University of Southern Indiana
Pott College of Science, Engineering, and Education
Engineering Department
8600 University Boulevard
Evansville, Indiana 47712

The Design of Mechanical Subsystems
for USI's Solar Splash Team
Lily Bittner, Alyssa Dudas, Melissa Kurz
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Approved by: _____

Faculty Advisor: Susan Ely, Ph.D.   Date

Approved by: _____

Department Chair: Paul Kuban, Ph.D.   Date

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Dr. Susan Ely

Dr. Paul Kuban

Justin Amos

Dr. Brandon Field

Jamie Curry

USI Solar Splash Team

Friends and Family

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Abstract

This is a University of Southern Indiana (USI) senior design report for the design of three of the mechanical subsystems for USI's 2022 Solar Splash boat: the solar panel frames, drive train, and trim angle adjuster and propulsion system connection. Solar Splash is a collegiate solar boating competition that takes place annually, and USI intends to participate in the 2023 competition. This report discusses the benefits of design and development of different mechanical systems. It discusses lessons learned from past projects and how best practices found in literature and benchmarking can be leveraged to solve previous deficiencies. It discusses the engineering knowledge required to complete the project and presents conceptual designs with a final design selection. Final critical engineering designs are analyzed and presented for each subsystem. The report establishes the objective, deliverables, schedule, and budget for the project. It also includes a concept of operations, a system hierarchy breakdown, and a failure mode and effects analysis for the competition. The report details how the solar panel frames were designed and how the subsystem was built. It also details how the drive train, propulsion system connection, and trim angle adjuster were designed and how assembly and manufacturing instructions will be provided to the future team for development of those systems. The mechanical subsystems for USI's Solar Splash team will improve the boat and team performance for the 2023 Solar Splash competition.

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1. Introduction

Solar Splash is the World Championship of Collegiate Solar Boating where teams construct a solar powered boat, compete in on-the-water events, write a technical report, and produce a video to earn points to win the competition [1]. Teams compete to test their boat's speed, Endurance, and maneuverability. The teams are evaluated on their performance and technical capabilities. Low scores at competitions in recent years has prompted the USI 2022 Solar Splash team to completely redesign the boat. The full redesign will be a two-year process, and the team plans to compete at the 2023 competition.

The objective of this project is to develop three of the mechanical subsystems for USI's Solar Splash boat: the solar panel frames, drive train, and the trim angle adjuster and propulsion system connection.

The deliverables of the project will include the following:

- Final critical engineering designs for each mechanical subsystem.
- Complete solid models in SolidWorks for each mechanical subsystem.
- Final construction of solar panel frames and fasteners.
- Manufacturing and assembly instructions for the trim angle adjustor, motor mount, shaft housing, and drive train.
- A presentation, poster, and senior design report.

Each senior design team member focused on one of the mechanical subsystems for the boat. Lily Bittner was responsible for the solar panel frames, Melissa Kurz was responsible for the drive train, and Alyssa Dudas was responsible for the trim angle adjustor and propulsion system connection. The propulsion system connection includes the hull mount, motor mount, and shaft housing. The remainder of the boat including, but not limited to, the hull, steering system, and electrical subsystems will be produced by other members of the USI Solar Splash team. A system hierarchy for the boat may be found in Figure 70 in Appendix A. The team plans to use the completed boat in the Solar Splash 2023 competition. The final constructions will be developed directly from the critical engineering designs to allow time for testing before the competition. Future tasks include the design of remaining subsystems, full assembly of the boat, and participation in the 2023 Solar Splash Competition. These future tasks will be turned over to USI's incoming 2023 Solar Splash team. The stakeholders of this project include Solar Splash, Inc., USI, and USI's 2022 Solar Splash team.

In this report, background information will be in Chapter 2. For each respective subsystem, this report will contain a literature review of past projects, what is learned from the literature review, concepts considered prior to critical engineering design, and the critical engineering design. Chapter 3 will review the solar panel frames, Chapter 4 will review the drive train, and Chapter 5 will review the trim angle adjustor and propulsion system connection. Chapter 6 will include the team's management plan, Chapter 7 will review teamwork, and Chapter 8 will conclude the report.

2. Background

Solar Splash is an annual intercollegiate boating competition that began in 1994. The Solar Splash competition gives students an opportunity to learn more about solar energy, energy management, and engineering design. Students get to interact with peers from other universities to learn and further develop their skills. An image of boats competing in the 2019 Solar Splash competition is shown in Figure 1. For the five-day Solar Splash competition, teams race their solar powered boats in Endurance, Sprint, and Slalom events. An image of the race paths can be found in Figure 2 for the Endurance qualifier, the Sprint qualifier, and the Slalom event. Boats must be powered through sunlight only, and wind and human power are not allowed [1]. Additionally, the boats are allowed to have multiple configurations for the different events where some parts may be taken off or interchanged [1]. USI has competed in several past competitions but typically performs in the latter half of placement amongst other teams. USI's 2022 Solar Splash team would like to increase their placement at future competitions and improve the performance of the boat. New designs of the mechanical subsystems will assist in improved performance for USI's boat and will contribute to better placement.

Figure 1: Boats at the 2019 Solar Splash Competition
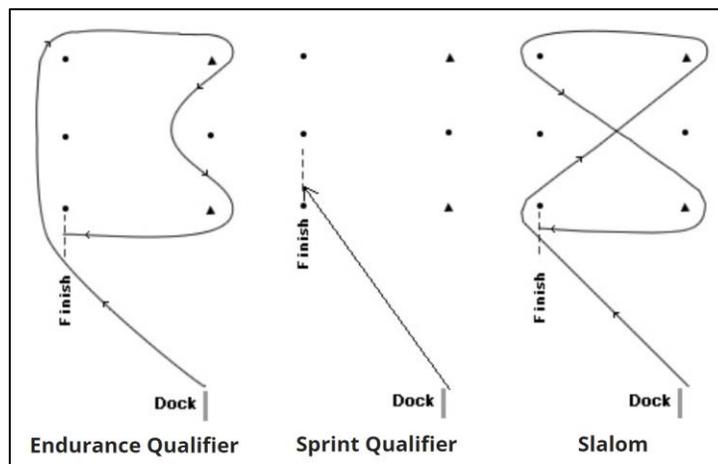
Figure 2: Solar Splash Race Path Diagrams [1]

The Slalom competition tests speed and maneuverability of the boats as they weave through buoys. The Sprint competition is a 300-meter straight course and demonstrates the full power and speed of each team's boat. The Endurance competition tests the stamina and durability of each boat with a 2-hour long race where the team completing the most laps is awarded the most points. Qualifying events test the safety and eligibility of each team's boat. In addition to the races, the technical report and video are judged and awarded a point value. Workmanship of the boat is also judged. Teams can win up to 1000 points [1].

The University of Southern Indiana first competed in Solar Splash in 2015 and has competed in several of the competitions since. Table 1 shows a summary of USI's results in past Solar Splash competitions, including USI's overall rank in the Solar Splash competition, the total number of boats that competed, and USI's total points out of 1000 for each year USI competed. USI has won several awards at Solar Splash, including the 'Outstanding Solar System Design' and 'Outstanding Rookie Team' awards in 2015, the 2nd Best Technical Report award in 2016, and the 'Sportsmanship Award' in 2018. In 2021, the USI team won the 'Perseverance Award' and placed last after receiving only 196.61 points out of 1000 [2]. The low scores in recent years emphasize the need for improvement to the USI solar boat and is the reason USI's 2022 Solar Splash team opted to completely redesign the boat this year.

Table 1: USI's Results in Past Solar Splash Competitions [2]

Year	USI's Overall Rank	Total # of Teams	USI's Total Points (Out of 1000)
2015	9	12	446.77
2016	10	13	510.44
2018	10	12	389.60
2021	9	9	196.61

3. Solar Panel Frames

Solar panel frames support the solar panels during transportation and during competition as they are attached to the boat. The solar panels, including their frames, may be removed during the Sprint and Slalom events of the competition but must be attached to the boat during the Endurance event with a mechanical fastener. For USI's team, solar panel frames and their attachments will be used to structurally support solar panels and provide potential for future configurations.

3.1 Motivation and Need

Solar panel frames are beneficial for adding strength and thickness to a solar panel and they can have a variety of configurations. A basic configuration with maximum allowable dimensions can be found in Figure 3. The solar panels USI plans to use during the 2023 Solar Splash competition are Renogy 175 W 12 V Flexible Monocrystalline Solar Panels [3]. These panels comply with the restrictions of commercially built solar panels used in competition, where they may not exceed an output of 480 Watts under normal sun conditions and their open circuit voltage does not exceed 52 VDC [1]. Additionally, they are IEC 60529 rated with IP67 and IP68 waterproof ratings for the junction box and solar connections. This will ensure that if the solar panels were to fall off into the water during competition they are sealed and protected against the accidental water contact. Since a framed panel can better withstand handling, transportation from place to place, and heavy wind conditions from storms or other impacts, solar panel frames are beneficial for the structural integrity of the solar panels [4]. Leading up to and during the Solar Splash competition, solar panels are often moved around for building, testing, and charging. Added strength can prevent damage to the solar panels that could decrease performance, and by extension, decrease overall team performance at the competition. Additionally, the solar panels must attach to the boat during the Endurance event [1]. USI's team plans to have multiple configurations, one where the solar panels are taken off the boat for the Sprint and Slalom events and another where they are on the boat for the Endurance event. This means the solar panels and frames are taken on and off the boat at least four times during the competition. Added strength and thickness prevents damage to the panels from wind and handling, helping mitigate problems that could arise with the panels. When not attached to the boat the solar panels charge on a table beside the team's tent.
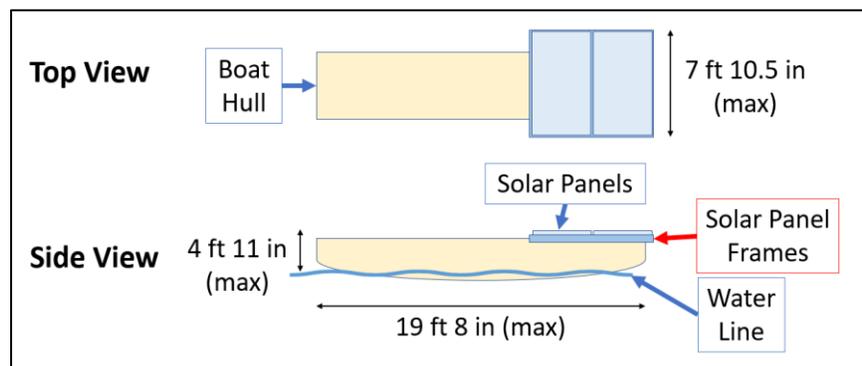
Figure 3: Flat Panel Layout Configuration

Solar panel frames are an important component of USI's Solar Splash boat that can improve the accessibility and functionality of other systems during the competition. USI's 2021 Solar Splash team designed solar panel frames that were attached to the hull of the boat with fixed bolted brackets [5]. Images from the 2021 boat may be found in Figure 4 and Figure 5 [6]. The frames were securely attached to the hull of the boat, however they limited access to the electrical components of the boat by being difficult to remove. The solar panels and solar panel frames had to be detached before troubleshooting problems with the electrical system, and the time and effort taken to do so could have been better spent solving electrical issues. Designing and building frames that are easily removable and allow access to other components of the boat would help improve other systems of the boat and could help improve team performance at competitions.

