordinator for project scheduling and timeline; in charge of compilation of final report. Assist in gear drawings, manufacturing, and design. Report Roles: Project Schedule, Roles & Responsibilities, & Design Process
Andrew Blancarte	Member	Lead in initial testing/troubleshooting of system. In charge of financial reimbursement/allocation of project funds. Assist in design and drawings. Report Roles: Competition and Design Process

* The actual writing of the report is a team effort and thus everyone has worked on the parts of the report as listed in the above table

OK

Schedule

Draft Schedule made on 4/15/14 updated to reflect any changes along the way.

Master Project Draft Schedule — Drafted: 4/15/2014

Team UV	Brian Martin, Ketton James, Andrew Blancarte, Abraham Paucar, Ben Saletta	
ECD	**RAA**	**Description**
4/16/2014	Ben	Brainstorming Session/Design Requirements/Subsystem Decomposition
4/17/2014	Andrew	Preliminary Draft Report
4/18/2014	Brian	Conceptual Renderings
4/20/2014	Ketton	Configure Pricing Requirements/Calculations Deadline
4/22/2014	Abraham	Purchasing Deadline
4/26/2014	Ben	Manufacturing Start
5/6/2014	Brian	Manufacturing Deadline
5/6/2014	Andrew	Testing/Troubleshooting/Report/Final Drawings Period
5/6/2014	Abraham	Draft Report #1
5/13/2014	Keton	Draft Report #2
5/20/2014	Brian	Testing/Troubleshooting/Report/Final Drawings Period
5/20/2014	Team	Project Duedate

Contact Information

Name	Phone	Email
Brian Martin	(818) 282-5186	bmartin@csupomona.edu
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Abraham Paucar	(562) 810-3002	aipaucar@csupomona.edu
Andrew Blancarte	(909) 510-0593	aeblancarte@gmail.com
Ben Saletta	(760) 877-9860	bdsaletta@csupomona.edu

Project Time Schedules

Name	M	Tu	W	Th	F	Sat	Sun
Andrew	n/a	1500>	n/a	1500>	1100>	All	All
Abraham	1200-1600; <1800	1200-1600	1200-1600; 1800>	1200-1600	1200-1600	<1400	n/a
Ketton	n/a	1100-1600	n/a	1100-1600; 1715>	Appointment	All	All
Ben	<1600; 2200>	n/a	<1600; 2200>	n/a	All	All	All
Brian	<1200; 1400>	<1715	<1200; 1400>	n/a	All	All	All

Good.

Marketability

First and foremost this product is a proof of concept for an underwater propulsion system. While it can be a final product for some very limited applications it is meant to be a component in a larger scale system or attached to control surfaces. The propulsion system itself cost around $700 (approximately 88% of the initial projected budget of $800) and about 50 hours of work a week for 5 weeks. The cost is quite low for a project of this scale because the vast majority of the components are constructed using standard pipe or ducting sizes and commercial off the shelf components (COTS). There are only eleven custom parts in the whole design which makes the product cheaper and easier to manufacture. Ten of these custom parts were 3D printed gears. These could easily be replaced with steel or brass gears in the future with a minimal upfront investment. The final custom component was the vane shell. This part contains some significant complexity due to the helix shape of the fin blades. In order to mass produce this part, a custom die would have to be made and this would cause a significant upfront investment cost. If another of these propulsion systems were to be made it would cost around $400; the prototype cost a bit more due to material waste and finding the proper manufacturing processes for the vanes and the shells. Overall there was not much wasted money because the design process was well thought out and involved detailed designs. There would be a significant tooling investment for the mass production of these propulsion systems due to the need for custom parts such as the gears and the vane shell approximately $10,000 for all of them, 3 gear dies at $1,500 and a custom vane die at $5,000.

While this is only a proof of concept demonstrator, there are no other products on the market that use a propulsion system similar to this design. All other underwater systems use a propellor based propulsion system. Bladed propellers (while historically the propulsion system of choice) can cause problems due to their high susceptibility to cavitation, their tendency to foul upon collision with cable or kelp depending on the size of the application, may cause turbulent flow, and produce a significant amount of noise that can lead to detection of the device (which may be problematic in a stealth environment). This new propulsion system uses a relatively low speed but adds energy into the flow for a longer time and thus enables higher thrust without the threat of cavitation, while also significantly reducing turbulence in the flow, and the noise typically associated with the choppy flow of a propeller. The rotating components are enclosed which prevents fouling due to the debris in the water. The stealth and robustness of this design makes it ideal for research applications where the water is full of debris. This could also be applied to the navy with the stealth characteristics this device would look like a tuna to most detection devices.

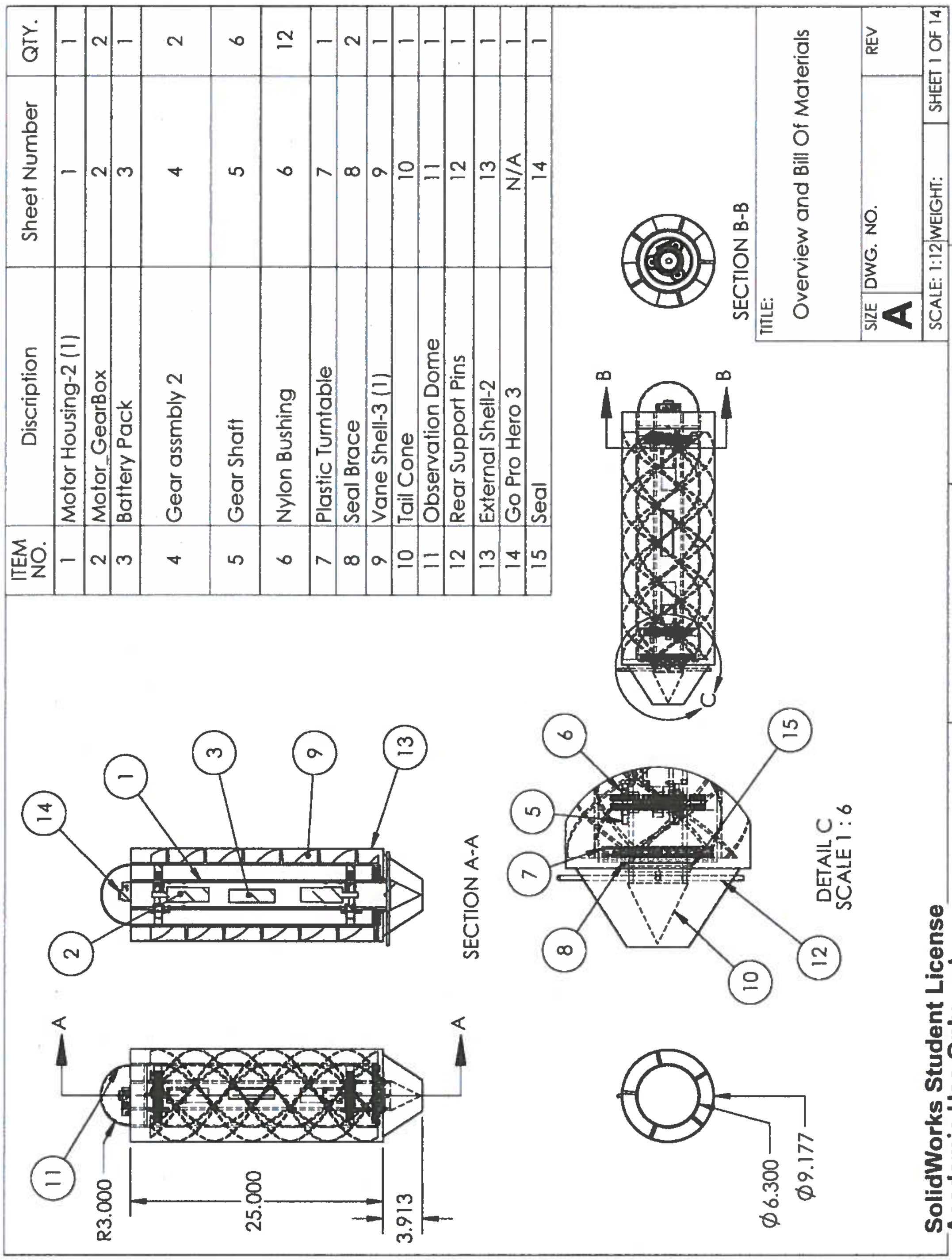
Good.

Conclusion

The original objective of this project was to explore the feasibility of using a new, innovative type of underwater propulsion for an underwater vehicle; in this respect, the project was a great success. SHEILA-D provides proof that water can be moved in order to produce a significant amount of thrust in a steady, even manner by rotating it through an annular region along an extruded length. The mechanics associated with this type of flow are little known and have been left virtually unexplored and painfully neglected by the world of marine and sub-marine propulsion. The propulsion demonstrator described in the preceding pages serves a function so as to merely glimpse through the proverbial looking glass into what this untapped innovative source of propulsion has to offer. This means of propulsion has the latent potential to revolutionize how underwater vehicles maneuver through and explore the world's oceans, lakes, and other bodies of water. From a quick, stealthy Information, Surveillance, and Reconnaissance (ISR) drone that may be used by US Navy SEALs behind enemy lines to a hidden-in-plain-sight observation vehicle that may be used by marine biologists to study marine life in an unimposing manner, the future underwater vehicle as will be developed from SHEILA-D will have the ability to provide unprecedented capabilities in a very diverse field of applications. This seemingly straight-forward, yet extremely complex, well thought out, comprehensively developed demonstrator project will have huge implications as it is carried through into the next phase of its existence as it is adapted into a fully operational underwater vehicle.

Good.