Figure 4: Mounting Bracket Used on USI's 2021 Solar Splash Boat [6]

Figure 5: USI's 2021 Solar Splash Boat on the Water at Competition [6]

3.2 Constraints and Engineering Design Choices

There are many constraints that drive the design of the solar panel frames and their attachments. The technical requirements for the solar panel frames are shown in Table 2 where the frames shall meet all listed requirements to comply with the official rules of Solar Splash [1].

Table 2: Solar Panel Frame Technical Requirements

Solar Panel Frame Technical Requirements	
Solar Splash Requirements	Be within 19.9 ft x 7.87 ft x 4.92 ft area.
	Be made of water safe, non-polluting material.
	Provide the skipper with an unobstructed view straight forward and 100° to either side.
	Be attached to the hull with a mechanical fastener.
	Be removable in a timeframe of one hour or less.
Additional Requirements	Weigh less than 50 lbs.
	Be removable for the Slalom and Sprint events of the competition.

In addition to these technical requirements, the frames must have a high strength to weight ratio, not flex when in a static position, not deflect excessively when in motion as they are attached to the boat, and not create excessive drag for the boat. The static equations of equilibrium may be used for static calculations [7] and external drag equations may be used to calculate drag produced by the solar panels and frames [8]. The solar panels themselves will be bolted to the frames using the reinforced mounting holes pre-manufactured on the solar panels. The reinforced mounting holes are shown in Figure 6. The frames will be designed to allow the panels to bolt securely at the mounting hole positions and the solar panels will remain bolted to the frames throughout the duration of the competition to maintain structural integrity.

IP68
Rated waterproof
junction box

Six Metal Reinforced
Mounting Holes

Figure 6: Renogy Solar Panels [3]

Another engineering constraint is that the solar panels and frames must attach with a mechanical fastener that can easily detach the frames from the hull of the boat. This engineering

problem is specified in the rules of the Solar Splash competition [1], and a design parameter determined by the team requires making the electrical components easily accessible while the frame is attached to the boat. This mechanical faster must securely attach the frames to the hull and must not break throughout the duration of the competition.

Material selection is an important design choice because selecting a high strength to weight ratio material is necessary to both reduce wight and structurally support the panels. The dimensions of the selected solar panels are 59.2 in long by 26.5 in wide by 0.08 in thick [3], so the dimensions of the frames will be a similar length and width. The structural design must reduce material where possible and be easy to move and handle. The materials must also be attainable for the current budget and schedule. Finally, the location of the panel must be optimized because conditions including temperature, position, and distance from water [9] play a role in solar panel performance [10]. While design is taking place, decisions will have to be made with both engineering and non-engineering constraints in mind.

3.3 Literature Review

The following details a review of literature used to develop the solar panel frames. It includes research on past projects, engineering knowledge, and conclusions from the research.

3.3.1 Past Projects

In past Solar Splash competitions, teams have developed different ways to support and attach their solar panels to their boat, whether that be complex frames or none at all. In one case, Stevens Institute of Technology connected their solar panels to their boat via connectors and grommets shown in Figure 7 [11]. While their design complied with Solar Splash rules, it was not stable or efficient. The design was simple, cheap, and lightweight, but provided no additional structure to prevent solar panel damage and the fastening method used was not easily removeable or adjustable.
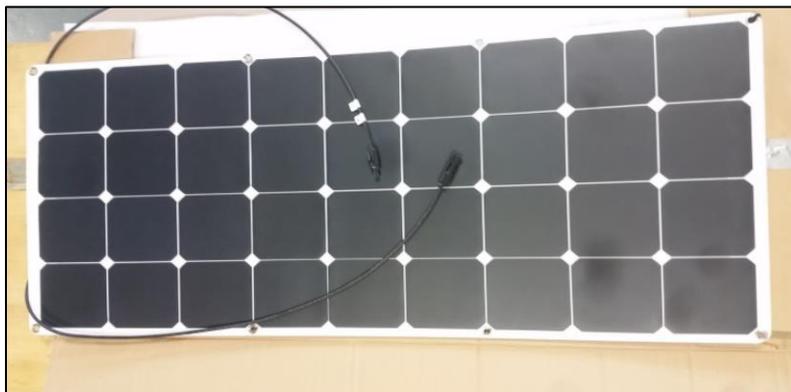
Figure 7: Stevens Institute of Technology – KingSolar Panel with MC4 Connectors and Grommets [11]

USI's 2021 Solar Splash team had a final design that can be seen in Figure 5. It was structurally efficient and cost effective, however its mount design to the boat, shown in Figure 4, caused issues with easy access to storage on the boat because it was not as easily removable as desired [5]. An alternative design is necessary for the frames to be both easily removable and allow quicker access to storage on the boat. Similar materials may be used for USI's 2022 solar panel frames to maintain a lightweight and cost-effective design, but the mechanical fasteners will need an alternative design to improve the overall efficiency of operating the boat.

Another Solar Splash Team, University of Puerto Rico at Mayaguez, designed and 3D printed their own carbon fiber filament frames and connectors [12], shown in Figure 8 and Figure 9. Focus was placed on making solar panel frames that were lightweight, corrosion resistant, and rigid to operate in a boating environment [12]. Their design was advanced, the strength to weight ratio of the material was high, and ultimately, they won the 2021 Solar Splash competition [6]. However, the cost of the frames was high due to the customized material, and the customization and manufacturing of the frame took a longer time than the USI Solar Splash team has allotted for the solar panel frames. A cheaper solution with a similar effectiveness can be designed for USI's team.
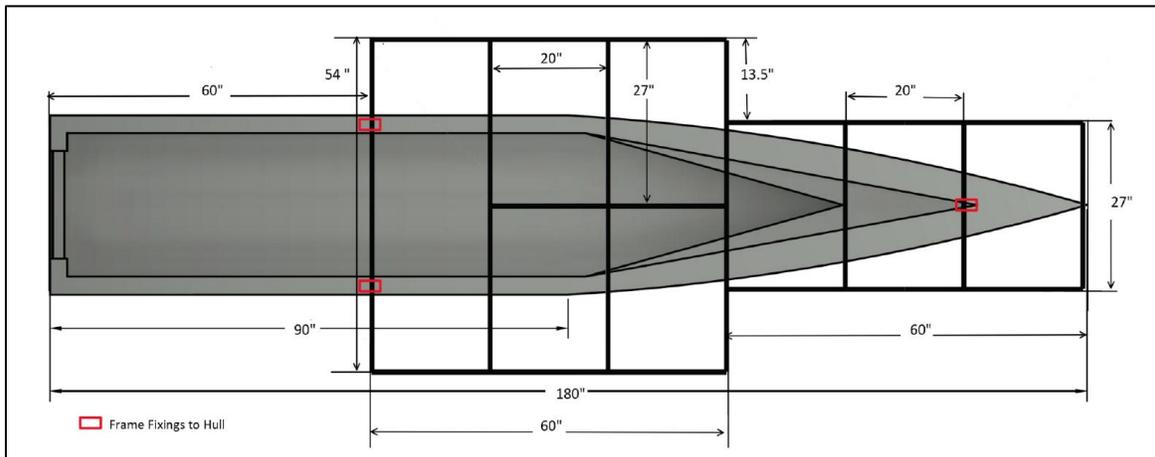
Figure 8: University of Puerto Rico at Mayaguez – Solar Panel Frame Layout on Hull [12]

Figure 9: University of Puerto Rico at Mayaguez – 3D Printed Unions and Assembly [12]

Looking beyond the frames, there are solutions for the attachments as well. For solving the problem of a mechanical fastener that can attach and detach the structural frames from the hull of

a boat easily, quick release mechanisms were suggested as a way to smoothly move the panels and frames on and off the hull. One quick release mount made by Ergotron can be found in Figure 10. These mounts allow for quick and easy attachment and removal, are sturdy, and are simple to install [13]. However, the design is bulky and not flexible in terms of changing the orientation.

Figure 10: Ergotron Quick Release LCD Bracket [13]

Another type of quick release is an awning mount made by RacksBrax. Suitable for use with solar panels these mounts are securely pinned, easy to mount, and are very durable [14]. These mounts are very stable and are tamper and theft resistant. However, they are expensive and would not be effective as a cheap and flexible solution for the USI Solar Splash Team.

Figure 11: RacksBrax Quick Release Awning Mount [14]

A simpler alternative to these options is a using a mechanical release pin. Many common types of release pins may be purchased from standard hardware suppliers. Shown in Figure 12 is a ring grip clevis pin and shown in Figure 13 is a loop grip quick release pin available from McMaster-Carr [15]. Each of these pins is easy to push in and out of hole to secure framing. They are cheap options that are flexible and can be adapted to different types of designs and

configurations. They are standard sizes and provide strong and secure attachment. However, with these options additional mount devices would need to be purchased to connect the frames to a point of contact on the hull.
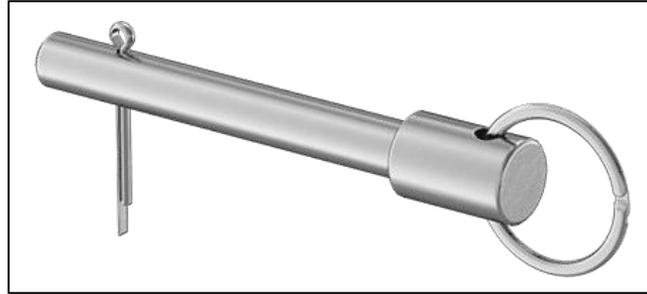
Figure 12: Ring-Grip Clevis Pin [15]
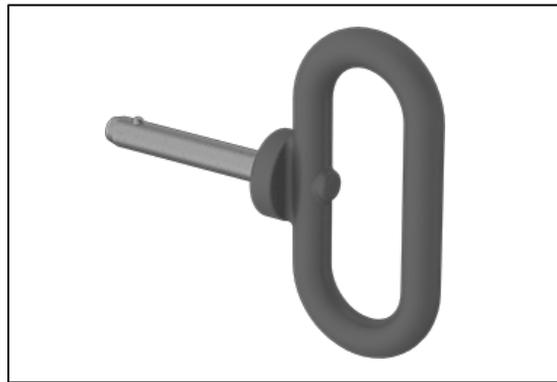
Figure 13: Loop-Grip Quick-Release Pin [15]

3.3.2 Engineering Knowledge

Engineering knowledge will be used throughout the process of the project. Subjects containing knowledge for the project include statics, dynamics, materials science, strength of materials, and fluid mechanics. Found in the literature, statics and dynamics may be used to analyze the stationary and moving load of the solar panels on the boat using equations of equilibrium [7], and a basic configuration showing how the solar panel, frames, and hull may interact can be found in Figure 3. Static and dynamics may also be used to determine the reaction forces that will be experienced by the hull of the boat [7]. Materials science and strengths of materials knowledge can help assess the viability of materials to be used. Yield strength, density, and other properties will be important in calculations [16]. Finally, fluid dynamics will be used to determine external drag on the frames [8]. During operations the solar panels cannot have large spikes in temperature or pressure, so the conditions of operation will be analyzed [10]. Additionally, the simulation and modeling of the frames, such as types of SolidWorks simulations may be used to analyze stresses and forces acting on the frames. A closer look at pressure and velocity distribution simulations can provide information on how the solar panel, frames, and boat may act during the competition. Figure 14 shows how forces can be applied to the frames in

different directions and Figure 15 shows how pressure and velocity may change along the length of solar panels during use on a moving vehicle. It will be important to ensure the flow during movement is not causing excessive pressures and forces on the panels.
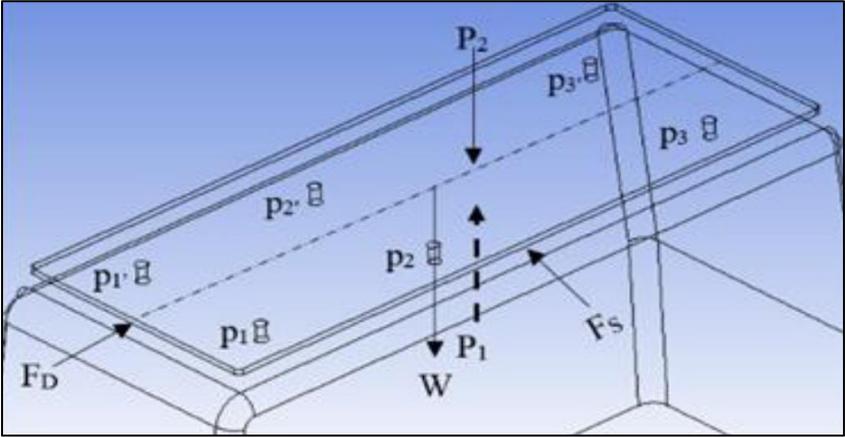
Figure 14: Six Pillars Supporting Flat Solar Panel on Top of Mini Bus [17]
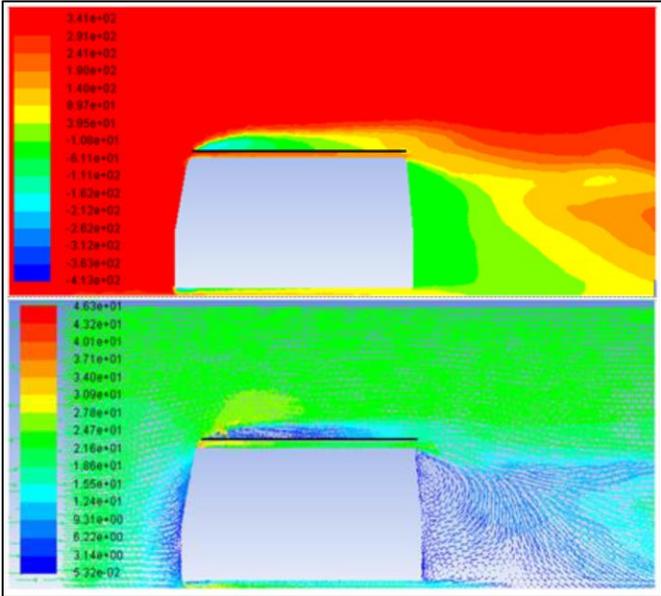
Figure 15: Pressure and Velocity Distribution Simulation for Mini Bus Model with Flat Solar Panel [17]

3.3.3 Conclusions

Parameters for this project are developed from the literature review. From technical reports of past Solar Splash teams, a lesson learned is that critical engineering design on the solar panel frames can contribute to a winning team. Without structural support the solar panels can be damaged. Excessive damage can cause solar panel failure and could cause the boat to fail during competition, resulting in a loss of points and possibly the competition. Additionally, designing a

method of fastening for the frames to the boat can make troubleshooting subsystems easier and more efficient. Ensuring that the design of the fasteners supports how the frames are utilized is also important to not create undue complexity or excessive expense. Possible future configurations of the solar panels as well as the potential purchase of new solar panels should be considered when purchasing fasteners. Ideally, the mounts will be reused and adapted for more than one year of competition. Finally, calculating forces, comparing material properties, and simulating the flow that the solar panels will experience is useful for determining safe operating conditions. Ultimately, iterating through design options can improve final results, and research done for the solar panel frames may be used to make informed decisions during the critical engineering design.

3.4 Conceptual Designs

Multiple position configurations and attachment concepts were reviewed before selecting a design for critical engineering analysis. The solar panels will only be attached during the Endurance race where the boat will be moving at a slower speed and with less sharp turns compared to the Sprint and Slalom events. The potential concepts will be discussed in this section.

When determining possible design concepts, only flat panel configurations were considered. Raised panels increase drag on the boat which would decrease speed decrease speed, which is disadvantageous and thereby not considered. The first configuration shown in Figure 16 places the panel out of the way of main hull space where it would provide unobstructed access to storage on the boat. However, the layout of this configuration places the panels very close to the water.
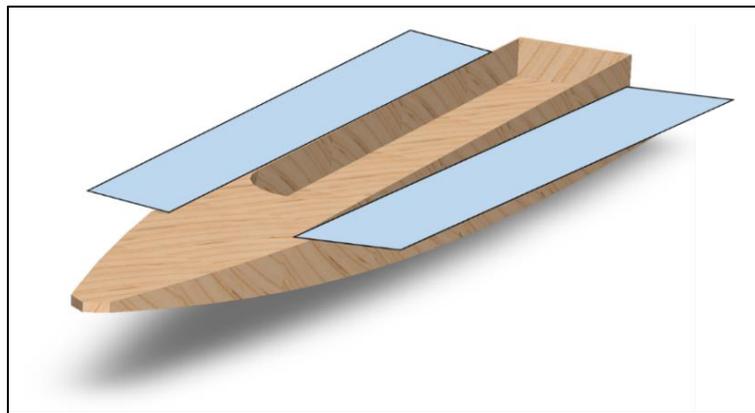
Figure 16: Position Configuration 1

The second configuration is shown in Figure 17. This concept provides a position that is advantageous for the electrical system. The electrical team, which may decide to connect the panels in a series circuit, will have to decide what circuit best aligns with the positioning of the solar panels. The closeness of the panels would make the connection easy with minimal wiring. It also provides mostly unobstructed access to storage on the boat. However, with the two frames being located right next to each other, there could be interference at the edges of the frames and with attachments and the frames are less supported by the hull which tapers at the front.
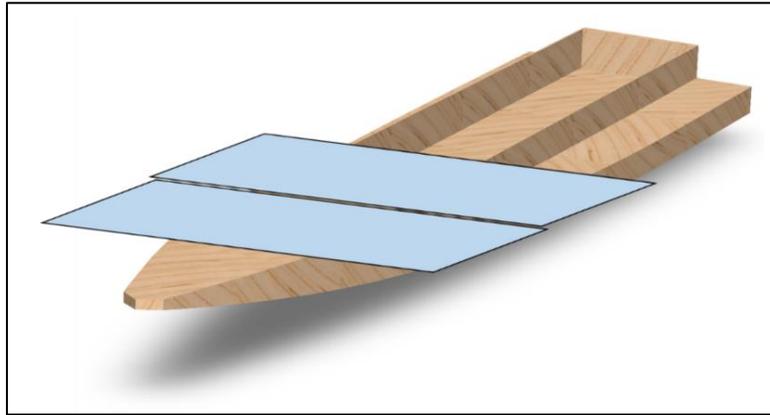
Figure 17: Position Configuration 2

The third configuration is shown in Figure 18. This concept provides relatively even weight distribution and is fully supported by the hull. It is the most structurally sound option because there is support away from the waterline and the weight of the panels would be evenly distributed along the hull. However, it limits additional storage space and would require longer wiring for the panel.
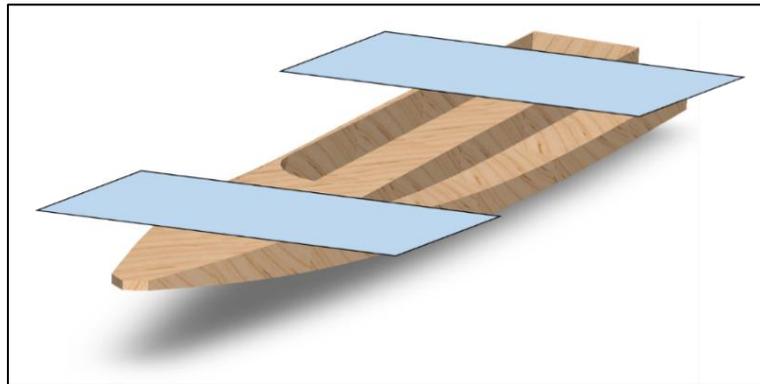
Figure 18: Position Configuration 3

After reviewing each of these configurations, the chosen position was Configuration 3. The solid structural design of this positioning would be the most beneficial for USI's Solar Splash team. Making the panels easy to remove would mitigate problems with limited access to boat storage and enough wiring will be purchased to make adjustments for a lengthened electrical connection between the panels.

Regarding frame attachment, one concept for mechanical fastening to the hull is shown in Figure 19. A hinge configuration would allow for the panel to remain on the boat while making adjustments to the other subsystems. The team elected to opt out of a hinge configuration because quick release mechanisms would provide quick complete detachment from the hull that

wouldn't require unnecessary hinging that may increase stress on the fasteners and panels themselves.
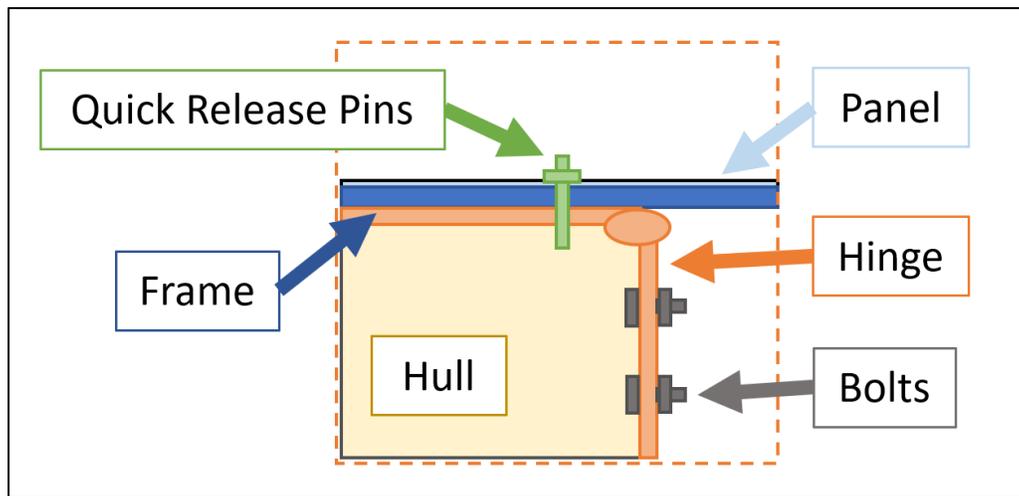
Figure 19: Hinge Attachment Concept

A second concept for attachment is shown in Figure 20. This configuration uses rectangular tubing as rails in order to attach the frame to the hull without causing excessive amounts of stress on the fiberglass that will be used for the hull. It also provides flexibility and potential for future configurations of solar panels. If the positioning of the solar panels is changed or if new solar panels are purchased in the future, additional holes may be easily drilled in the rectangular tubing without compromising the strength of the solar panel frames. However, this design would increase weight of the system.
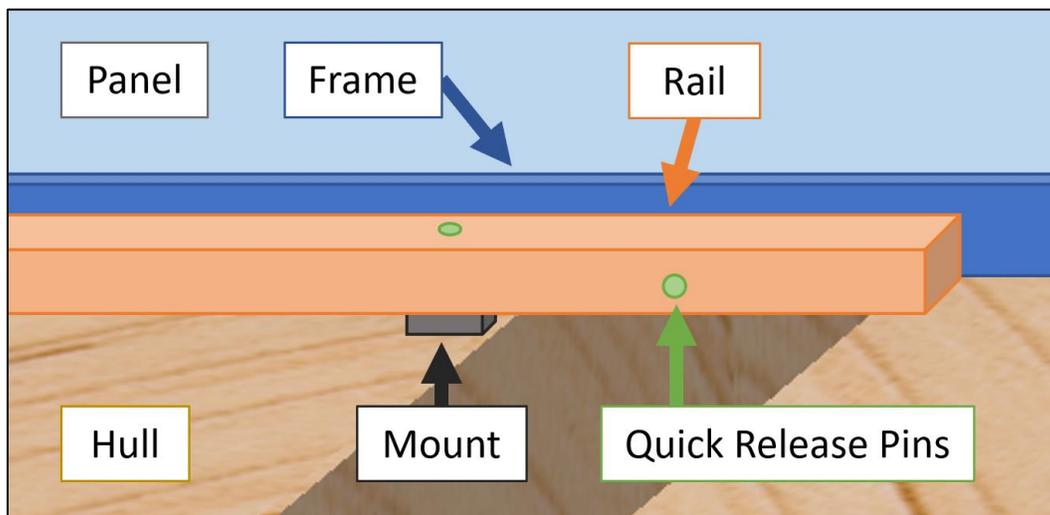
Figure 20: Rail Attachment Concept