Appendix A: Drawings



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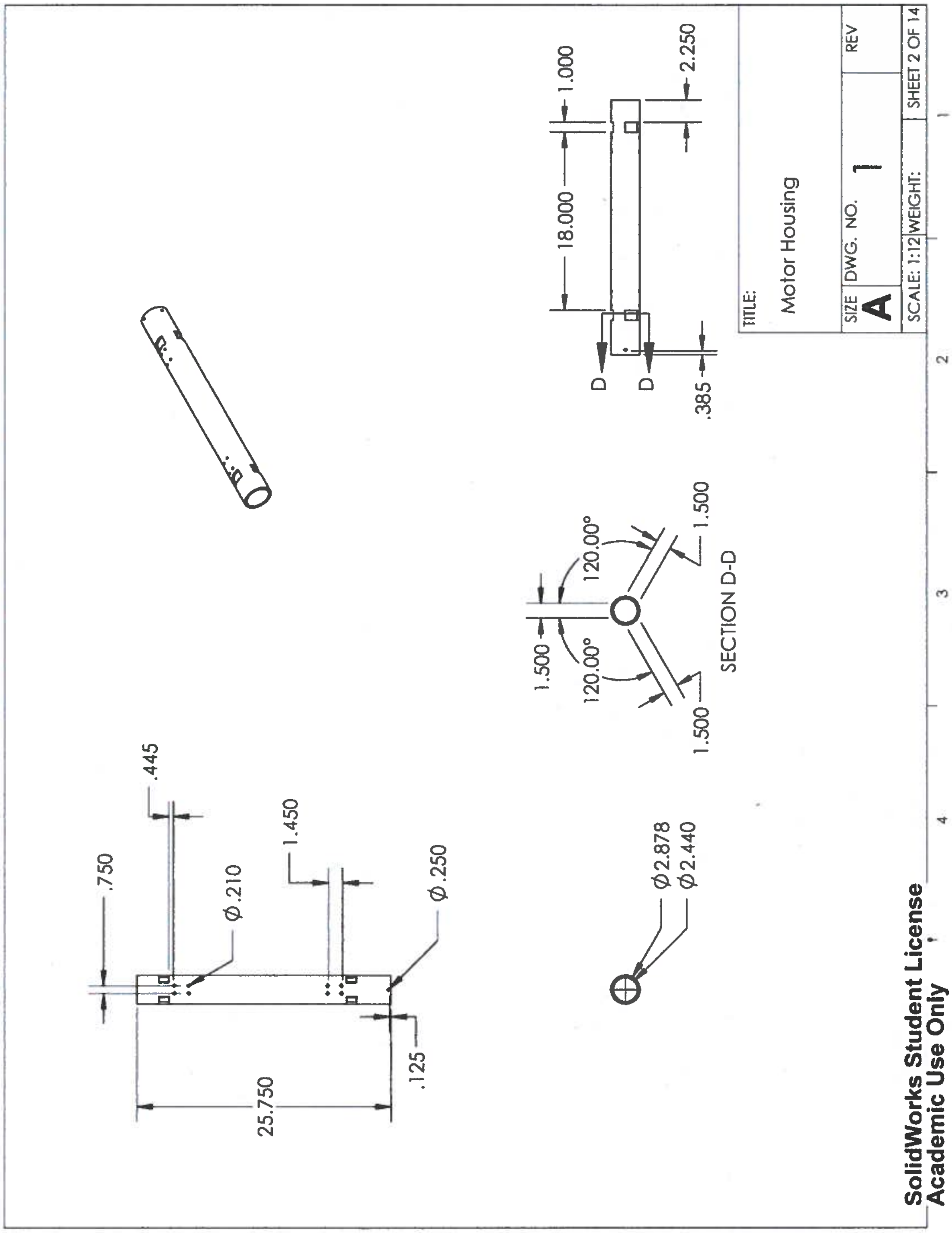
5	Gear Shaft	5	6
6	Nylon Bushing	6	12
7	Plastic Turntable	7	1
8	Seal Brace	8	2
9	Vane Shell-3 (1)	9	1
10	Tail Cone	10	1
11	Observation Dome	11	1
12	Rear Support Pins	12	1
13	External Shell-2	13	1
14	Go Pro Hero 3	N/A	1
15	Seal	14	1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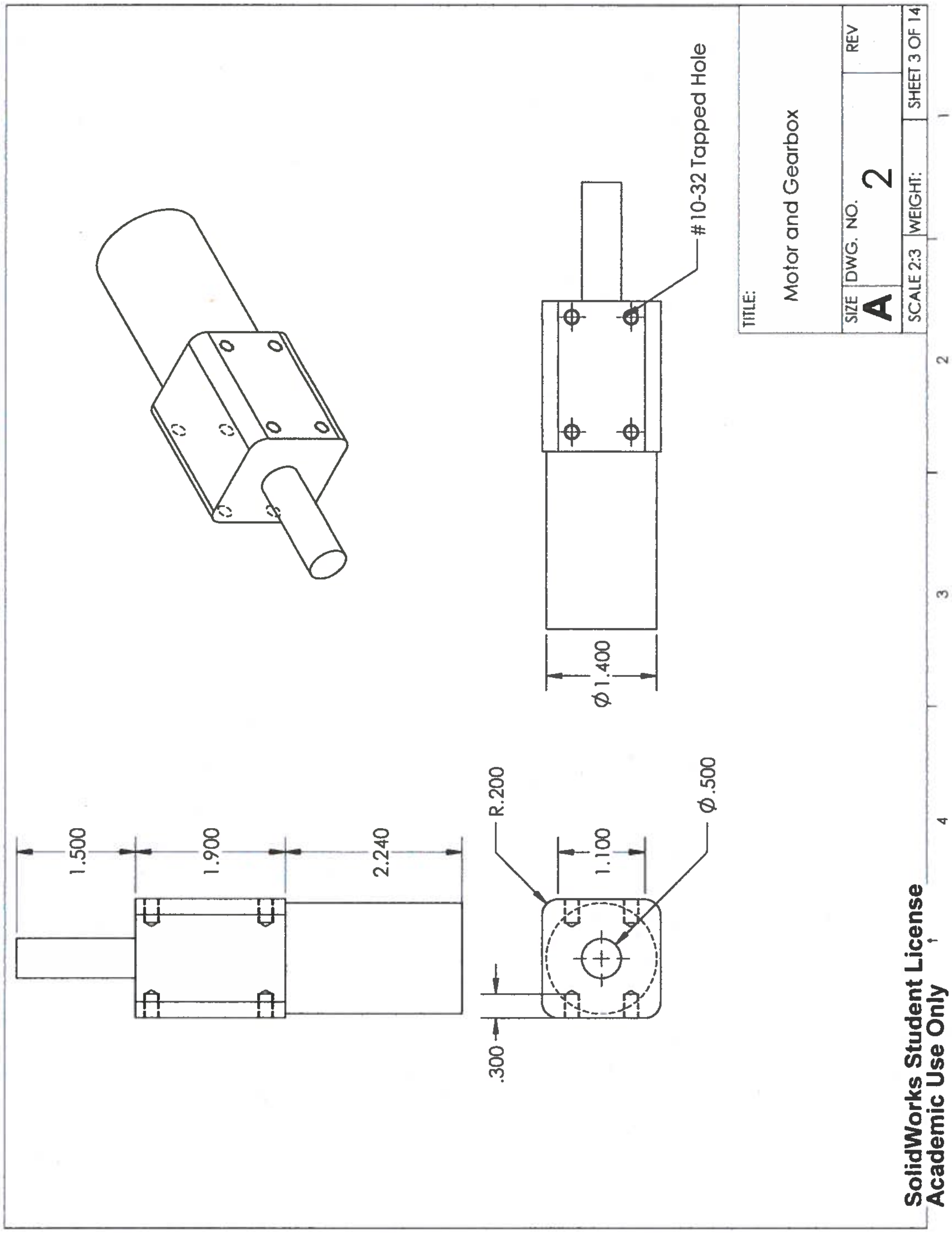
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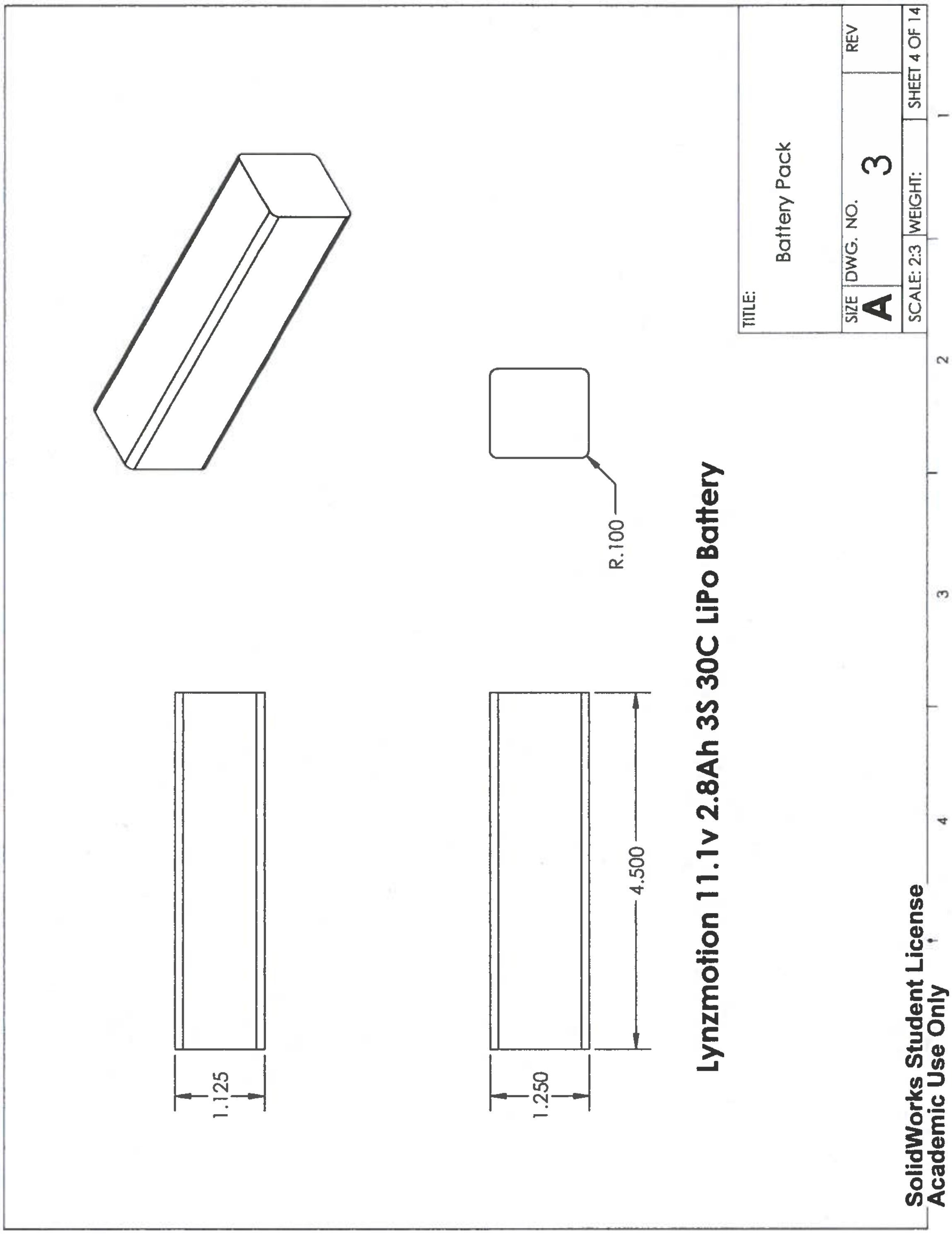