A third concept for attachment is shown in Figure 21. This configuration uses posts attached to the frame and the hull. It is lightweight and would take up minimal space. However, it would not provide as much potential for future configurations and would not be as stable.
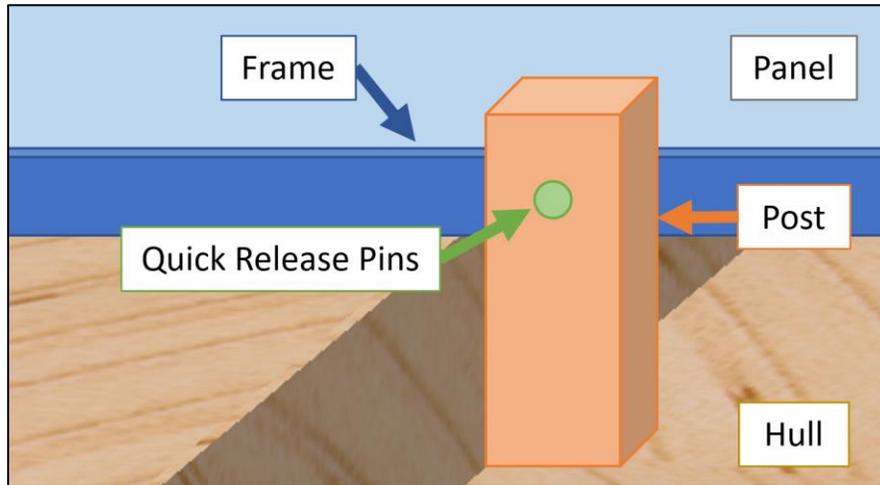
Figure 21: Post Attachment Concept

After reviewing each of these concepts, the chosen attachment was the rail attachment concept shown in Figure 20. The potential for future use would be a benefit to the team and the selection of lightweight flexible solar panel frames counters the weight added by the rails.

3.5 Engineering Design

For the engineering design, the framing for the solar panel frames is analyzed. The frames will sit underneath the solar panels, and an image of where the solar panels will be located on the hull is shown in Figure 22. The frames are 59.5 in long by 26.75 in wide by 13.25 in above the waterline and the length of the boat is 12.5 ft.
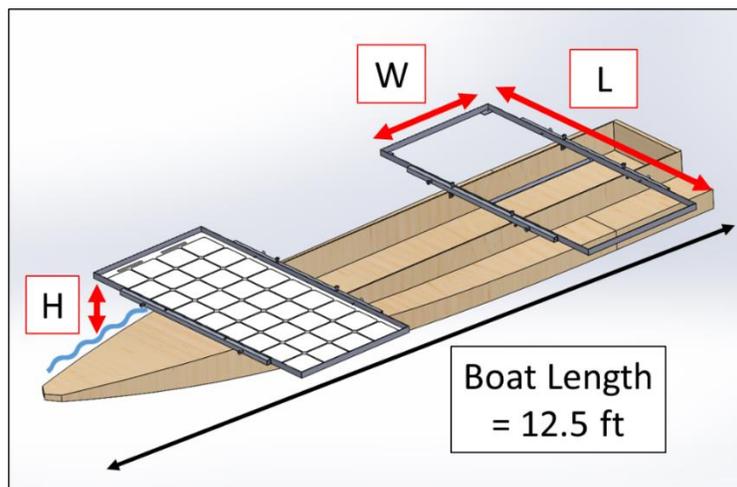
Figure 22: Model of USI's Solar Splash Boat and Solar Panels

3.5.1 Engineering Calculations

Static force calculations on the frames were performed in SkyCiv and the results may be found in Appendix B [18]. The values showed low forces acting on the frames in static position and the values verify acceptable amounts of force on the frame. Drag coefficient calculations were also performed and can be in found in Appendix B. It was confirmed in the calculations that flat panels have a drag coefficient that is nearly negligible. This verifies that the flat panel configuration limits the drag the boat will experience.

3.5.2 Material Selection

Five materials were compared for the solar panel framing. The materials chosen for comparison were Marine Grade Plywood, 3003 Aluminum (Al), 5052 Al, 316 Stainless Steel and 6061 Al. The materials were compared based on four primary properties, density, modulus of elasticity, yield strength, and cost. Density is compared because weight reduction is important to keeping the solar panel frames lightweight and easy to move. Modulus of elasticity is a factor because the material should be manufacturable yet maintain a stiffness that can structurally support the solar panels. Yield strength is also important for the structural integrity of the frames and determining a factor of safety. Cost should be as low as possible and is also compared.

Material properties were found using Matweb's online site [19]. The properties are summarized in Table 3. The green color designates "good", the yellow color designates "acceptable", and the red color designates "bad". In addition to the tabled properties, Marine Grade Plywood was eliminated as an option because the joining method would be difficult to manufacture and ultimately less reasonable. Stainless steel was determined to be too heavy and provided an unnecessary amount of strength. All types of aluminum compared had acceptable properties, but the material with the best combination of properties was 5052 Al. While it was the most desirable material, 6061 Al and was selected as the final choice because it was the easiest to obtain. The aluminum bars used in the project were readily available and purchased from West Metals in Evansville. It is compliant with the Solar Splash rule of being made of water-safe, non-polluting material and is an environmentally safe option.

Table 3: Material Property Comparison

	Material (Type)	Density (g/cc)	Modulus of Elasticity (GPa)	Yield Strength (MPa)	Cost ($/kg)
☒	Marine Grade Plywood	0.59	94.5	339.2	4.40
☒	3003 Aluminum	2.73	68.9	41.4	1.32
☑	5052 Aluminum	2.68	70.3	89.6	0.92
☒	316 Stainless Steel	8.00	193.0	240.0	1.96
☒	6061 Aluminum	2.7	68.9	241	1.25

3.5.3 Engineering Model

Using the 6061 Aluminum, an engineering model of the framing was designed and is shown in Figure 23. The pre-manufactured bolt holes shown in Figure 6 match the locations and dimensions of the holes on the frames. The overall weight of one frame is 3.14 lbs. and only two will be used on USI's boat.
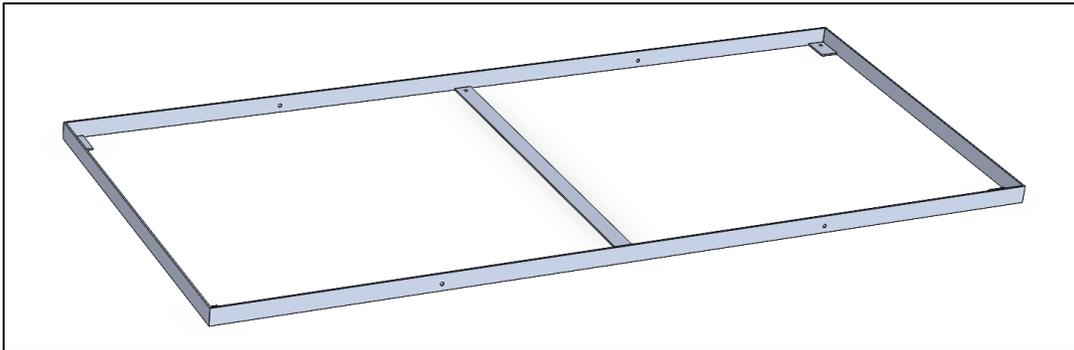
Figure 23: Model of Solar Panel Frame

A view of the attachments may be found in Figure 24. Recessed quick release locking pins are used to provide easy attachment and removal but also prevent accidental displacement during handling and transportation during competition. The mounts and bolts remain in position on the hull for the duration of the competition to avoid disruptions to the fiberglass used for the hull. A bill of materials for cost and weight budget for all of the materials used may be found in Appendix A in Table 11 and Table **12**, respectively.
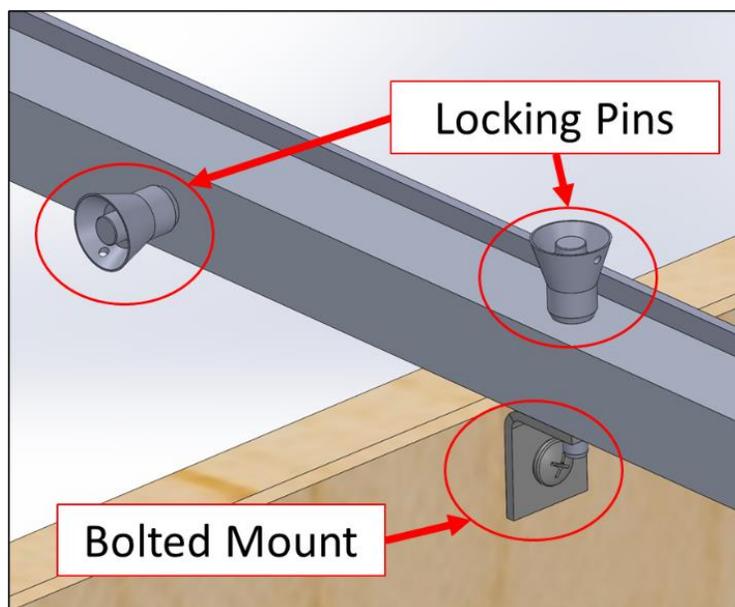
Figure 24: Model of the Solar Panel Frames on the Boat

3.5.4 Engineering Simulations

Once the dimensions and materials were determined, simulations were performed for stress, deflection, and flow analysis. Figure 25 shows where the force was applied to the panels to simulate the weight of 6.2 lbs. per solar panel.

Simulation of stress and displacement was completed in a static simulation in SolidWorks. Figure 25 also show an image of the fixed positions and location of maximum deflection where the maximum experienced was 0.099 in. The maximum stress in the simulation was 1.64 ksi, with a minimum factor of safety of 4.87. Because stress and displacement are so low, the design is suitable for the application.
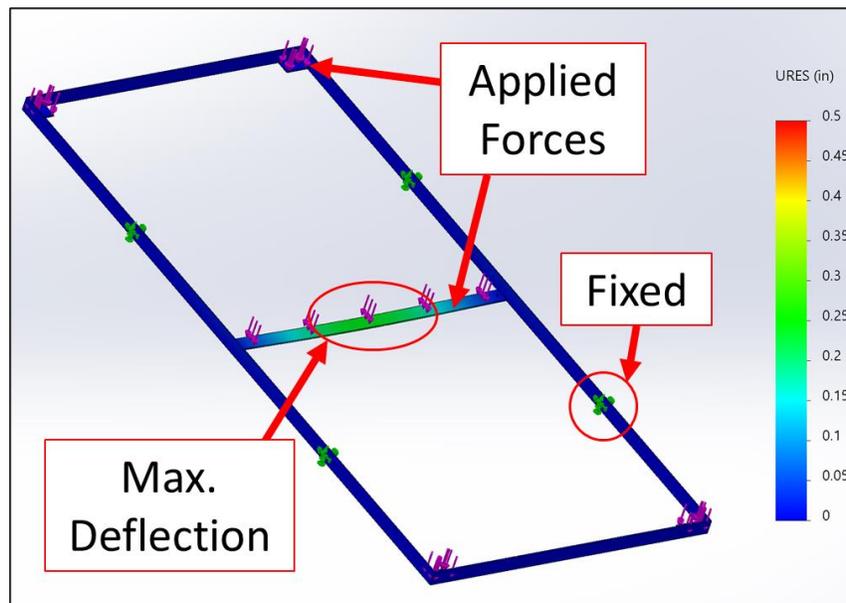
Figure 25: FEA Analysis Using 6061 Aluminum

After the static simulation a flow simulation in SolidWorks was performed. The flow simulation had a few relevant parameters. The water level for the boat has not yet been determined and finalized, and because the solar panels and solar panel frames should be above water during boat operation, an air only simulation was performed. The panels, frames, and hull were simulated as solid bodies experiencing an air velocity of 15 mph. This is the maximum velocity the boat will be designed to experience.