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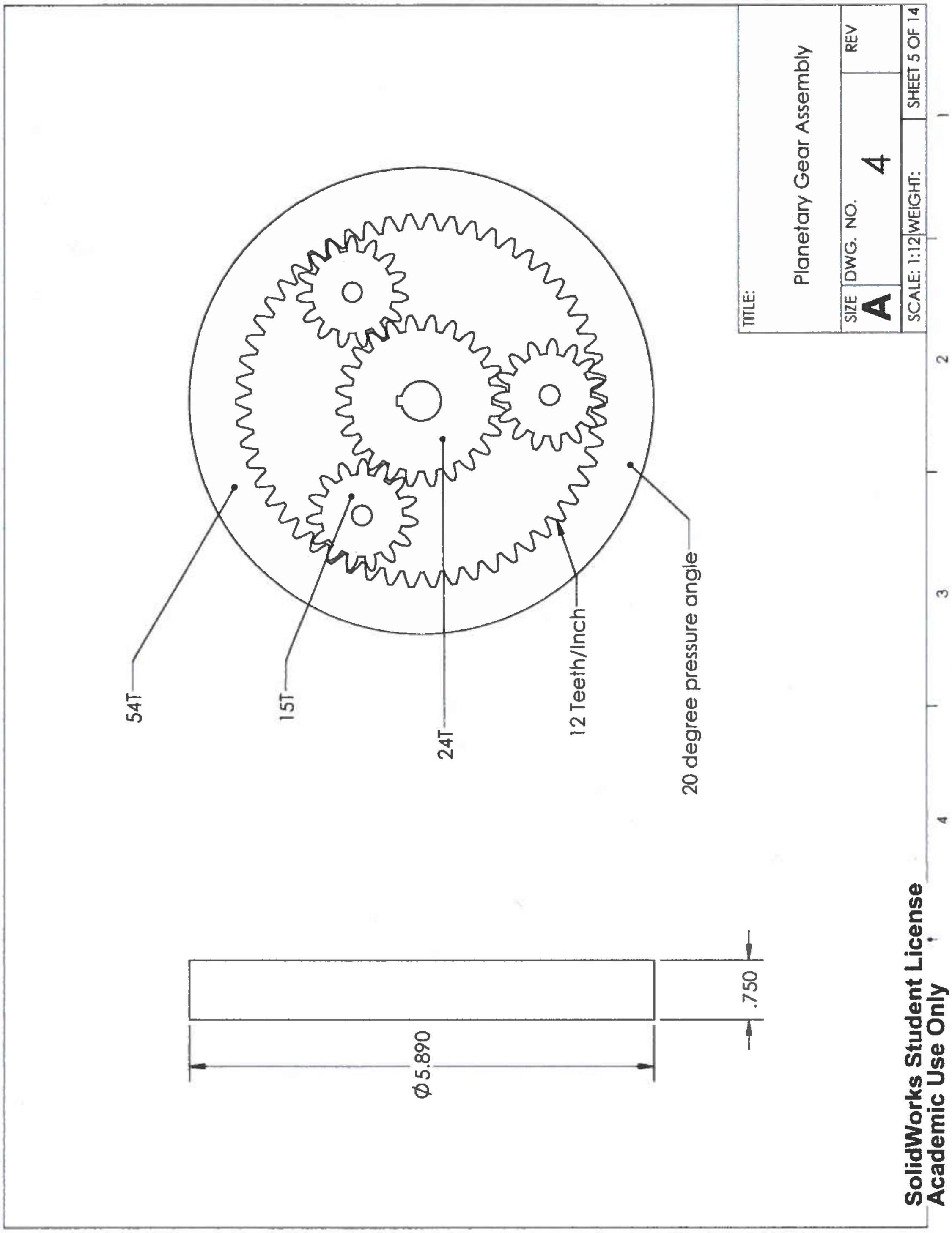
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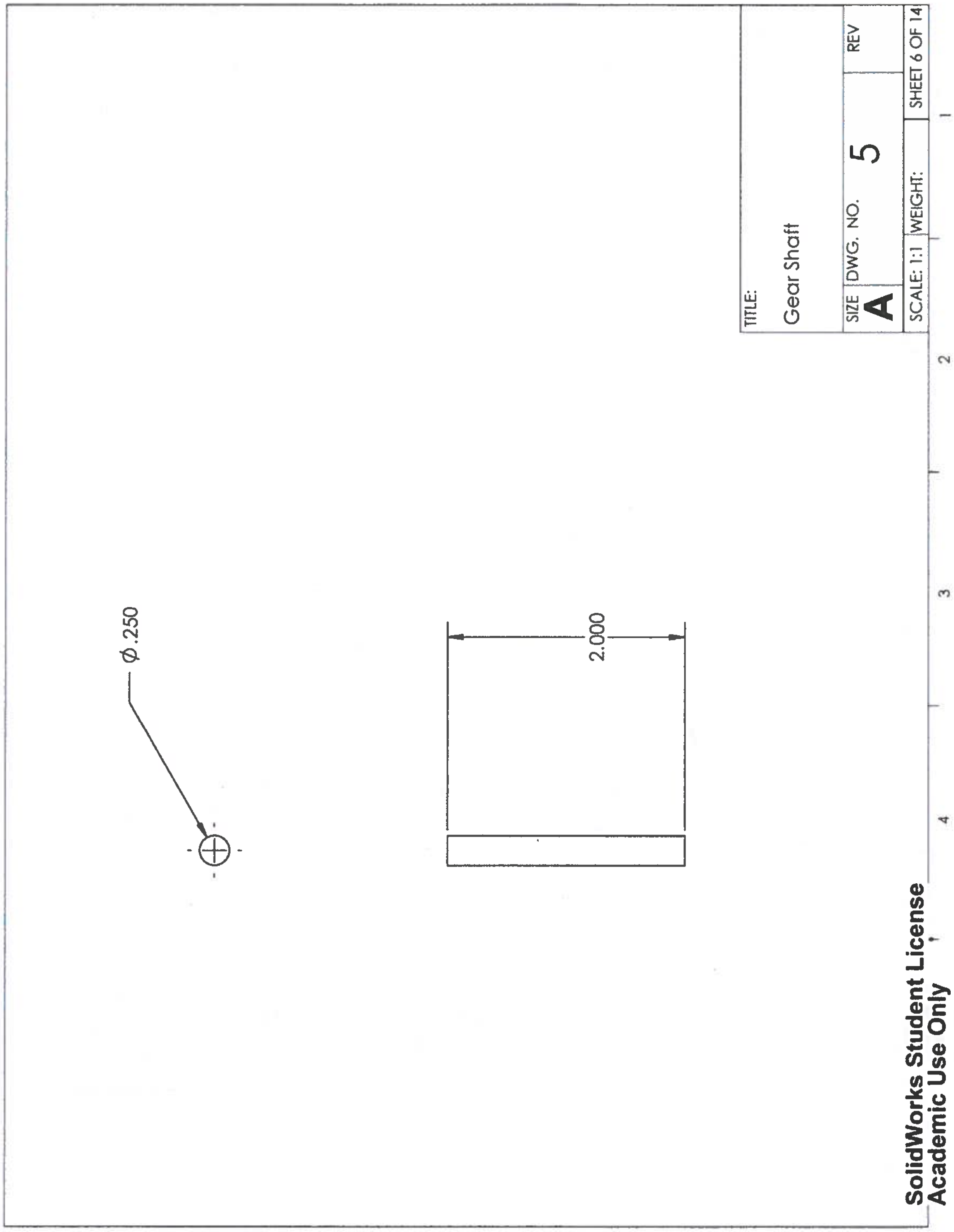
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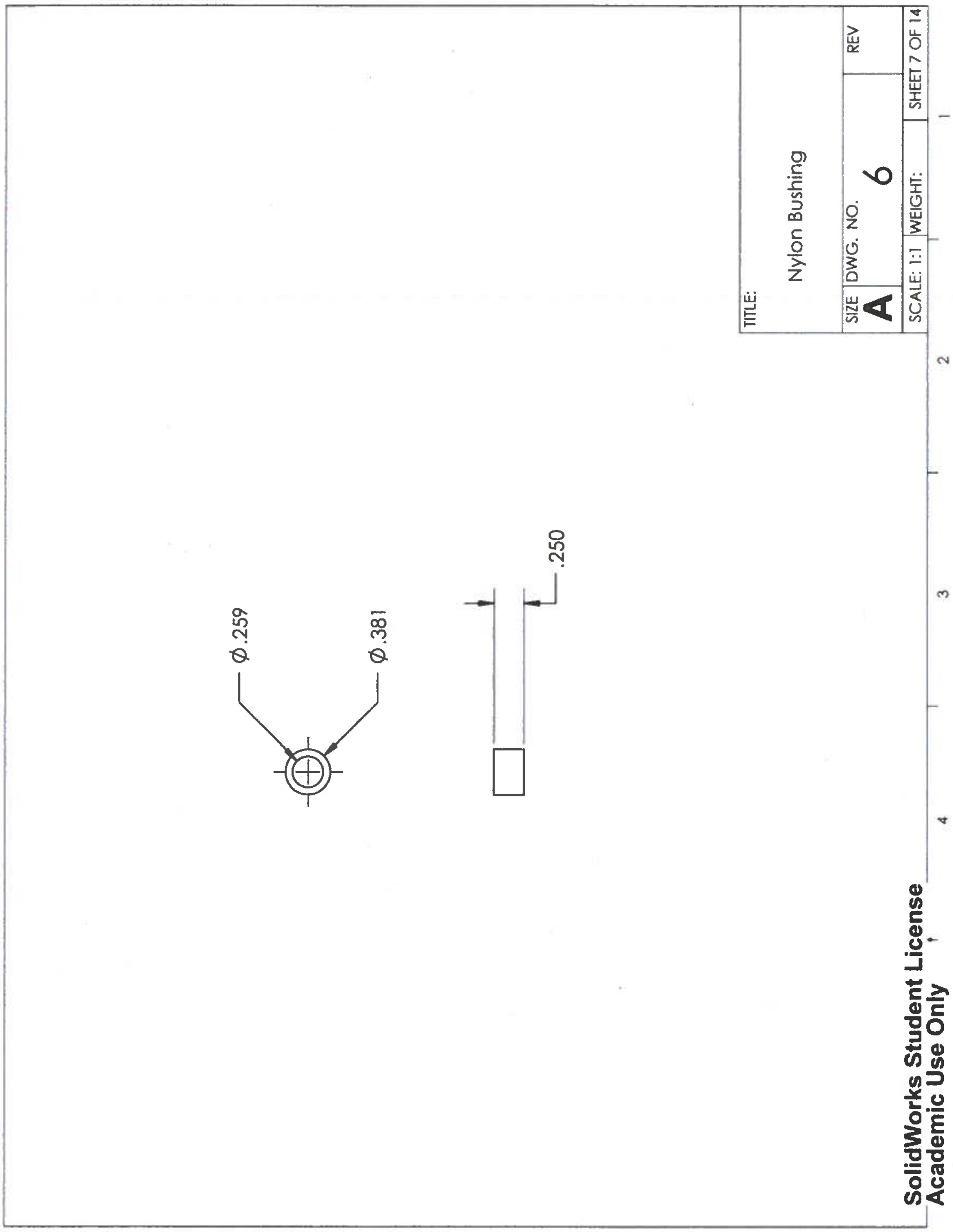
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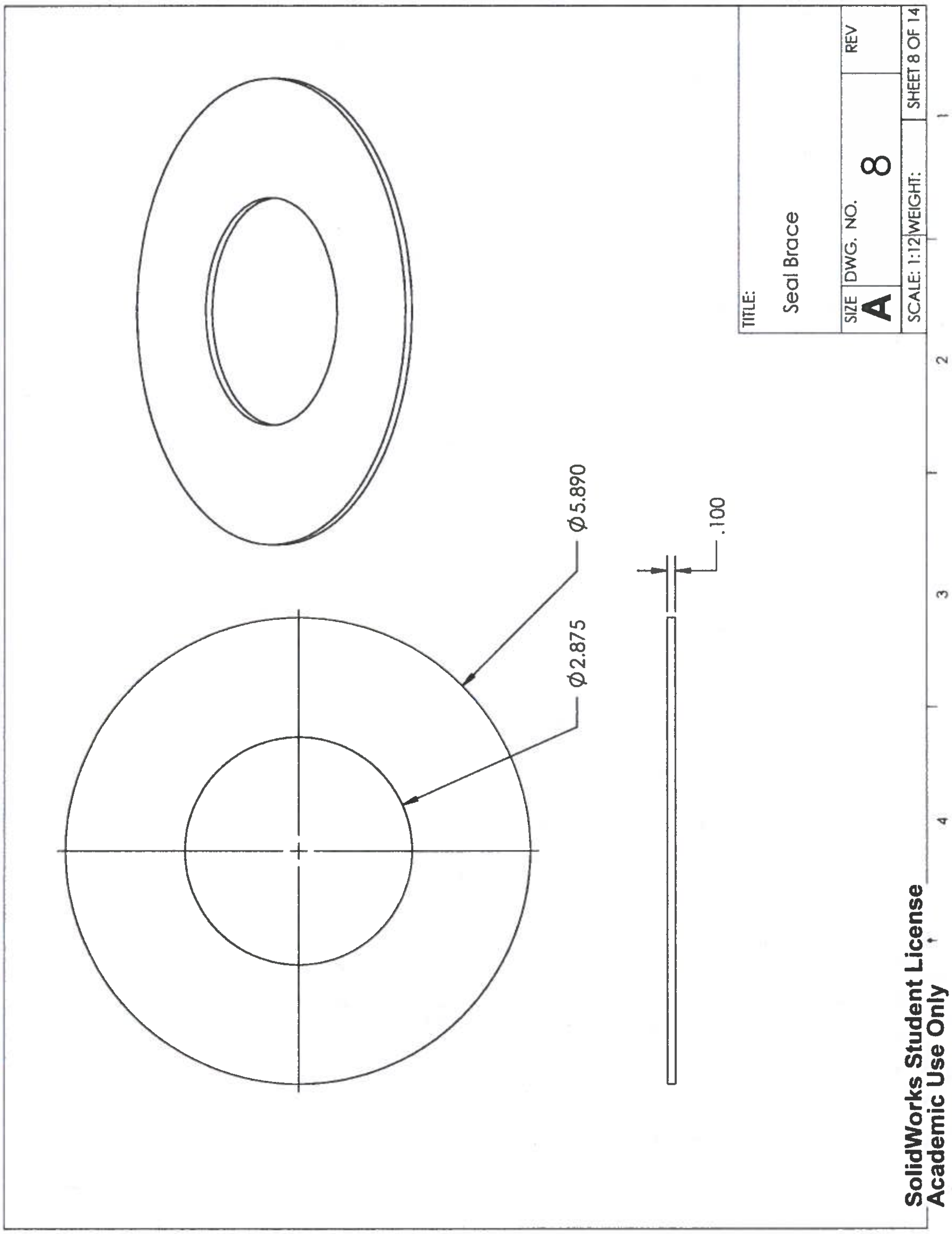
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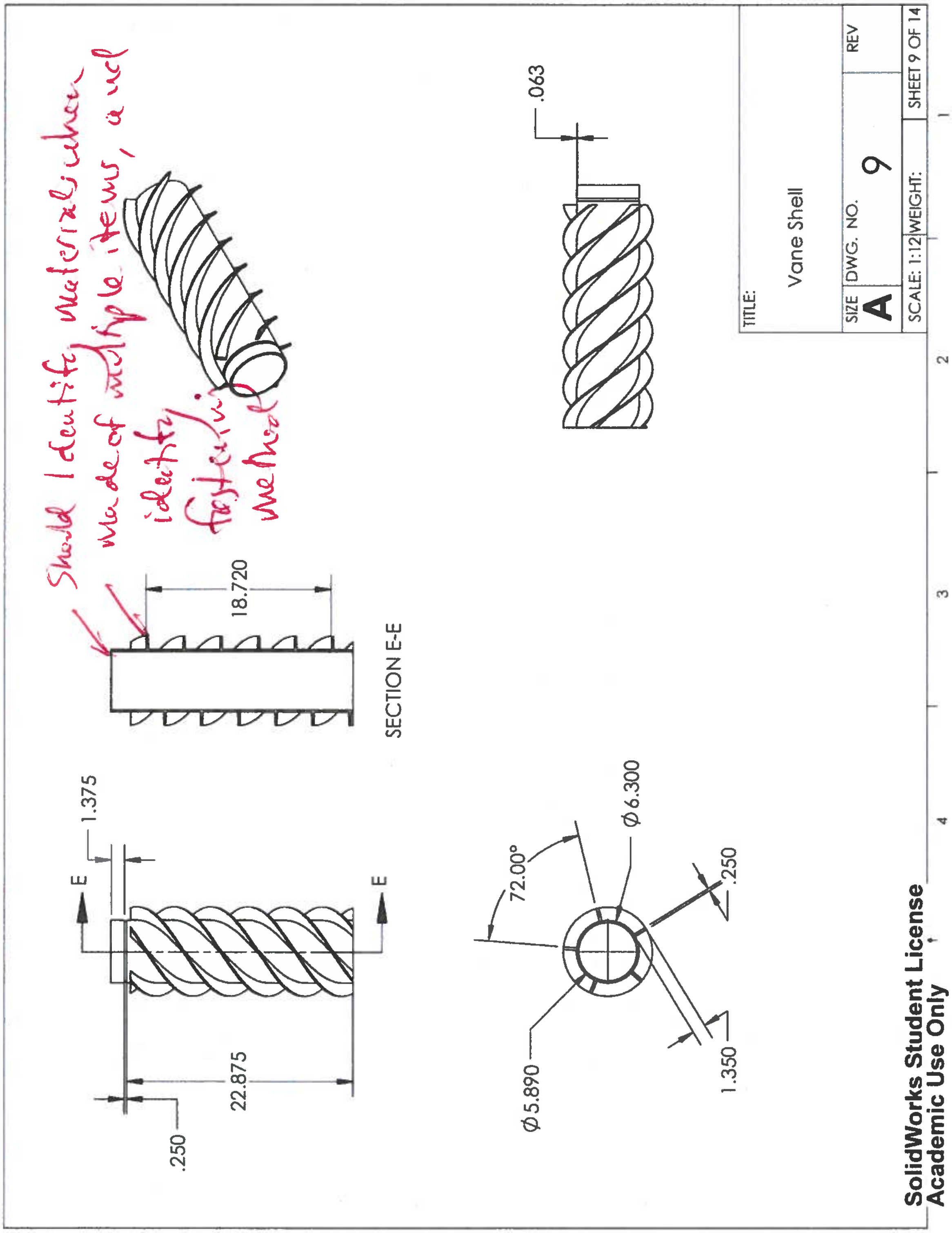
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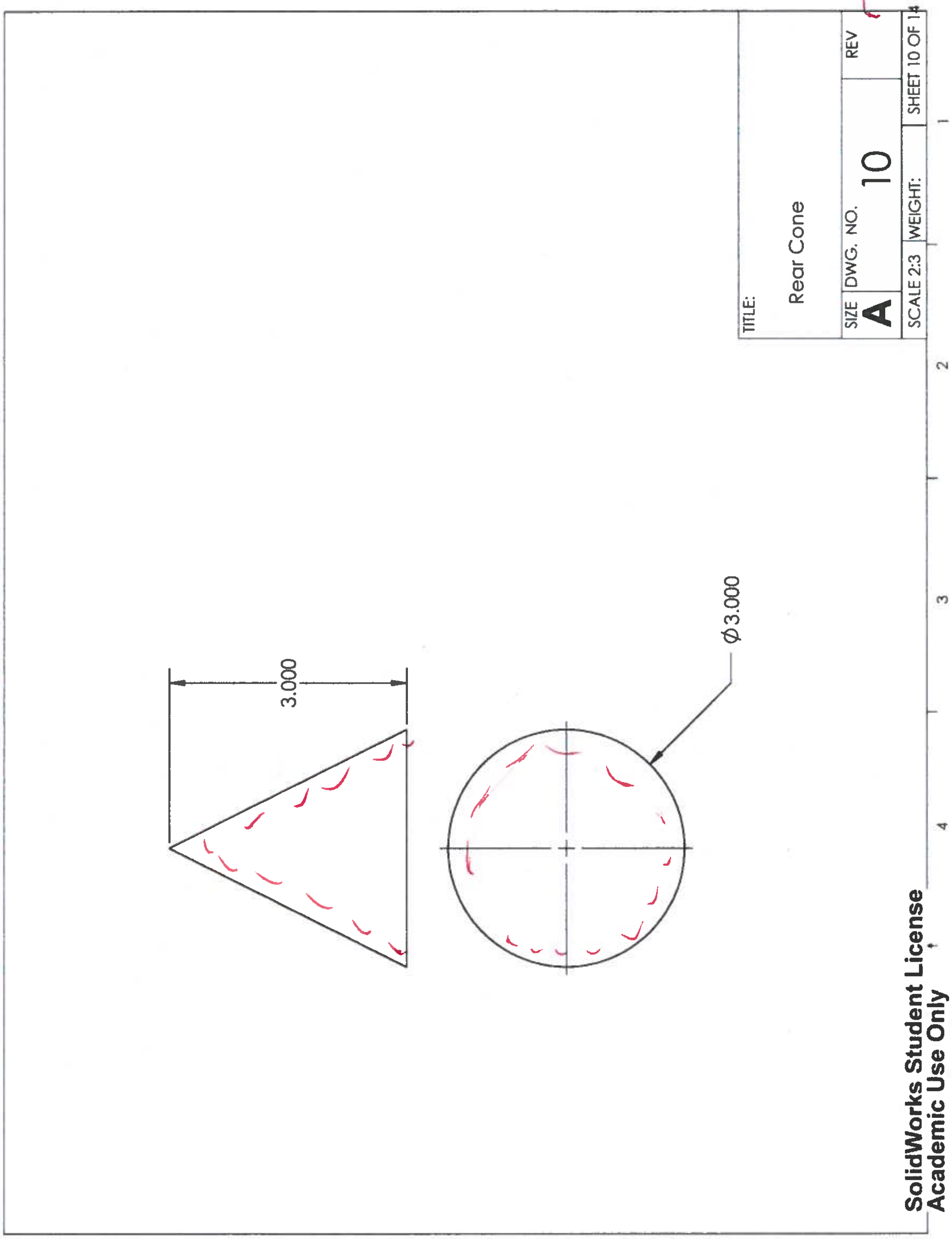
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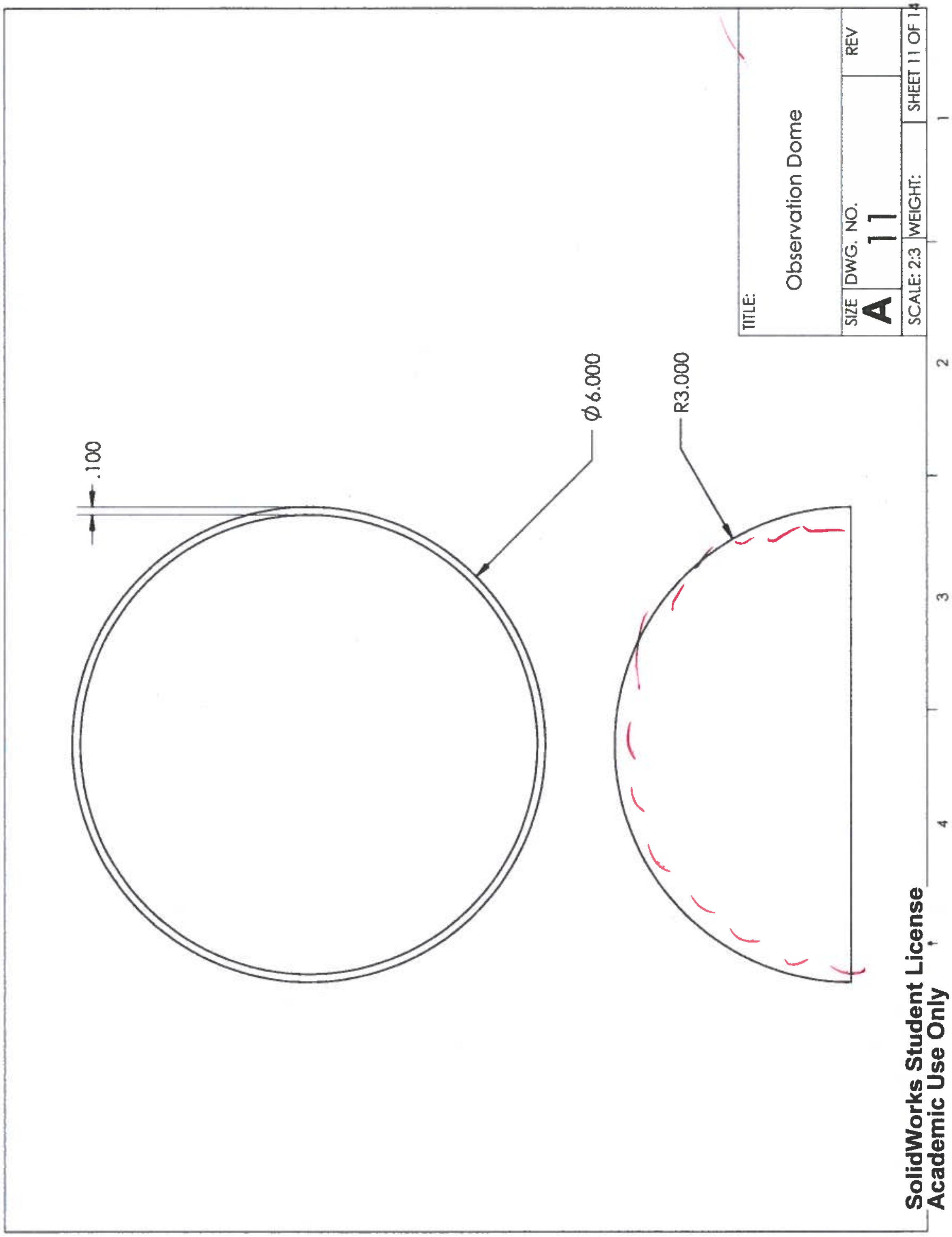
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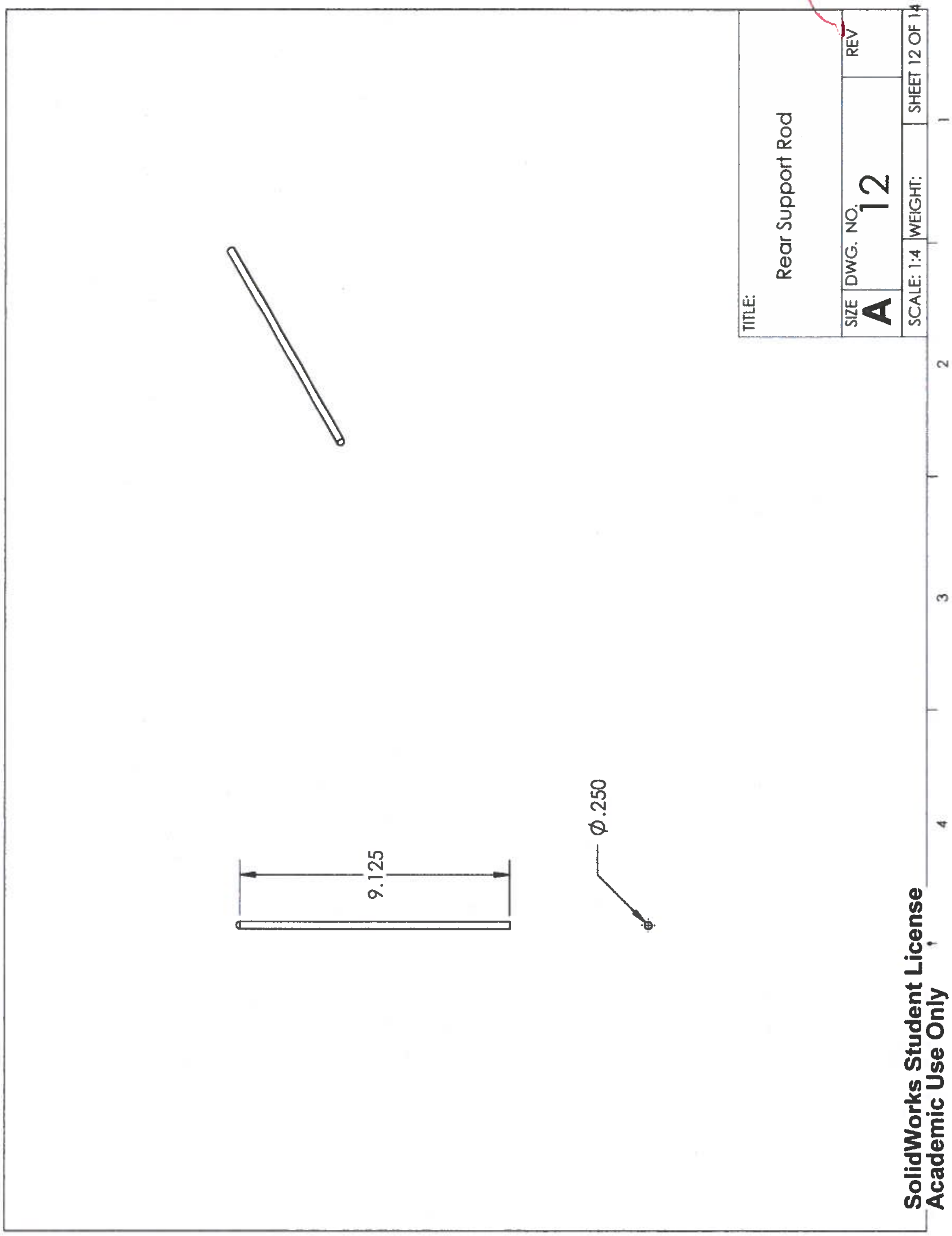
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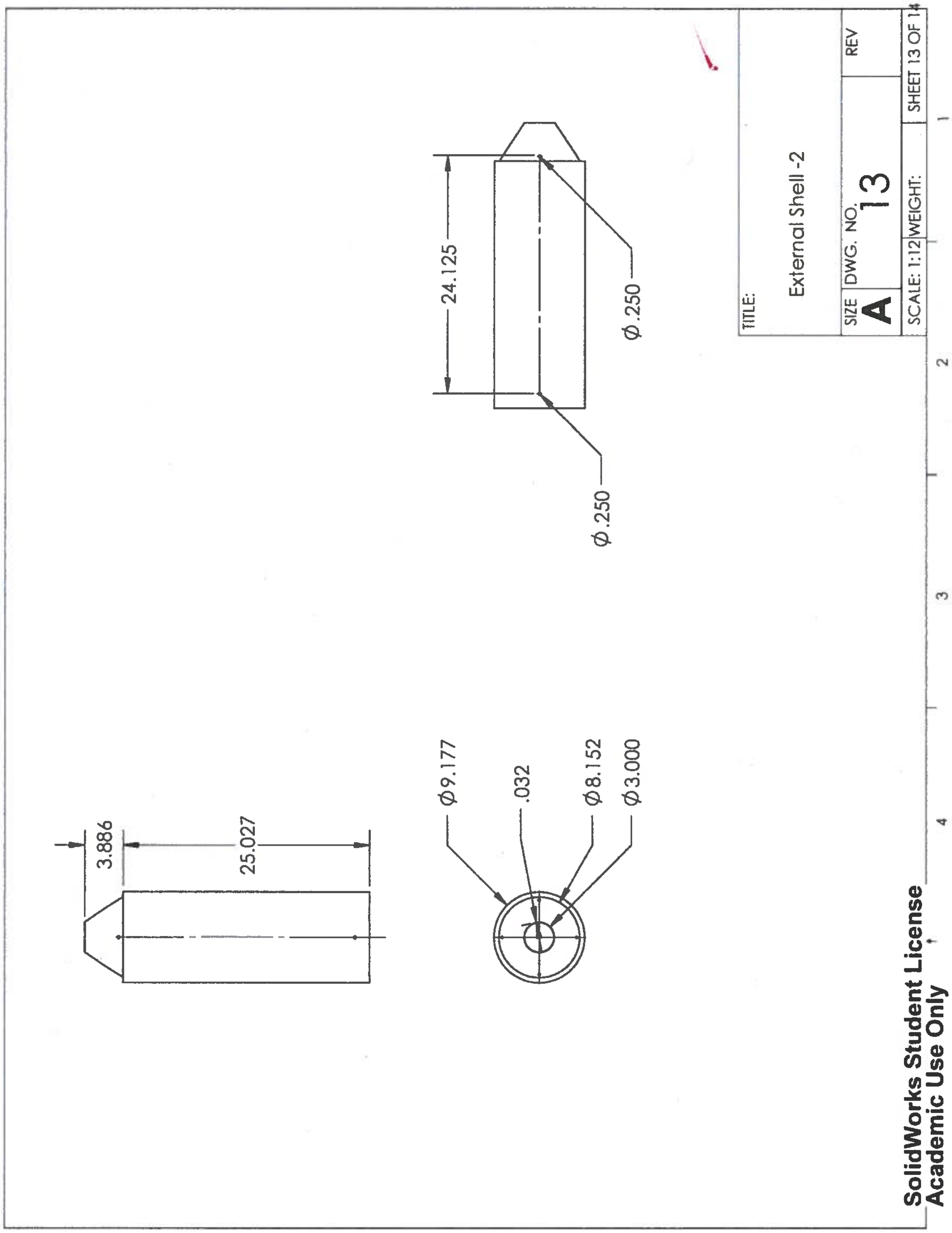
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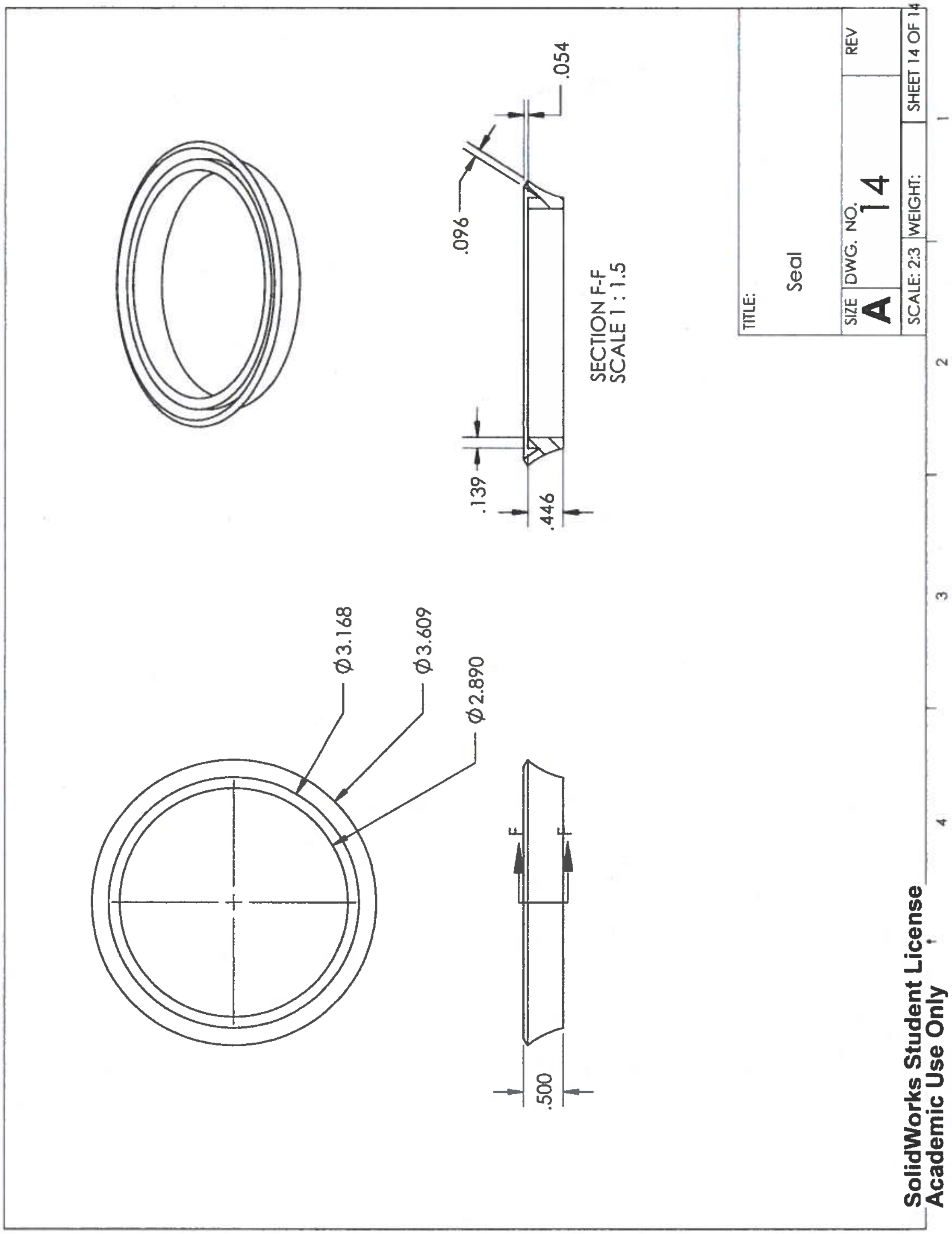
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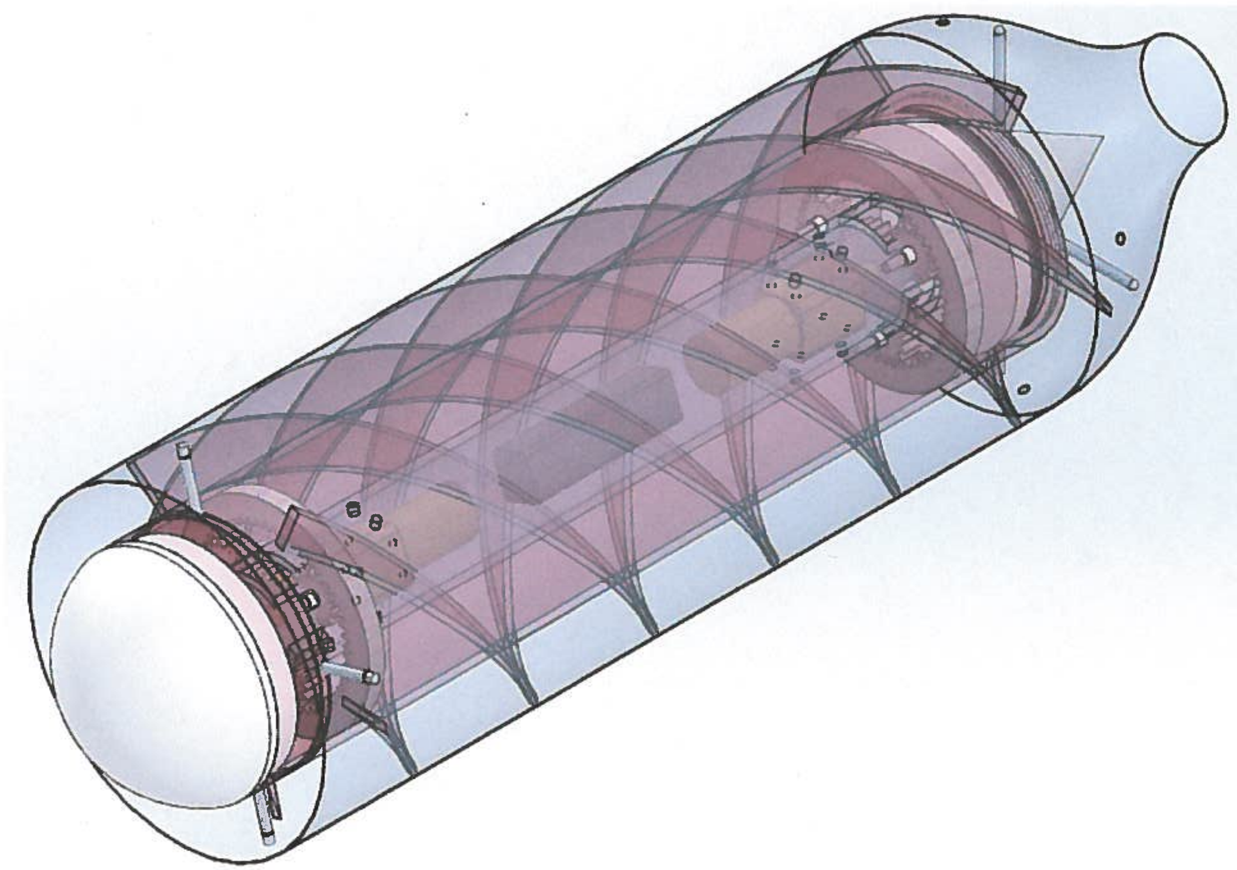