From the simulation, shown in Figure 26 and Figure 27, drag was determined to be negligible with the solar panels and frames acting as parallel flat plates. The maximum pressure experienced was found to be 14.65 psi. A minor pressure spike of less than 1 psi was found at the front of the solar panels and frames. Additionally, there was a rise temperature of less than 1.8 degrees Fahrenheit across the solar panel and frames. This simulation showed that at the current properties and speed, solar panel performance should not decrease in the conditions given.
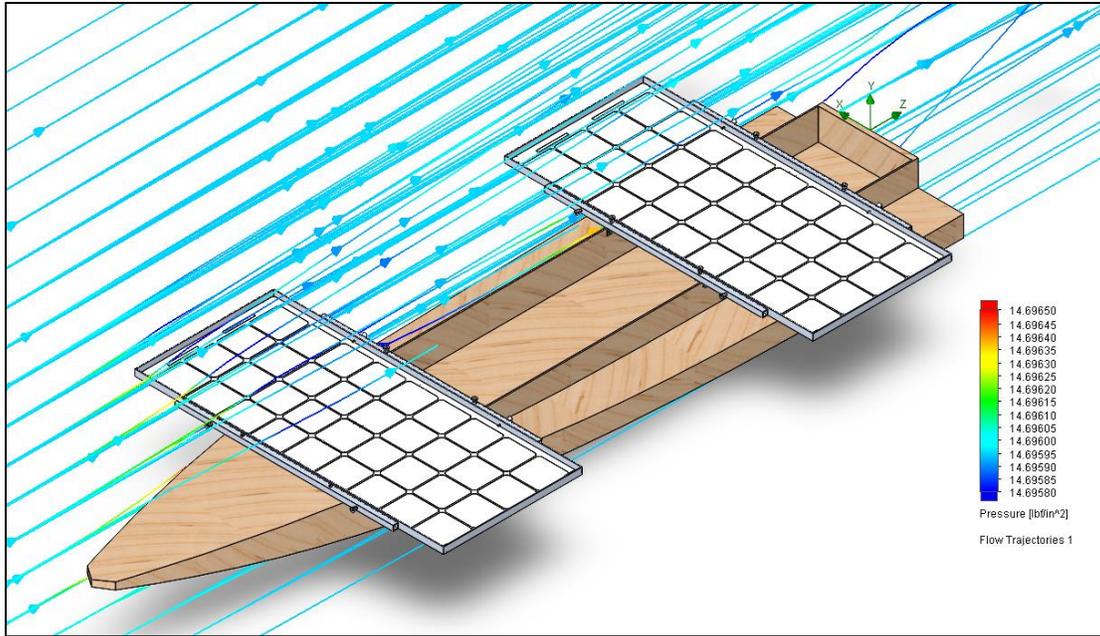
Figure 26: Flow Analysis Isometric View
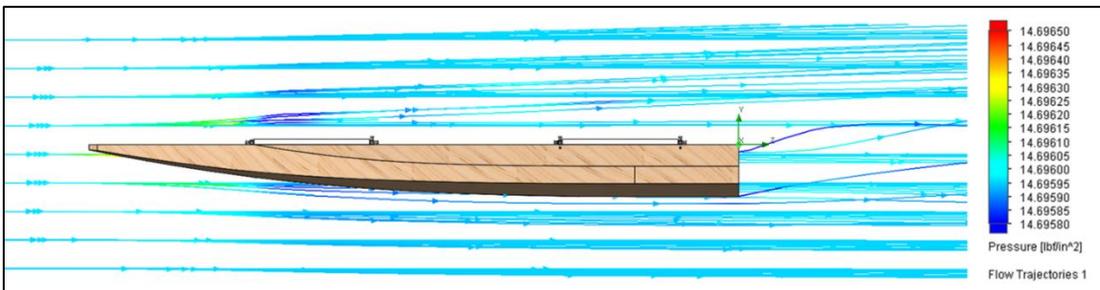
Figure 27: Flow Analysis Side View

3.6 Final Construction

The final frames were constructed in the Applied Engineering Center (AEC) at USI. The 6061 Aluminum bars shown in Figure 28 were cut to size before being welded in place. The aluminum was tig welded with the help of Justin Amos.

Figure 28: 6061 Aluminum Bars

An image of a fully welded frame can be found in Figure 29. It is lightweight and can structurally support a solar panel. Holes were drilled into the frame with a power hand drill. The holes are located where the panels will bolt on and where the rails will be attached with quick release pins. The panels bolted to the frames using M5 nuts, bolts, and washers is shown in Figure 30.

Figure 29: Cut and Welded Frame
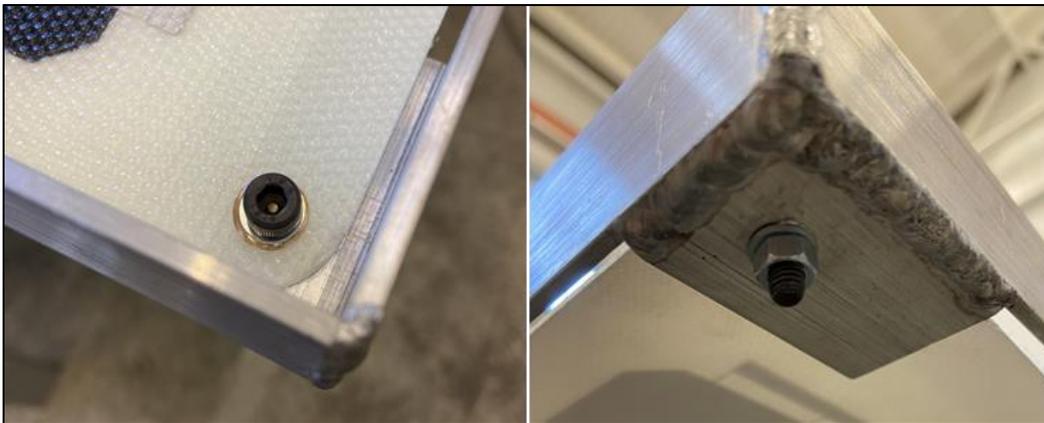
Figure 30: Panel Bolting

The final construction of the frames is shown in Figure 31. Here the rectangular tubing rails have been drilled through where the quick release pins attach the frame to the rails. The holes that will be pinned to the hull mounts will be drilled once the location of the panels has been finalized and confirmed by next year's team. The rails are currently bolted to the frame for

ease of handling and to keep the main pieces together. The panels may remain bolted to the frames throughout the duration of the competition and the system itself may be removed for the Slalom and Sprint events. Additional holes may be drilled in the rectangular tubing and on the mounts if the position of the solar panels changes or if new panels are bought. If the team decides to retire the frames from use, the material may be cut down and reused for other systems or may be used to manufacture panel stands for use at future competitions.
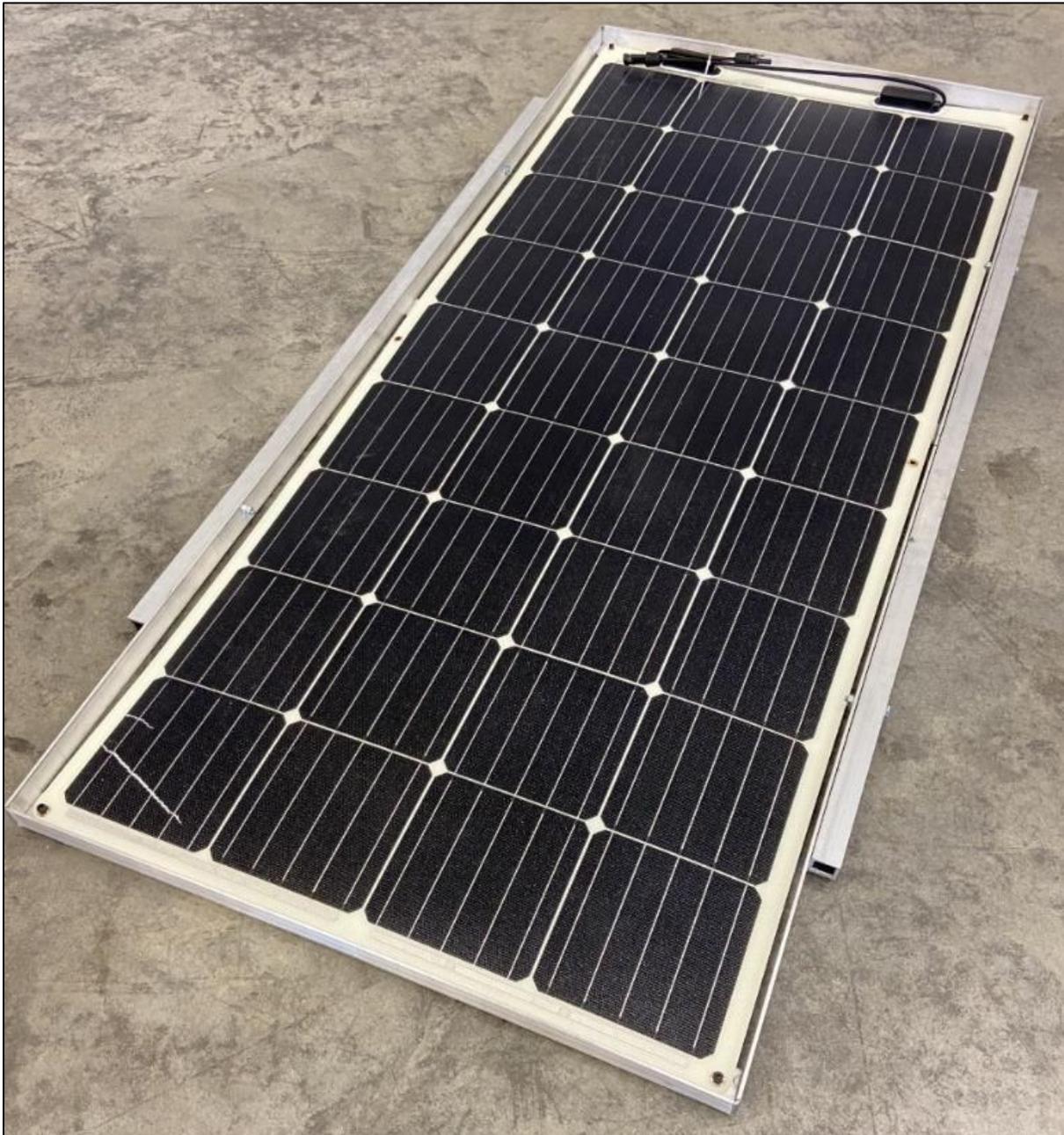
Figure 31: Final Assembly of Frames, Rails, and Solar Panels

4. Drive Train

       A new drive train design is included in the scope of the USI Solar Splash 2022 team's strategy of a complete redesign. The drive train connects the motor to the propeller, allowing the propeller to produce the thrust that propels the boat. Improvement of the drive train is necessary because the drive train directly affects the speed of the boat.