Figure 1: Isometric View of SHEILA-D

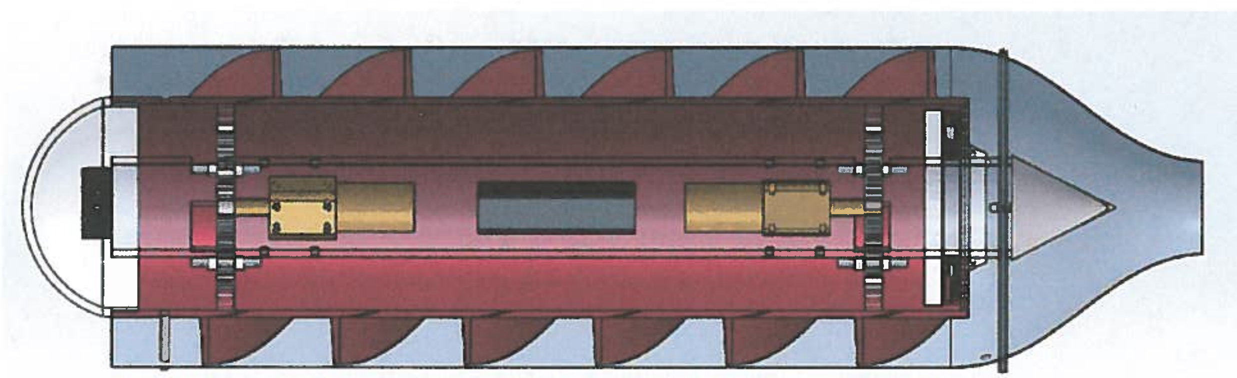
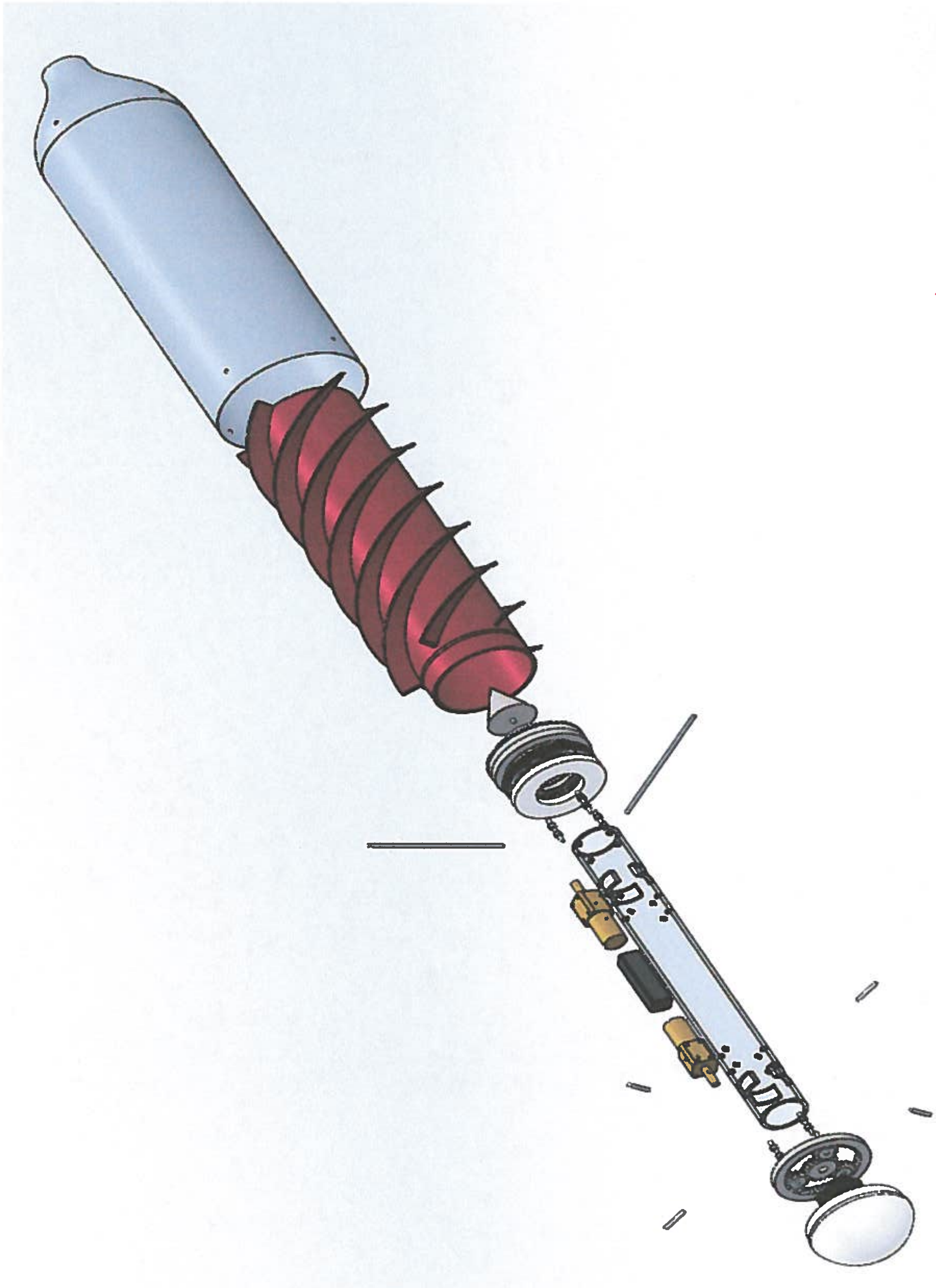


Figure 2: Cutaway Side View

Nice View



Nice view

Figure 3: Exploded View of SHEILA-D

Appendix B: Calculations

The following pages contain the calculations performed with regards to the design of an innovative water propulsion system, namely SHEILA-D (Submerged Hydrodynamically propelled Explorer, Implementation: Los Angeles – Demonstrator). The calculations in the following pages represent an interdisciplinary approach to the development of a propulsion system based off of mechanics not currently utilized for this application; that is, this design strays from the traditional use of a propeller, necessitating the use of the aforementioned interdisciplinary approach.

An initial set of hand calculations was performed in order to develop the necessary equations, after which the equations were programmed into Excel in order to facilitate a more efficient iterative technique, which was then used in order to select design parameters that produced the optimal design points/results. This package is organized in accordance with the following list of sections:

- Fluid mechanics hand (sample) calculations
  - Input, Output
  - Calculations
- Classical mechanics hand (sample) calculations
  - Input, Output
  - Calculations
- Machine design hand and typed (sample) calculations
  - Input, Output
  - Calculations
- Excel spreadsheet iterative results page
- Performance Design Curves
- Stall Torque Variation Plot

Fluid Mechanics

1. Fluid Mechanics

Inputs: Speed, Vane Geometry, Nozzle, Water Mass

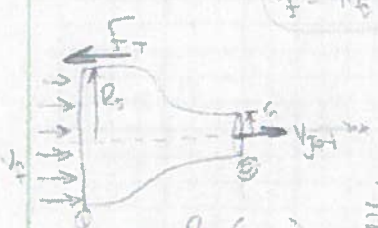
Outputs: Thrust, Velocity, Friction Losses, Running Torque

ϕ = Vane Angle

$$F_D = \frac{T}{r_{avg}} \cos(90° - \phi)$$

$$\int_{v_0}^{v_f} v\,dv = \int_0^L a\,dx \rightarrow \frac{V_f^2 - V_0^2}{2} = aL$$

$$V_f = \sqrt{V_0^2 + 2 r_{avg} \frac{TL}{m} \sin\phi}$$



$$\dot{V} = V_0 A_s = V_J A_n \rightarrow \boxed{V_J = \frac{A_s}{A_n} V_0} ; \; A_s = \pi(r_o^2 - r_i^2) ; \; A_n = \pi r_n^2$$

$$\frac{\partial}{\partial t}\int_{CV} V\rho\,d\forall + \int_{CS} V\rho\, V\cdot \hat{n}\, dA - P_{①}\pi R_s^2 + F_T - P_{②}\pi r_n^2$$

0 (Steady Flow) ; 0 (Neglect)

$R_s = (r_o - r_i)$

$$\dot{m}(V_2 - V_1) = P_{①}\pi R_s^2 + F_T$$

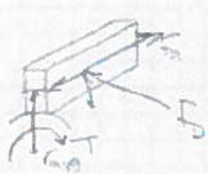
$$-F_T = \dot{m}(V_1 - V_2) + P_{①}\pi R_s^2$$

$$F_T = \dot{m}\left(V_0 - \frac{R_s}{r_n} V_0\right) + P_{①}\pi R_s^2$$

$$\boxed{F_T = \rho V_0^2 R_s^2 \pi \left(1 - \frac{R_s}{r_n}\right) + P_{①}\pi r_s^2}$$

$$V_{avg} = \frac{V_0 + V_0}{2} ; \; \boxed{h_L = f\frac{L}{D}\frac{V^2}{2g}} ; \; Re = \frac{VD}{\nu} ; \; f = \left[-1.8\log\left(\left(\frac{\varepsilon/D}{3.7}\right)^{1.11} + \frac{6.9}{Re}\right)\right]^{-2}$$

ε drawn tubing = 5(10^-6) ; $D_h = \frac{4A}{P}$; $\nu = 1.407(10^{-5})\, ft^2/s$



$$\omega = RPM \cdot \frac{2\pi}{60} \rightarrow V_{avg} = r_{avg}\,\omega$$

$$\frac{P}{\gamma} = \frac{V^2}{2g} \text{ @ Stagnation pt. on plate}$$

$$\frac{P}{\gamma} = \frac{V_{avg}^2}{2g} \rightarrow F_D = \frac{V_{avg}^2}{2g}\gamma A \rightarrow T = \frac{F_D \cos\phi}{r_{avg}}$$

$$\boxed{T = \frac{r_{avg}\left(RPM\left(\frac{\pi}{30}\right)\right)^2}{2g}\gamma A \cos\phi}$$

Classical Mechanics

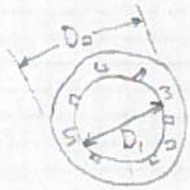
2. Classical Mechanics

Inputs: Vane geometry, Annular Region, Shell Length, Material Properties
Output: Total required stall torque

$$\boxed{\begin{aligned} L &= 20.85\,in \\ \theta &= 20^\circ \end{aligned}}$$

S_1 = length of vane

$$\cos\theta = \frac{L}{S_1} \rightarrow \boxed{S_1 = 22.19\,in}$$



$$\boxed{\begin{aligned} D_1 &= 6.25\,in \\ D_2 &= 9.12\,in \end{aligned}}$$

S_2 = arc length between vanes on inner cylinder
S_3 = arc length between vanes on outer cylinder

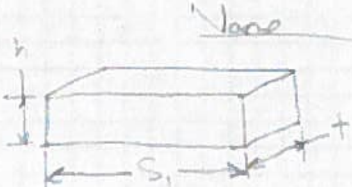
$$S_2 = \frac{\pi D_1}{5} \rightarrow \boxed{S_2 = 3.93\,in}$$

$$S_3 = \frac{\pi D_o}{5} \rightarrow \boxed{S_3 = 5.73\,in}$$

Vane Cross Section

$$\boxed{\begin{aligned} h &= 2\,in \\ t &= 0.25\,in \end{aligned}}$$

Vane



$$\boxed{\gamma_{PVC} = 0.0469\,\frac{lbf}{in^3}}$$

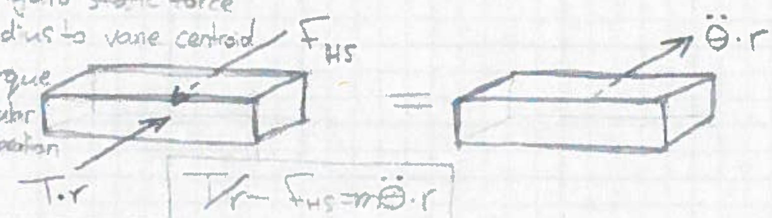
$$m = \frac{\gamma}{g}\forall_{vane}$$

$$= \frac{0.0469\,\frac{lbf}{in^3}(2\,in)(0.25\,in)(22.19\,in)}{32.2\,ft/s^2}$$

$$\boxed{m = 0.0162\,lbm}$$

FBD of Individual Vane

F_{HS} = Hydro Static Force
r = radius to vane centroid
T = Torque
$\ddot{\theta}$ = angular acceleration



$$\boxed{T/r - F_{HS} = m\ddot{\theta}\cdot r}$$

$$F_{HS} = \gamma_{H_2O} \, V_{H_2O}$$ — Between Vanes

$$\boxed{\gamma_{H_2O} = 62.4 \frac{lbf}{ft^3}}$$

$$F_{HS} = \gamma_{H_2O}\left(\frac{\pi}{4}D_2^2 - \frac{\pi}{4}D_1^2\right)\left(\frac{72°}{360°}\right)(L)$$

$$= 62.4 \frac{lbf}{ft^3}\left[\frac{\pi}{4}(1.12in)^2 - \frac{\pi}{4}(6.25in)^2\right]\left(\frac{1}{5}\right)(20.25in)$$

$$\beta = \frac{360°}{5} = 72°$$

$$F_{HS} = 9015 \frac{lbf}{ft^3} in^3 \left(\frac{ft}{12in}\right)^3$$

$$\boxed{F_{HS} = 5.22 lbf}$$

$$T/\left(\frac{D_1+b}{2}\right) - F_{HS} = m\ddot{\theta}\left(\frac{D_1+b}{2}\right)$$

(Assume 150 RPM max, 5 second acceleration

$$\ddot{\theta} = \frac{180 RPM}{5s\left(\frac{1min}{60s}\right)} \rightarrow \boxed{\ddot{\theta} = 1200 \frac{rev}{min^2}}$$

$$T/\left(\frac{6.25in + 2in}{2}\right) - 5.22 lbf$$

$$= 0.0162 lbm\left(\frac{32.2 lbf}{1bm}\right)\left(1800 \frac{rev}{min^2}\right)\left(\frac{2\pi}{rev}\right)\left(\frac{1min}{60s}\right)^2\left(\frac{6.25in + 2in}{2}\right)$$

$$\boxed{T_{vane} = 49.3 lbf \, in = 4.11 lb\cdot ft}$$

$$T_{total} = 5 T_{vane} FS$$

Assume FS = 1.25 to account for rotating cylinder inertia

$$= 5(4.11 lb\cdot ft)(1.25)$$

$$\boxed{T_{total} = 25.7 lb\cdot ft}$$

Machine Design (Hand Calcs)

Given: $N_{pin} = 24$ teeth $N_{pk} = 15$ teeth $N_{int} = 54$

$F = 0.5''$ measured

$k_o = 1$
Fig 14-17

Power source assumed uniform
Driven machine assumed uniform

$$K_v = \left(\frac{A + \sqrt{V}}{A}\right)^B = \left(\frac{45.43 + \sqrt{7063.5}}{45.43}\right)^{1.082} = \underline{2.116263901}$$

Assuming $Q_v = 3$ for low quality commercial gears

14-28

$$B = 0.25(12 - Q_v)^{2/3} = 0.25(12-3)^{2/3} = 1.081687178$$

$$A = 50 + 56(1-B) = 50 + 56(1 - 1.081687178) = 45.42551885$$

$$(V_t)_{max} = [A + (Q_v - 3)]^2 = [45.43 + (3-3)]^2 = 2063.5 \text{ ft/min}$$

$$K_s = 1.192\left(\frac{0.75\sqrt{0.337}}{12}\right)^{0.0535} = 0.9982$$

Table 14-2

$N_{pin} = 24 \Rightarrow Y = 0.337$

$$K_m = 1 + C_{mc}(C_{pf} C_{pm} + C_{ma} C_e) = \underline{1.291981969}$$

$C_{mc} = 1$ assuming uncrowned teeth

$$C_{pf} = \frac{F}{10d} - 0.025 = \frac{0.75}{10(2)} = \underline{0.0125}$$

$C_{pm} = 1$ assumed because we do not have info for non straddle mounted gears

$$C_{ma} = A + BF + CF^2 = 0.247 + 0.0167(.75) + (-.765(10^{-4}))(.75)^2 = 0.2595$$

Assuming open gearing condition

$A = 0.247$ $B = 0.0167$ $C = -.765(10^{-4})$

$C_e = 1$ assuming all other conditions

$t_R = .4090''$

$h_t = 0.1635''$

$$m_B = \frac{t_R}{h_t} = 2.5015290\ldots \Big\} \underline{K_B = 1}$$

$$\sigma_{Bending} = W_t K_o K_v K_s \frac{P_d}{F} \frac{K_m K_B}{J} = \boxed{4.6704\ ksi}$$

$T_{rotating} = 40\ lb\text{-}ft$ to the outside

$T_{p\ starting} = 18\ lb\text{-}ft \cdot \frac{12\ in}{1\ ft} = 216\ lb\text{-}in$

motor to gearbox

$r = 1\ in$

$$W_{t_p} = \frac{\frac{T}{r}}{\frac{3}{2}} = \frac{\frac{216\ lb\ in}{1}}{\frac{3}{2}} = 36\ lb$$

2 sets of gears

$E = 326\ ksi$

Assumming: weakest elastic coefficient

$C_p = 1650\ \sqrt{psi}$ ← this is incorrect.

$$\sigma_c = C_p \sqrt{W_t K_o K_v K_s \frac{K_m}{d_p F} \frac{C_f}{I}} = \boxed{53.161\ ksi}$$

C_f assumed to be 1 for lack of data

$\phi_t = 20°$

$$I = \frac{\cos\phi_t \sin\phi_t}{2 m_N} \frac{m_G}{m_G + 1} = \frac{\cos(20)\sin(20)}{2(1)} \cdot \frac{0.625}{0.625+1} = 0.061806501$$

$m_G = \frac{N_G}{N_P} = 0.625$

$m_N = 1$ for spur gears

$\sigma_{all\ Bending} = \frac{S_t}{SF} \frac{Y_N}{K_T K_R}$

$= 4.6704$ ksi

* unreasonable data from steel gears

flexural yield $= 8.793$ ksi

$\boxed{SF_{Bending} = \frac{8.7993}{4.6704} = 1.884}$

$\sigma_{all\ contact} = \frac{S_c}{S_H} \frac{Z_N C_H}{K_T K_R} = \boxed{53.107 \text{ ksi}}$

S_H: 2.64; C_H: (1); K_T: (1)

$S_c = 113.5 \times 1000$ lb/in   Fig 14-5 ← not correct due to lack of info for our mat'l.

$Z_n = 1.05$
assuming $N = 10^5$

C_H only used for the gear

$K_T = 1$

$K_R = 0.85$

$S_H = \frac{S_c Z_n C_H / (K_T K_R)}{\sigma_c} \approx 2.64$

$H_B = 264\ H_B \Rightarrow S_t \approx 32$ ksi. Fig 14-2

$Y_N = 1.6$ Fig 14-14

Assuming Reliability $= 0.90$

$K_R = 0.85$

Assuming $K_T = 1$ bc Temp $< 100°C$

$SF = \frac{S_t Y_N / (K_T K_R)}{\sigma_{bending}}$

$= 12.88$

$SF_{contact} = \frac{53.107 \text{ ksi}}{53.161 \text{ ksi}} = \boxed{0.999}$

↑ incorrect data mostly from steel.

Planetary gears

$$\sigma_{bending} = W_t K_o K_v K_s \frac{P_d}{F} \frac{K_m K_B}{J} = \boxed{6.190 \text{ ksi}}$$

$$K_s = 1.192 \left(\frac{F\sqrt{Y}}{P}\right)^{0.0535} = 1.192 \left(\frac{.75\sqrt{.290}}{12}\right)^{0.0535} = 0.9942$$

9942

$J = 0.25$ Fig 14-6

$$\sigma_{allow} = \frac{S_t}{S_f} \frac{Y_N}{K_t K_R}$$

Bending planetary

$$SF = \frac{S_t Y_N / K_T K_R}{\sigma_{bending}}$$

$= 6.08$

$= 6.190$ ksi

unreasonable St take from Fig 14.2 which is for steels not plastic gears.

$$SF_{Bending} = \frac{\text{flexural yield}}{\sigma \text{ bending}} = \frac{8.793}{6.190} = \boxed{1.45}$$

$$\sigma_{contact\ allow} = \frac{S_c}{S_H} \frac{Z_n C_H}{K_T K_R} = 53.107 \text{ ksi}$$

same as before

CH assumed to be 1
although not through hardend steel pinion nor gear used

$$\sigma_c = C_p \sqrt{W_t K_o K_v K_s \frac{K_m}{d_p F} \frac{C_f}{I}} = \boxed{67.782 \text{ ksi}}$$

$d_p = 1.25"$

$$SF_c = \frac{53.107 \text{ ksi}}{67.782 \text{ ksi}} = \boxed{0.789}$$

incorrect due to lack of data

Internal gear

$$\sigma_{bending} = W_t K_o K_v K_s \frac{P_d}{F} \frac{K_m K_B}{J} = \boxed{3.890 \text{ ksi}}$$

$$K_s = 1.192 \left(\frac{F \sqrt{Y}}{P} \right)^{0.0535} = 1.00378$$

$Y = 0.415$
Table 14.2

$J \approx 0.40$
Fig 14-6

$$\sigma_{allow\ bending} = \frac{S_t}{S_F} \cdot \frac{Y_N}{K_t K_R} = 3.900 \text{ ksi}$$ ← St value from shigley's for steel not polymers

$$SF = \frac{S_t Y_N / K_T K_R}{\sigma_{bending}} = 9.67$$

$$SF_{bending} = \frac{\text{flexural yield}}{\sigma_{bending}} = \frac{0.773}{3.890} = \boxed{2.26}$$

Cp value from book meant for steel on steels

$$\sigma_{contact} = C_p \sqrt{W_t K_o K_v K_s \frac{K_m}{d_p F} \frac{C_f}{I}} = \boxed{18.62 \text{ ksi}}$$

$$I = \frac{\cos\phi_t \sin\phi_t}{2 m_N} \frac{m_g}{m_g - 1} = \frac{\cos 20 \sin 20}{2(1)} \frac{3.6}{3.6 - 1} = 0.22503$$

$$m_G = \frac{54}{15} = 3.6$$

$$\sigma_{contact\ allow} = \frac{S_c}{S_H} \frac{Z_n C_H}{K_T K_R} = 53.107 \text{ ksi}$$ (same as before)

$$S.F._{contact} = \frac{53.107 \text{ ksi}}{18.62 \text{ ksi}} = \boxed{2.85}$$

Machine Design Typed Calculations

As the scope of the class is more Machine Design than Fluids or Dynamics, relevant sample calculations have been typed for better review.