4.1 Motivation and Need

       A drive train for a boat includes all the components between the motor and the propeller. The motor and propeller are not considered part of the drive train, but they are very important in its design. A simplified drive train is shown in Figure 32. The drive train begins with a coupler that connects the motor to the drive shaft, allowing it to rotate and transmit torque. The gearbox is attached to the drive shaft with another coupler and converts vertical rotation to horizontal rotation. The propeller is attached to the other end of the gearbox. The drive shaft and couplers are located inside the shaft housing, not shown in the figure below, for protection and drag reduction. The engineering problem the drive train must solve is to transmit as much power as possible from the motor to the propeller.
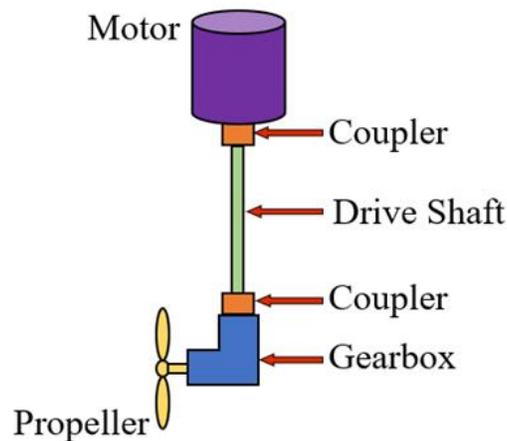
Figure 32: Drive Train Block Diagram

       The 2021 USI Solar Splash drive train can be seen in Figure 33. The drive shaft was a 24-inch-long hollow shaft made of 316 stainless steel, with an outer diameter of 1 inch and a 0.12 inch wall thickness. The gearbox was a Lampin MITRPAK R-121-C1 right angle gearbox with a 1:1 gear ratio. No couplers were used in the system. Due to factors associated with the pandemic, the design and construction of the drive train was delayed, and no testing or analysis was completed. The drive train was one of the subsystems that experienced failure and was repaired during the 2021 competition. During manufacturing, the drive shaft became misaligned, but it was only noticed when the motor was turned on for the first time at competition. To fix it, the team added a pillow block bearing to the top of the drive shaft and added vertical supports, both of

which can be seen in Figure 33. To avoid repeating past mistakes, the new drive train will be designed to reduce misalignment.
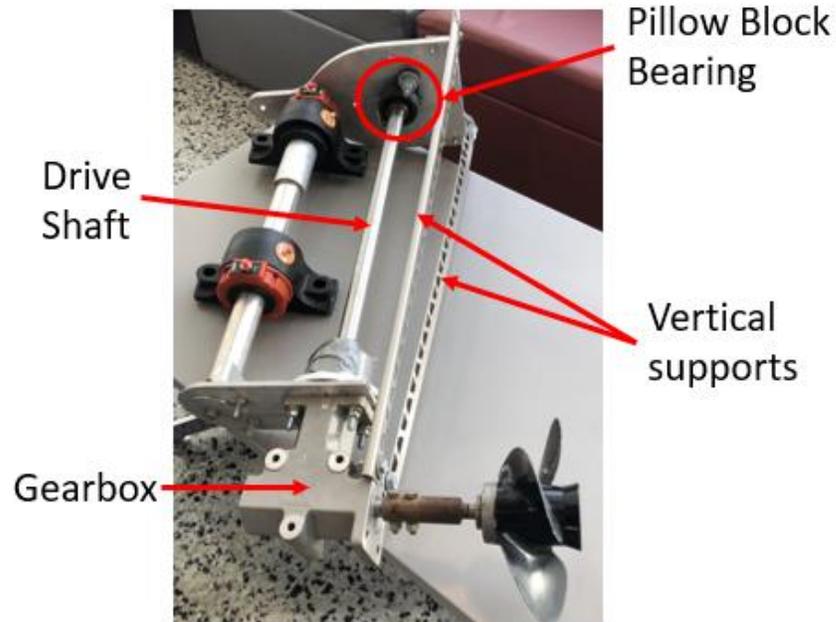
Figure 33: USI's 2021 Drive Train

4.2 Constraints and Engineering Design Choices

The technical requirements for the drive train are listed below in Table 4. Like all systems on the USI Solar Splash boat, the Solar Splash rules and regulations must be followed. The only official rule that directly relates to the drive train is that all materials must be water-safe and non-polluting, as stated in the first technical requirement [1].

Table 4: List of Drive Train Requirements

1. The drive train for the solar boat shall be made of water-safe and non-polluting materials [1].
2. The drive train for the solar boat shall transmit rotational power from the motor to the propeller.
3. The total weight of the drive train shall be less than 25 lbs.
4. The drive train for the solar boat shall be designed to reduce misalignment.

As mentioned previously, the motor and propeller are very important in the design of the drive train. The 2022/2023 motor was chosen by the electrical team and is a Lynch LEM 200-95 2x2. The 2x2 means it is two identical motors joined together on one shaft by the manufacturer, which provides double the power and torque of a single motor of the same type. This is a 48-volt motor, but the maximum system voltage is 36 volts to comply with Solar Splash rules [1]. At 36 volts, the motor has a maximum power of 15 kilowatts (20.1 horsepower), a maximum torque of

42 newton-meters (372 inch-pounds), and a maximum speed of 2916 rotations per minute (rpm). All components of the drive train must be rated to handle the motor's maximum power, torque, and speed. Another consideration for the motor is that it was manufactured in the UK, and all measurements are in metric units. The motor has a 19 mm output shaft, which affects the drive train design. Most of USI's previous Solar Splash boats have used English units, and most subsystems of the current boat use English units. Having a metric motor shaft means that all the drive train components must be metric to ensure good connections and good tolerances.

The propeller produces the thrust that makes the boat move forward in the water. Therefore, it is important that the drive train allows for the maximum transmission of power from the motor to the propeller. Also, the length of the drive train must allow for a safe and acceptable propeller clearance between the bottom of the hull and the top tip of the propeller. The propeller clearance should be between 20% and 70% of the propeller's diameter but should never be less than 2 inches [20]. The correct propeller size for the hull and motor was calculated by Melissa Kurz, not included in her senior design project. The propeller will have an 8-inch diameter and will be purchased and installed by USI's 2023 Solar Splash team.

4.3 Similar Projects

The Solar Splash website displays the top five reports from previous competitions [2]. Due to the competitive nature of the competition, teams often share the minimum necessary technical information for their designs. However, much can be learned from the previous reports. The following three teams had innovative drive train designs that helped them score very well overall in the Solar Splash events.

The University of Puerto Rico – Mayaguez (UPRM) team placed first in the overall Solar Splash competition in 2021. Their 2021 drive train was the same they used in 2019, when they placed second overall [2]. While designing the drive train in 2019, their goal was to reduce weight and drag by designing a drive train consisting of only two parts. They accomplished this and it led to a 43% weight reduction [21]. Figure 3 below shows the UPRM team's 2019 drive train design. The bottom piece is the gearbox, manufactured with a sleek, drag reducing design. The upper piece contains the drive shaft and shaft housing, which connects directly to the motor plate with screws. The UPRM team chose aluminum as the material for their drive train and manufactured it with a CNC machine and milling machine after conducting several stress tests [21]. A disadvantage to UPRM's boat and drive train design is that many of their components are not commercially available and had to be manufactured by the team.
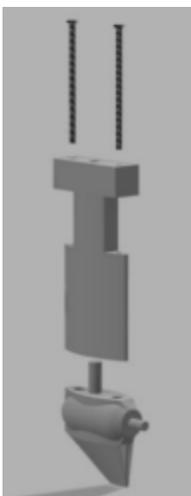
Figure 34: UPRM's 2019 Drive Train [21]

The College of New Jersey (TCNJ) team's boat won fourth place overall in Solar Splash 2021 and was awarded the Outstanding Drive Train Design for 2021 [2]. This team designed two drivetrains, one for the Sprint event and one for the Endurance event. According to the Solar Splash rules, the boat configuration may be modified, but all extra components must be stored on the boat for all races [1]. TCNJ's design was modular and allowed their team to easily reconfigure the boat for different competitions [22]. Both configurations are shown in Figure 35 below. Their Sprint drivetrain included two motors, a custom drive shaft, and a surface-piercing propeller that TCNJ has used since 2012. This configuration had an approximate output to the propeller of 4,500 RPM. The Endurance configuration included one motor, a longer drive shaft, and a larger custom-built propeller that sits fully under water. The Endurance configuration was designed to propel the boat long distances under low power draw. Both systems were made of aluminum to reduce weight. Having two separate configurations helped TCNJ optimize power for the different types of races but having two drive trains added 60 pounds to the total weight of the boat [22].

Figure 35: TCNJ's Sprint configuration (left) and Endurance configuration (right) [22]

Cedarville University won first place overall in Solar Splash 2019 [2]. The Cedarville team built a combined drive train, instead of using two separate drive trains for Sprint and Endurance from previous years. This reduced the weight by 80 pounds [23]. They accomplished this by using one motor, two different propellers, and an adjustable drive train length and trim angle. Cedarville's SolidWorks model of the combined drive train is shown in Figure 36 below. The yellow piece labeled 'Yamato LGU' is the gearbox and propeller shaft. Cedarville's drive train was designed by calculating the efficiency and gear ratio requirements for the drive train after using OpenProp software to determine the necessary speed requirements for both propellers [23]. The Endurance propeller was much larger than the Sprint propeller and required a longer driveshaft. Cedarville's solution was to add plastic and aluminum extensions to the drive shaft, shown as the red and blue parts, respectively, in Figure 36. To extend the length, a clamp is loosened, the entire drive train is slid down to the correct length, then the clamp is retightened, and the extensions are added [23]. The completed design was simple and easy to reconfigure between events while reducing weight. However, it is difficult to find a single motor that will satisfy all the needs of both the Sprint and Endurance events.
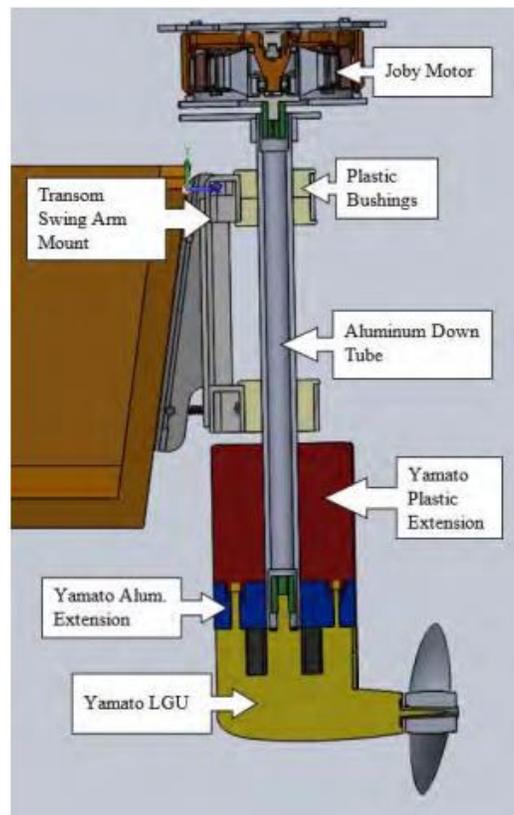
Figure 36: Cedarville University's 2019 Drive Train SolidWorks Model [23]

4.4 Conceptual Designs
4.4.1 Concept 1: Outboard Motor

The first concept is an outboard motor system and is the most common type of propulsion system for small boats. Figure 37 shows a drawing of the drive train design for an outboard motor.
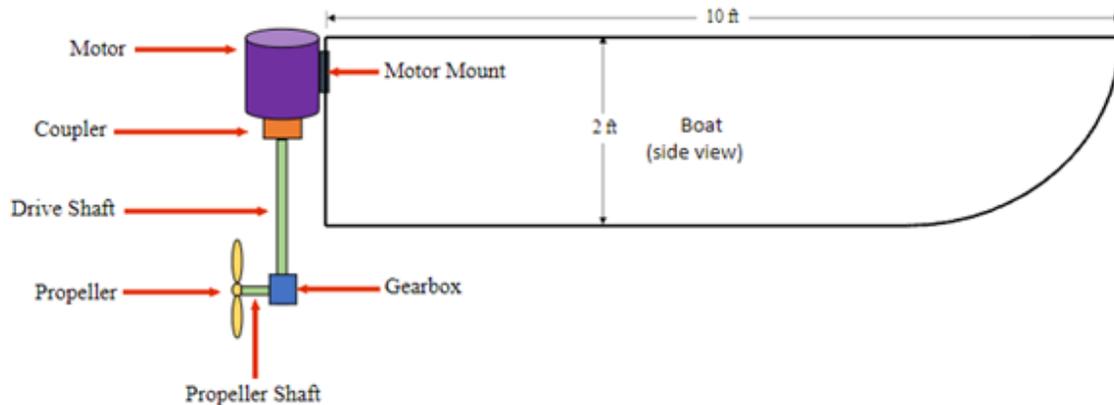
Figure 37: Conceptual Design 1: Outboard Motor

This type of propulsion system is called an outboard motor system because the motor is located outside the boat. Because this is the most common type of motor configuration, parts are more commercially available and often cheaper. Another advantage is that this system can be easily repaired because all components are easily accessible from the back of the boat. Also, due to the location of this system, the motor and drive train can be easily removed for travelling to and from the Solar Splash competition. The disadvantages include an increase in drag and a decrease in stability due to the components' locations on the back of the boat.

4.4.2 Concept 2: Inboard Motor

An inboard motor is a common type of propulsion system for larger boats, such as yachts. Figure 38 shows a drawing of the drive train design for an inboard motor. The components are in the same order as the outboard design, except the motor is located inside the boat and the drive shaft penetrates the hull at an angle. Also, a rudder is required for steering.
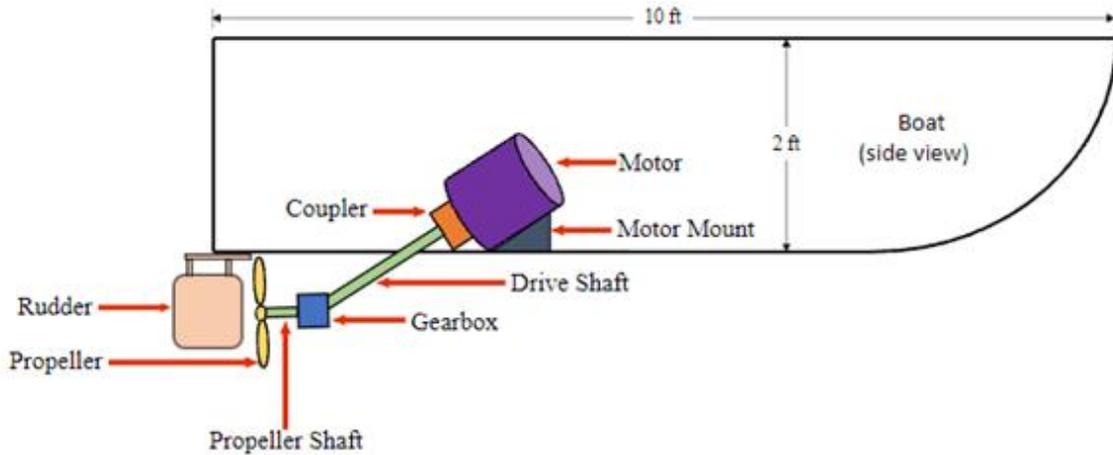
Figure 38: Conceptual Design 2: Inboard Motor

This type of propulsion system is called an inboard motor system because the motor is located inside the boat. The advantage of an inboard motor is that it has less drag than an outboard motor and a lower center of gravity because the components are located inside the boat, not outside the boat. The disadvantages of this system are that a hole must be drilled through the hull, which weakens the hull and increases the risk of water leaking inside the boat. Another disadvantage is that an inboard motor takes up space inside the hull, which limits the locations of the other components of the solar boat that need to be stored in the hull.

4.4.3 Concept 3: Outboard Surface Drive

In a surface drive design, the propeller is not fully submerged in the water. An outboard surface drive design, shown in Figure 39, is like a regular outboard motor design, except the drive shaft is shorter and the propeller shaft is longer to allow the propeller to pierce the water's surface.
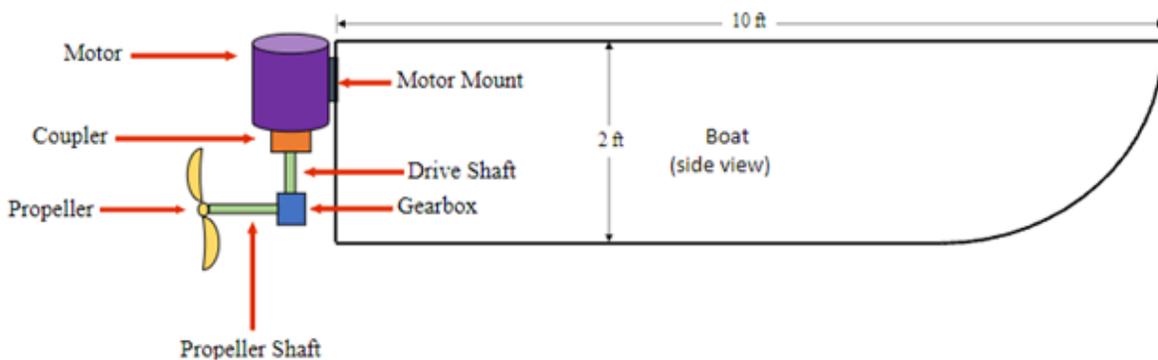
Figure 39: Conceptual Design 3: Outboard Surface Drive

The advantages of an outboard surface drive are that it yields higher speeds and has less drag than an outboard motor system. This is because the propeller is not fully submerged, so there are less components in the water. For the same reason, an outboard surface drive system eliminates the possibility of cavitation. This system has the same steering benefits as an outboard motor

system. A disadvantage of an outboard surface drive is that it decreases stability more than an outboard motor system because the propeller shaft is longer, which puts weight further from the boat's centroid. Also, it requires a surface drive propeller, which is a more expensive and less common type of propeller.

4.4.4 Selected Concept

After weighing the advantages and disadvantages of all three possible designs, Concept 1 was chosen for critical design. The outboard motor system was chosen for its ease of maintenance and ability to be removed for travel. Concept 2, the inboard system, was eliminated due to the hull. There is already limited room inside the hull, and the team decided it would be too difficult to try to fit the motor and drive train inside the hull with the skipper and all other components that must be inside the hull. Also, the USI team wants to be able to reuse the hull, and an inboard system requires a hole to be drilled through the bottom. This would impact the integrity of the hull for future teams and limit future configuration options. Concept 3, the outboard surface drive, was eliminated because the USI team wanted to keep the subsystems relatively simple during the complete redesign of the boat this year. A surface drive also places more stress on the hull than a regular outboard system because of the longer propeller shaft, and the team wanted to avoid any unnecessary stress on the back of the hull. In addition, the USI team opted to use one motor, drive train, and propeller for all the races, to aid in the simplicity of the boat.

4.5 Engineering Design

For the final design of the drive train, the drive shaft dimensions, and material were selected using a fatigue analysis and other calculations. Next, a gear ratio was chosen, and a gearbox was selected to meet the speed requirements of the motor and propeller. Finally, couplers were selected that meet the speed and torque requirements of the system and connect the components. The final SolidWorks assembly of the drive train can be seen in Figure 40 below.

Figure 40: SolidWorks Drive Train Assembly Exploded View

4.5.1 Drive Shaft

The drive shaft is the component of the drive train that transmits rotational power from the motor to the gearbox. Because the motor shaft is keyed, a fully keyed shaft was selected to transmit torque. The length, material, and diameter of the drive shaft were chosen.

Calculating the length of the drive shaft consists of adding the vertical heights of the components on the rear of the boat, including the gearbox, hull, and motor mount. Figure 41 below shows a view of the back of the boat that shows these vertical distances needed for this calculation. Note that Figure 41 is not to scale. The distances are shown in both inches and millimeters because the motor mount, hull, and propeller were designed in English units, but the gearbox was designed in metric units.
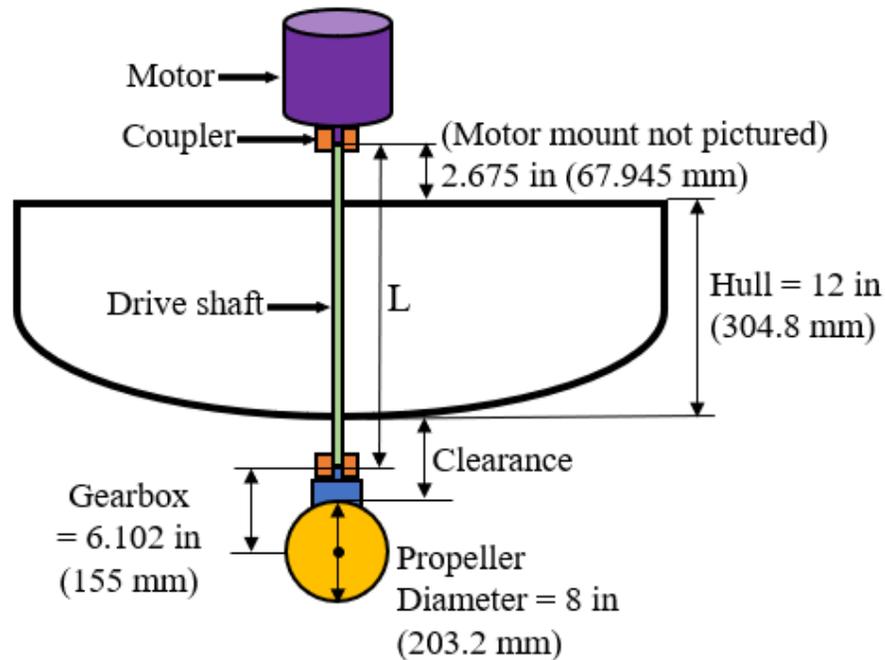
Figure 41: View of the Back of the Boat for Drive Shaft Length Calculations

The motor mount, not pictured in Figure 41, holds the bottom of the motor shaft 2.675 in (67.945 mm) above the top of the hull. The height of the back of the hull is 12 in (304.8 mm) in the center. Clearance is the distance between the bottom of the hull and the top tip of the propeller and depends on the propeller diameter. The gearbox has a height of 6.102 in from the top of the vertical input shaft to the middle of the horizontal output shaft. The chosen gearbox will be discussed in further detail in section 4.5.2. As mentioned previously, the propeller diameter is 8 in (203.2 mm).