Machine Design Calculations Pinion (Sun)

Finding Transmitted Load W_t

W_t=transmitted load, lb

r_p=radius of pinion, in

T_p=stall torque at pinion, oz-in

d_p=diameter of pinion, in

Given:

T_p=28.88 oz-in

d_p=2 in

$$r_p = \frac{d_p}{2} = \frac{2\ in}{2} = 1\ in$$

$$T_p = 28.88\ oz - in * \frac{1\ lb}{16\ oz} * \frac{16\ gearbox\ out}{1\ gearbox\ in} = 28.88\ lb - in$$

$T_p = W_t r_p$ rewrite as

$$W_t = \frac{T_p}{r_p} = \frac{28.88}{1} = 28.88\ lb$$

Using Figure 14-17, assuming uniform power source and light shock

$$K_o = 1.25$$

Q_v=quality number/AGMA transmission accuracy-level number—assumed to be 3 for lowest quality commercial gear given in Shigley's textbook

$$A = 50 + 56(1 - B) = 50 + 56(1 - 1.082) = 45.43$$

$$B = 0.25(12 - Q_v)^{2/3} = 0.25(12 - 3)^{2/3} = 1.082$$

$$(V_t)_{max} = [A + (Q_v - 3)]^2 = [45.43 + (3 - 3)]^2 = 2063.477\ ft/min$$

Using $(V_t)_{max}$ for added conservatism

$$K_v = \left(\frac{A + \sqrt{(V_t)_{max}}}{A}\right)^B = \left(\frac{45.43 + \sqrt{2063.477}}{45.43}\right)^{1.082} = 2.538$$

$$K_s = 1.192\left(\frac{F\sqrt{Y}}{P}\right)^{0.0535} = 1.192\left(\frac{0.75\sqrt{0.337}}{12}\right)^{0.0535} = 0.998205449$$

Given F=face width, in=0.75 in (measured) and N_{pinion} =24 teeth

$$P_d = \frac{N_{pinion}}{d_p} = \frac{24}{2} = 12\frac{teeth}{in}$$

$$K_m = 1 + C_{mc}(C_{pf}C_{pm} + C_{ma}C_e)$$

$$C_{mc} = 1$$

Assuming uncrowned teeth

$$C_{pf} = \frac{F}{10d} - 0.025 = \frac{.75}{10(2)} - 0.25 = 0.0125$$

$$C_{pm} = 1$$

Assumed due to lack of data for non straddle − mounted gears

Using Open Gearing Condition

$$A = 0.247 \quad B = 0.0167 \quad C = -0.765(10^{-4})$$

$$C_{ma} = A + BF + CF^2 = 0.02595$$

$$C_e = 1 \text{ Assuming all other conditions}$$

$$K_m = 1 + C_{mc}\left(C_{pf}C_{pm} + C_{ma}C_e\right) = 1.271981969$$

$$K_b = 1 \text{ Due to constant thickness of teeth}$$

$$J \approx 0.33 \text{ Figure } 14.6$$

$$\sigma_b = W_t K_o K_v K_s \frac{P_d}{F}\frac{K_m K_b}{J} = 28.88 * 1.25 * 2.54 * 0.998 \frac{12}{0.75}\frac{1.272 * 1}{0.33} = 5650.5\ psi$$

S_t and S_c will be assumed to be 8.793 ksi which is the flexural yield of ABS given from our material properties. The values of S_t and S_c in Shigley's are for metals gears

S_F will be assumed to be 1 due to added conservatism throughout calculations

$$K_T = 1 \text{ because temperature is less than } 250°\text{F}$$

$$K_R = 0.85 \; assuming\ a\ reliability\ of\ 0.90\ and\ Table\ 14\text{-}10$$

$$Y_N = 4.9404N^{-0.1045} = 4.9404(10^6)^{-0.1045} = 1.166$$

$$Assuming\ a\ life\ of\ N = 10^6\ cycles\ and\ a\ given\ Brinell\ Hardness\ of\ 264\ H_B$$

$$(conservatively\ estimating\ to\ be\ 250\ H_B)$$

$$\sigma_{b\,all} = \frac{S_t}{S_f} * \frac{Y_N}{K_T * K_R} = \frac{8793}{1} * \frac{1.166}{1 * 0.85} = 12063\ psi$$

$$C_p = \left[\frac{1}{2\pi\left(\frac{1-v^2}{E}\right)}\right]^{\frac{1}{2}} = \left[\frac{1}{2\pi\left(\frac{1-0.35^2}{8793}\right)}\right]^{\frac{1}{2}} = 39.935\sqrt{psi}$$

$$C_f = 1$$

$$I = \frac{\cos\emptyset_t \sin\emptyset_t}{2m_n} \frac{m_G}{m_G + 1} = \frac{\cos 20 \sin 20}{2(1)} \frac{0.625}{0.625 + 1} = 0.071645$$

$$Given\ v_{ABS} = 0.35 \quad E = 8.793\ ksi \quad m_N = 1\ for\ spur\ gears \quad m_G = \frac{N_G}{N_P} = \frac{15}{24} = 0.625 \quad \emptyset_t = 20°$$

$$\therefore \sigma_c = C_p \sqrt{W_t K_o K_v K_s \frac{K_m}{d_P F} \frac{C_f}{I}} = 1314\ psi$$

$$\sigma_{c\,all} = \frac{S_c}{S_H} \frac{Z_N C_H}{K_T K_R} = 11768\ psi$$

$$\therefore F.S._{Bending} = \frac{\sigma_{b\,all}}{\sigma_b} = \frac{12063}{5650.5} = 2.13$$

$$\therefore F.S._{Contact} = \frac{\sigma_{c\,all}}{\sigma_c} = \frac{11768}{1314} = 8.96$$

Machine Design Calculations for Idler (Planet) Gears

$$All\ factors\ are\ the\ same\ except\ for\ the\ following$$

$$N_{planetary} = 15\ teeth\ and\ using\ Table\ 14-2\ to\ find\ Y$$

$$Also\ for\ a\ single\ planetary\ gear, W_t\ will\ be\ assumed\ to\ be\ shared\ equally\ by\ the\ 3\ planets$$

$$\therefore W_t = \frac{28.88}{3} = 9.67\ lb$$

$$K_s = 1.192\left(\frac{F\sqrt{Y}}{P}\right)^{0.0535} = 1.192\left(\frac{0.75\sqrt{0.290}}{12}\right)^{0.0535} = 0.9947$$

$$C_{pf} = \frac{F}{10d} - 0.025 = \frac{0.75}{10 * 1.25} - 0.025 = 0.035$$

$$J \approx 0.25\ From\ Figure\ 14-6$$

$$I = \frac{cos\emptyset_t\ sin\emptyset_t}{2m_n}\frac{m_G}{m_G+1} = \frac{cos20\ sin20}{2(1)}\frac{3.6}{3.6+1} = 0.1457$$

$$m_G = \frac{N_G}{N_P} = \frac{54}{15} = 3.6$$

$$\therefore Km = 1 + C_{mc}\left(C_{pf}C_{pm} + C_{ma}C_e\right) = 1.2945$$

$$\therefore \sigma_b = W_tK_oK_vK_s\frac{P_d}{F}\frac{K_mK_b}{J} = 2530\ psi$$

$$\therefore \sigma_{b\,all} = \frac{S_t}{S_f} * \frac{Y_N}{K_T * K_R} = 12063\ psi$$

$$\therefore \sigma_c = C_p\sqrt{W_tK_oK_vK_s\frac{K_m}{d_PF}\frac{C_f}{I}} = 1173\ psi$$

$$\therefore \sigma_{c\,all} = \frac{S_c}{S_H}\frac{Z_NC_H}{K_TK_R} = 11768\ psi$$

$$\therefore F.S._{Bending} = \frac{\sigma_{b\,all}}{\sigma_b} = \frac{12063}{2530} = 4.77$$

$$\therefore F.S._{Contact} = \frac{\sigma_{c\,all}}{\sigma_c} = \frac{11768}{1173} = 10.0$$

Machine Design Calculations for Internal (Ring) Gear

$$All\ factors\ are\ the\ same\ except\ for\ the\ following$$

$$N_{planetary} = 54\ teeth\ and\ using\ Table\ 14-2\ to\ find\ Y$$

$$K_s = 1.192\left(\frac{F\sqrt{Y}}{P}\right)^{0.0535} = 1.192\left(\frac{0.75\sqrt{0.415}}{12}\right)^{0.0535} = 1.00378$$

$$C_{pf} = \frac{F}{10d} - 0.025 = \frac{0.75}{10*4.5} - 0.025 = -0.00833$$

$$\therefore Km = 1 + C_{mc}\left(C_{pf}C_{pm} + C_{ma}C_e\right) = 1.251$$

$$Using\ the\ appropriate\ equation\ for\ internal\ gears\ 14-23$$

$$I = \frac{cos\emptyset_t\ sin\emptyset_t}{2m_n}\frac{m_G}{m_G - 1} = \frac{cos20\ sin20}{2(1)}\frac{3.6}{3.6-1} = 0.2579$$

$$m_G = \frac{N_G}{N_P} = \frac{54}{15} = 3.6$$

$$m_G\ is\ same\ here\ because\ the\ internal\ gear\ is\ still\ being\ driven$$

$$J \approx 0.40\ From\ Figure\ 14-6$$

$$\therefore \sigma_b = W_tK_oK_vK_s\frac{P_d}{F}\frac{K_mK_b}{J} = 4602\ psi$$

$$\therefore \sigma_c = C_p\sqrt{W_tK_oK_vK_s\frac{K_m}{d_PF}\frac{C_f}{I}} = 459.1\ psi$$

$$\therefore F.S._{Bending} = \frac{\sigma_{b\,all}}{\sigma_b} = \frac{12063}{4602} = 2.62$$

$$\therefore F.S._{Contact} = \frac{\sigma_{c\,all}}{\sigma_c} = \frac{11768}{459.1} = 25.6$$

Excel Spreadsheet Iterative Results

Constants due to Material or Geometry		
Vane Shell Length	L (in)	22.875
Vane Twist Angle	theta (deg)	45.000
Vane Length	s1 (in)	32.350
Vane Shell Diameter	D1 (in)	6.300
Outer Shell Diameter	D2 (in)	9.624
Vane Shell Inter-Vane Arc Length	s2 (in)	3.958
Outer Shell Inter-Vane Arc Length	s3 (in)	6.047
Vane Height	h (in)	1.350
Vane Thickness	t (in)	0.250
PVC Specific Weight	gammaPVC	4.69E-02
Water Specific Weight	gammaH2O	62.400
Vane Mass	mvane (lbm)	0.016
Hydrostatic Vane Force	FHS (lbf/vane)	6.868
Inter-Vane Water Mass	mvaneH2O (lbm)	0.213
Numer of Vanes	n (vanes)	5.000
Cylinder Inertia Factor of Safety	FScyl	1.250
Vane Shell Radius	R1 (in)	3.150
Outer Shell Radius	R2 (in)	4.812
Average Radius	Ravg(in)	3.981
Nozle Radius	Rn (in.)	1.562
Water Density	p (slugs/in^3)	1.12E-03
Mass of Device	m (slugs)	1.188
Hydrolic Diameter	Dh (in)	6.237
Surface Roughness	e (ft/ft)	0.050
Kinematic viscostiy	g (in^2/s)	2.03E-03
Vane Area	A (in^2)	38.018
Kv	RPM/V	644.000
Kt	Lb-in/A	0.121
Voltage	Volts	11.100
GearBox Ratio	1:16	0.063
Planitary Ratio	12:27	0.444
Total Rato	1:36	0.028
Speed	RPM	198.567
Battery Capacity	Ah	2.800
Global Factor of Safety	FOS	1.500

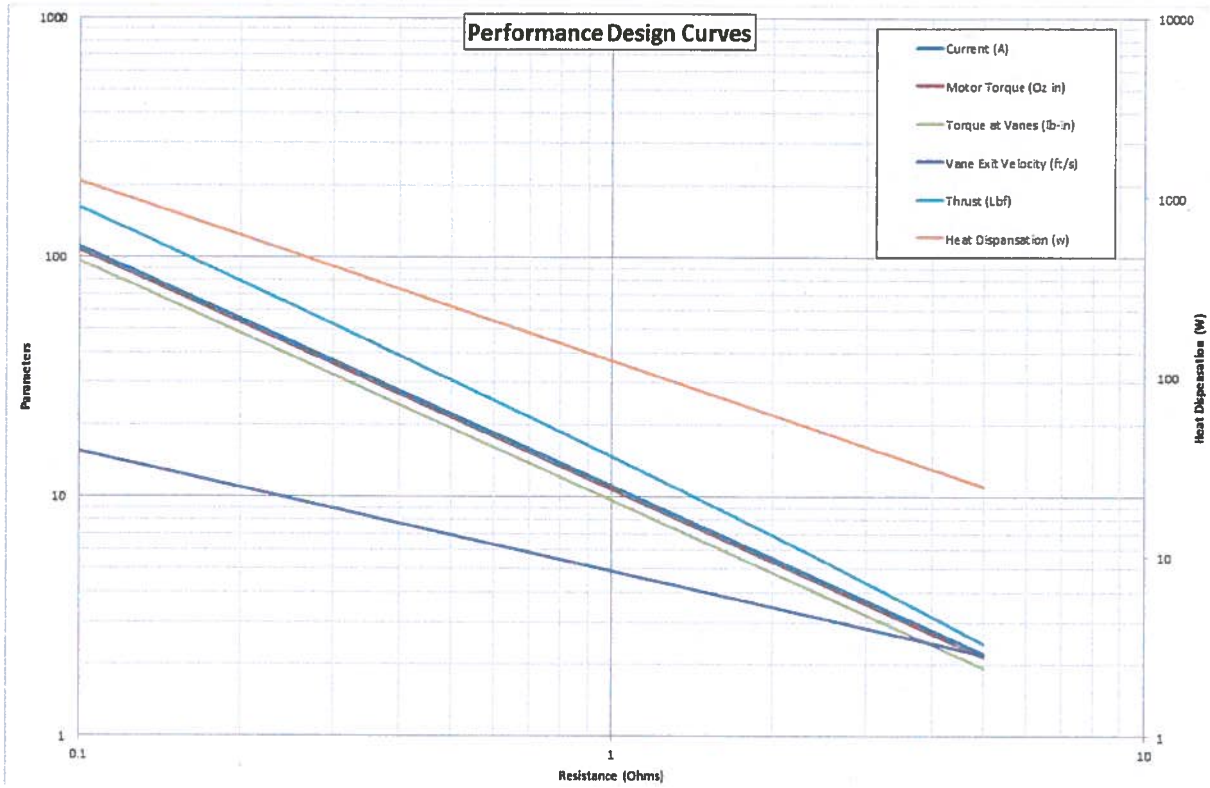
Variables	
Initial Velocity (in/s)	0
Pressure Change (psi)	5
Startup Time (sec)	4

Intermediate Values	
Vane Exit Velocity (in/sec)	80.795
Average Velocity (in/sec)	40.398
Acceleration (ft/s^2)	234.287
f	0.036
Re	124355.37
hL (in)	0.277
Exit Velocity Loss (in/s)	2.566
True Exity Velocity (in/s)	78.230
Drag Torque (lb-in)	43.040
Current (A)	7.395
Angular Acceleration (rev/min2)	2978.500
Start Torque Per Vane (lbft/vane)	2.302

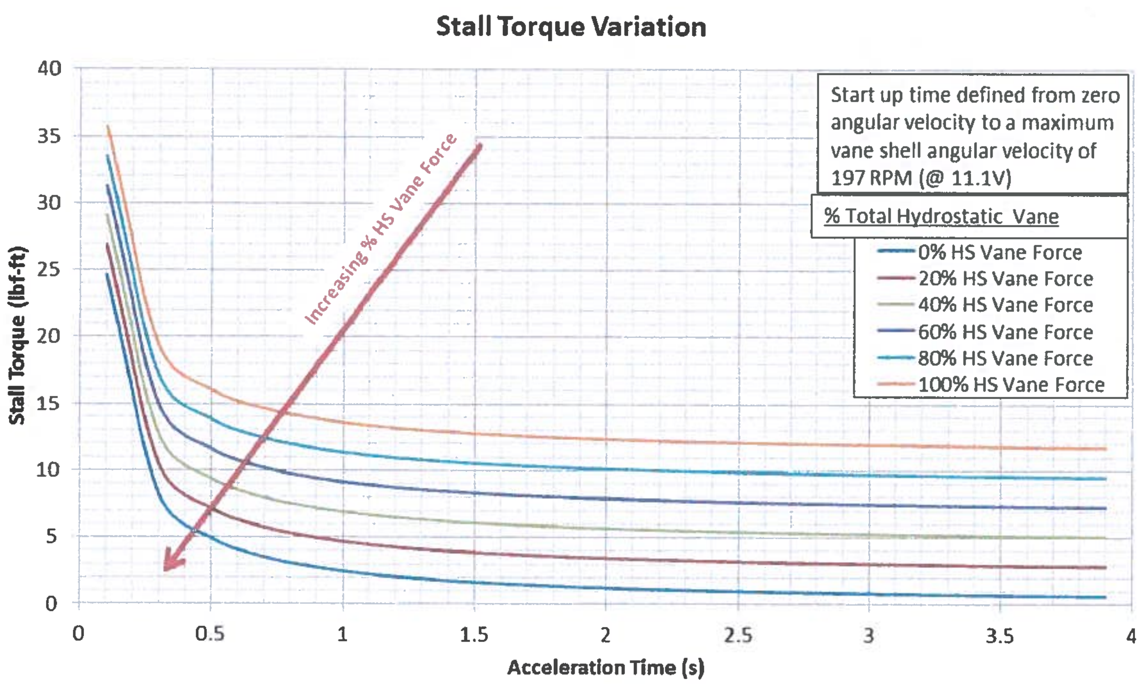
Final Results	
Thrust (lbf)	58.57
Motor Running Torque (oz-in)	14.35
Running Time (min)	22.72
Start Torque (oz-in)	3.84

Note: The table above displays final design numbers as determined through many iterations completed by way of functions within Excel spreadsheets, as programmed from the hand calculations listed above.

Nice



Note: This plot displays the upper operating conditions (worst-case scenarios) as functions of the power supply inputs; in other words, this plot shows the upper limits as can be provided for by the demonstrator (a function of SHEILA's capabilities).



Note: This plot displays the maximum thrust that can be encountered while starting SHEILA just beneath the water surface if the water between all vanes is assumed to act completely normal to the vanes (conservative); in other words, these are the worst-case scenarios that can be encountered in a testing environment (a function of the environment).

Appendix C: Photographs

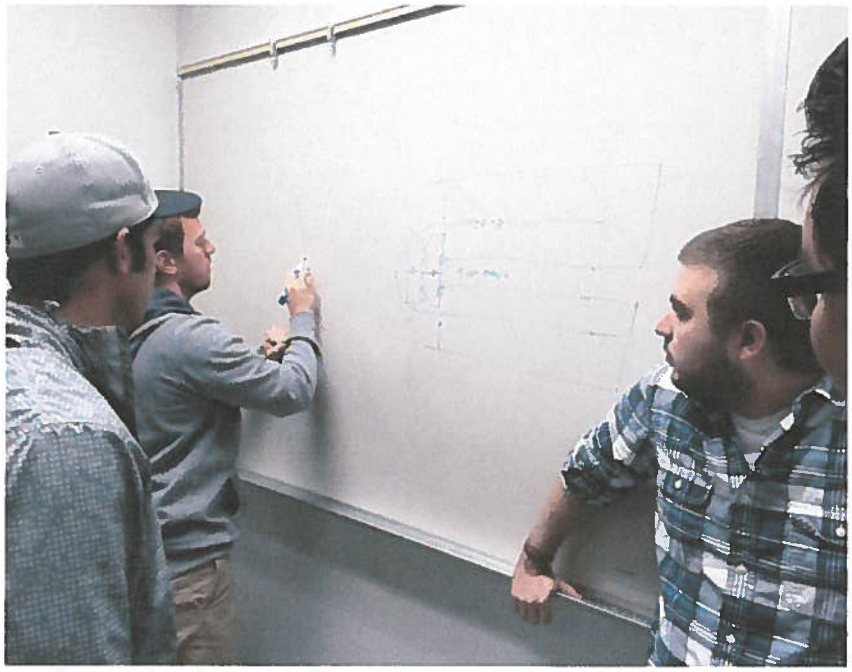


Figure 4: Team UV Discussing Preliminary Designs

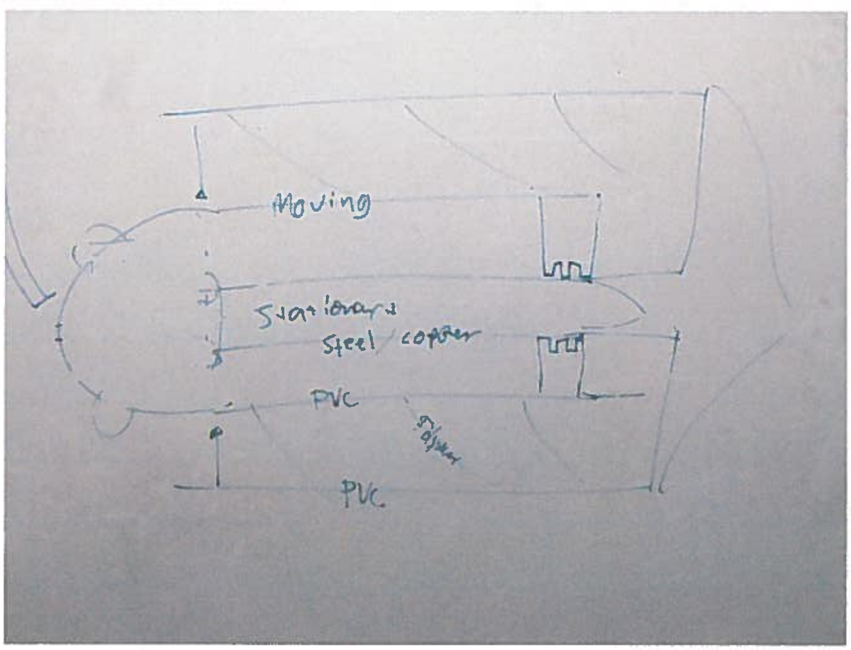


Figure 5: First SHEILA-D Concept

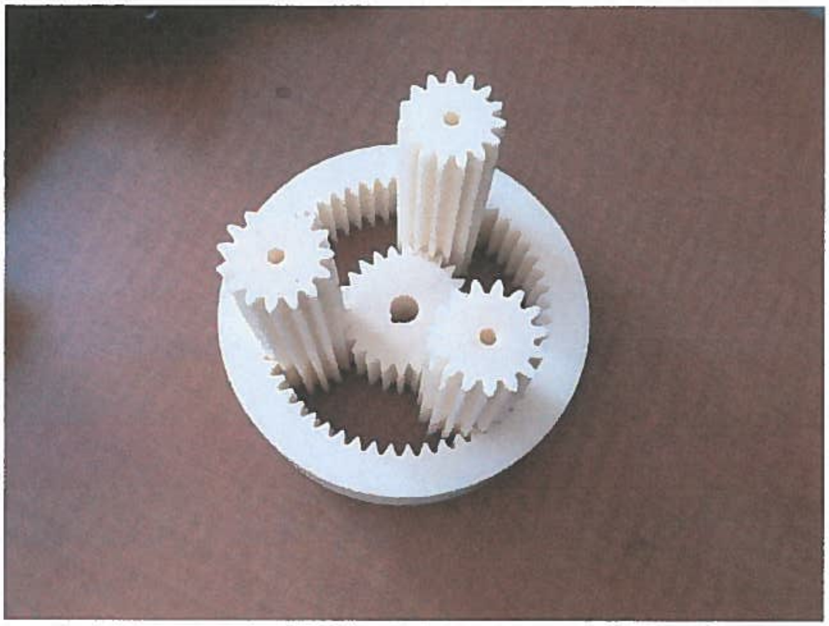


Figure 6: 3D Printed Gearset Extruded and Ready to Cut

Figure 7: Motor Housing and Inner Workings

Figure 8: Front View of SHEILA-D Prior to Installing Camera and Observation Dome

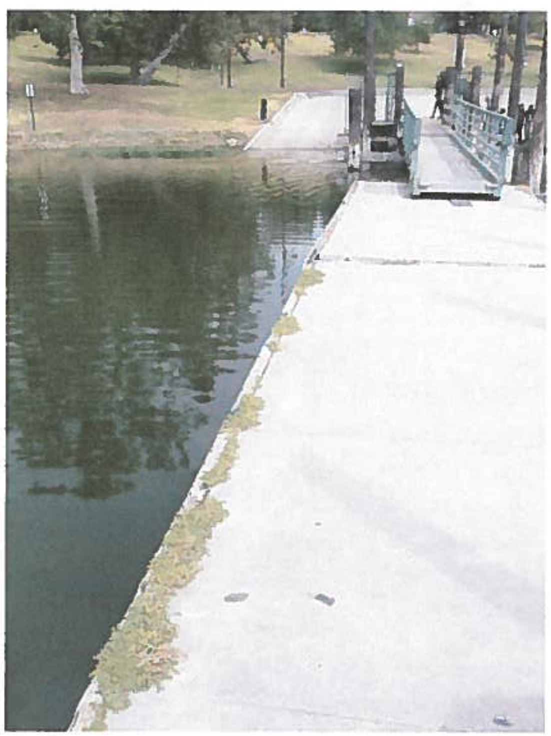


Figure 9: "Sail Boat Cove" Testing Location

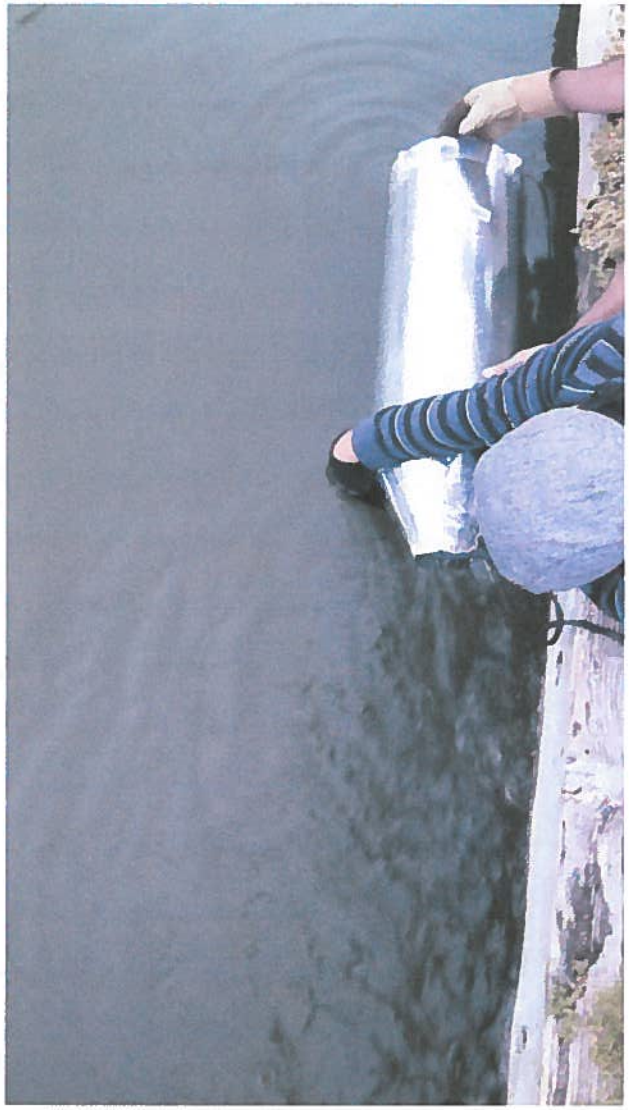


Figure 10: SHEILA-D Operating with Free Jet