The clearance is the most important value for this calculation. For safety, the minimum clearance should be 20% of the propeller diameter but should be no less than 2 in [20]. Because 20% of 8 in is 1.6, a minimum clearance of 2 in was used for calculations. The maximum clearance between the bottom of the boat and top tip of the propeller should be 70% of the propeller diameter. The propeller is most efficient with a clearance less than or equal to this distance. Also, propellers that are located farther below the boat are more likely to get damaged [20]. The maximum clearance is 5.6 in. Using a clearance of 2-5.6 in and adding the distances, the minimum drive shaft length is 14.573 in (370.15 mm), and the maximum drive shaft length is 18.173 in (461.59 mm). For fully keyed shafts, 400 mm lengths are commercially available and would not require any additional machining. The length of the drive shaft is selected to be 400 mm (15.748 in).

Next, the material of the drive shaft was chosen. The materials considered were 2024 Aluminum, 1045 Carbon Steel, and 304 Stainless Steel. The material properties that were considered include yield strength, ultimate tensile strength, corrosion resistance, and density. A high yield strength is desired. Ultimate tensile strength is used to determine the fatigue life of the drive shaft, so a high value is also desired. The drive shaft must be corrosion resistant because it

will get wet, and it will allow future teams to reuse the drive shaft. Density was considered because the weight of the system is limited to 25 lbs., as listed in requirement 3 in Table 4.

Material properties were found using MatWeb, an online database of material properties [19]. The properties are summarized below in Table 5. The colors on Table 5 indicate how the materials compare for each property. The green color designates "best", the yellow color designates "acceptable", and the red color designates "worst". 1045 Carbon Steel was eliminated because it is not corrosion resistant. The material with the best properties for the drive shaft was determined to be 2024 Aluminum. However, 304 Stainless Steel was selected as the final material for the drive shaft because there are more size options for this material for fully keyed shafts in metric units.

Table 5: Drive Train Material Property Comparison

Material	Yield Strength (MPa)	Ultimate Tensile Strength (MPa)	Corrosion Resistance	Density (g/cc)
2024 Aluminum	324	425	yes	2.78
1045 Carbon Steel	530	625	no	7.85
304 Stainless Steel	210	564	yes	8.00

Next, the diameter of the drive shaft was chosen. A fatigue analysis was conducted for several common diameters of keyed shafts. The diameters analyzed were 18 mm, 20 mm, and 22 mm, because these sizes all have the same size keyway as the motor shaft, 6 mm by 6 mm, according to ISO/R773. A fatigue analysis was completed for a 304 Stainless Steel fully keyed shaft for each diameter, using the material properties shown in Table 5. For simplicity and ease of calculation, a template was created using an Excel spreadsheet following the process outlined in *Shigley's Mechanical Engineering Design* textbook [24]. The Excel spreadsheets for each analysis can be seen in Appendix C.

The loading conditions for the drive shaft include a torque from the motor, and no axial loads or moment loads. The alternating moment load, Ma, is zero because the thrust force from the propeller was determined to be negligible within the analysis of the drive shaft due to the type of couplers and gearbox chosen. The mean torque load, Tm, was found from the motor specifications. The maximum motor torque provided by the manufacturer is 56 Nm, but due to the voltage of the USI boat, the maximum torque the USI boat will use is 42 Nm. However, 56 Nm was used as a worst-case scenario.

The keyseat on the drive shaft causes a stress concentration. The maximum stress is increased by a constant known as the stress concentration factor to account for the increased stress at the stress concentration. The stress concentration factors are estimated to be $K_t = 2.14$ for bending and $K_{ts} = 3.0$ for torsion [24]. These factors are adjusted to fatigue stress concentration factors by the notch sensitivity, q, which is a function of the notch radius and material's ultimate tensile strength [24]. The notch radius for the keyseat was estimated to be 0.02 times the diameter of the shaft.

The Endurance limit, Se, is found by modifying the ideal Endurance limit with modifying factors for this specific application. These modifying factors are called k factors, and each one is denotated with a k and a subscript a, b, c, d, or e, which correspond to surface finish, size, load, temperature, and reliability factors, respectively [24]. The subscript f represents a miscellaneous factor that considers anything that hasn't been previously accounted for. The surface finish was assumed to be ground. The size factor is found from the diameter. The loading factor was set equal to one because Von-Mises equivalents stresses were used for the analysis. The temperature and miscellaneous factors were set equal to one because they are not relevant in this application. The reliability was assumed to be 95% [24].

Table 6 shows the Distortion-Energy-Goodman fatigue factors of safety and the static yield factors of safety for each diameter of the drive shaft. All the fatigue factors of safety were greater than one, meaning that a 304 Stainless Steel fully keyed shaft with an 18 mm, 20 mm, or 22 mm diameter will have infinite life in fatigue. The static yield factors of safety are also all greater than one, meaning the drive shaft will not yield under these loading conditions.

Table 6*: Factors of Safety for the Drive Shaft*

Diameter (mm)	Goodman Fatigue Factor of Safety	Static Yield Factor of Safety
18	2.866	2.479
20	3.893	3.401
22	5.136	4.527

The final diameter of 20 mm was selected because this easiest diameter to obtain for 304 Stainless Steel shafts. Some manufacturers do not make 18 mm or 22 mm stainless steel shafts. Also, a drive shaft diameter of 20 mm makes it easier to select couplers, as 20 mm is a common size for bore holes. More information about the couplers is in section 4.5.3.

4.5.2 Gearbox

A MITRPAK R-121-C1-M Right Angle Gearbox was selected for the drive train. The outer housing is aluminum alloy, and the shafts are stainless steel. This offers good corrosion resistance. A 1:1 gear ratio was selected so there is no reduction of speed or torque from the motor to the propeller. This gearbox is rated for 3000 rpm, which is acceptable for the motor's maximum speed. The USI team has used this brand of gearbox in past competitions and knows that it functions well in this application. A picture of the gearbox can be seen in Figure 42 below.

Figure 42: MITRPAK Right Angle Gearbox

4.5.3 Couplers

There are two couplers used in the drive train. One coupler connects the 19 mm keyed motor shaft to the 20 mm keyed drive shaft, and the other coupler connects the 20 mm keyed drive shaft to the 19 mm keyed gearbox shaft. Because the motor and gearbox have the same shaft size, the same couplers can be used in both locations. The selected couplers are jaw and spider style couplers, seen below in Figure 43. Each coupler contains two metal hubs and an elastomeric spider. This style of coupler protects against parallel and angular misalignment and replaces the need for the pillow-block bearing that the 2021 team added during the 2021 competition. Each coupler for the drive train will include a 19-mm bore hub, a 20-mm bore hub, and a polyurethane spider.

Figure 43: Jaw and Spider Style Coupler

The selected hubs are clamping keyway hubs made of 2024 Aluminum. A clamping connection was chosen over a set screw connection because clamp-style connections have more

holding power and do not mar the shaft. The drive train may be assembled and disassembled multiple times between its initial assembly and the conclusion of the 2023 competition, and the clamp-style allows for easier disassembly and assembly. Also, the current team hopes that components such as the motor, drive shaft, and gearbox will be reused on future USI boats after the 2023 competition and avoiding marring the shafts increases the chances that these components will be reused. The keyways on the coupler match the standard size keyways of the motor, drive shaft, and gearbox, and allows for the transmission of torque between these components. For material selection, most manufacturers offer an aluminum alloy option and carbon steel option for the coupler jaws. 2024 Aluminum was chosen over carbon steel for its good corrosion resistance and lighter weight.

The elastomeric spider of a jaw-and-spider style coupler is usually made of rubber or plastic. The spider's material is chosen based on the speed and torque requirements of the application. The selected spider is a Durometer 98A spider. It is rated for a maximum speed of 8,000 rpm and a maximum torque of 550 in-lbs., which is acceptable for the motor's speed and torque. Its misalignment capability is 0.005 inches of parallel misalignment and 0.8º of angular misalignment. Additional benefits include zero backlash and fair vibration damping. Another benefit of a jaw-and-spider style coupler is that if the spider breaks away during use, the jaws directly connect, and the coupler continues to operate. This causes accelerated wear and noise, but it allows the spider to be replaced at a more convenient time. So, if the spider were to completely fail at competition in the middle of a race, the USI team would be able to finish the course and replace the spider afterwards. The only disadvantage to this coupler is that it cannot support any axial loads. Thus, the drive shaft housing must be designed to avoid axial loads in the couplers, which will be discussed in more detail in section 5.5.3 **Propulsion System Connection**

4.6 Handoffs

The drive train has met all its technical requirements. It is made of water-safe, non-polluting materials. The materials also offer good corrosion resistance so that components may be reused for future competitions. The total weight of the drive train is 18.1 lbs., which meets the requirement of a total weight of less than 25 lbs. Lastly, the drive train was designed to reduce misalignment with the use of jaw and spider style couplers. A bill of materials for all the drive train components for cost and weight can be found in Appendix A in Table 11 and

Table **12**, respectively. All the drive train components have been purchased. Detailed assembly instructions, found in Appendix E, will be provided to USI's 2023 Solar Splash team to be used in the 2023 competition.

5. Trim and Propulsion System Connection

The goal of this project is to design and build a connection between the drive train, the motor, and the hull with trim angle adjustment considerations. In this report, the propulsion system connection will refer to the shaft housing, motor mount and bearing that connects the drive shaft to the hull.

5.1 Motivation and Need

A trim angle adjustor is a mechanism used to change the angle of the motor mount and the drive train to reduce drag forces and change the center of mass when the boat is in motion [25]. Adjusting this angle will enable the drive train to adapt to the different needs of each of the three races.

For the purposes of this project, the propulsion system connection will be defined as how the motor and drive train connect to the trim angle adjuster. In this case, that connection is the shaft housing, the motor mount, and the bearing.

In addition, connecting the motor and drive train (the propulsion system), to the hull is a necessity. Without that interconnectivity, the boat will not have anything to propel it through the water. To enable this connection, a hull mount, a motor mount, and shaft housing were designed, and a bearing which bolts to the trim angle adjuster was selected. The hull mount is used as transom reinforcement, and the bearing and collars are used to keep the drive shaft vertically fixed, but still able to rotate in response to steering. Below in Figure 44 is the entirety of this propulsion system connection with trim angle adjuster feature, and Figure 45 is a closeup of the system without the hull.
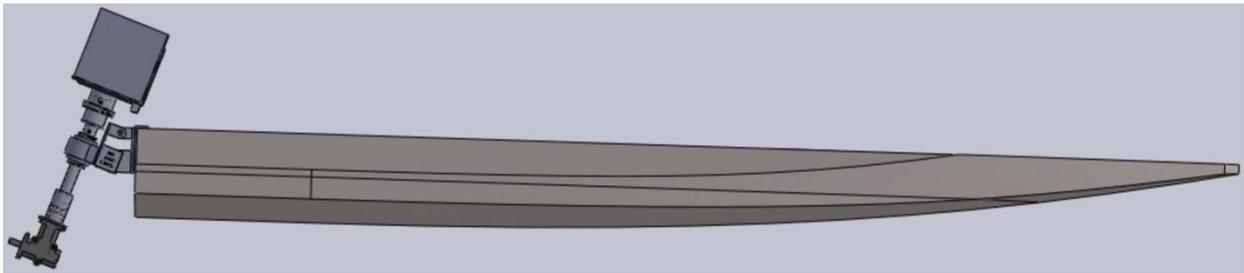
Figure 44: Hull with Propulsion System Configuration.
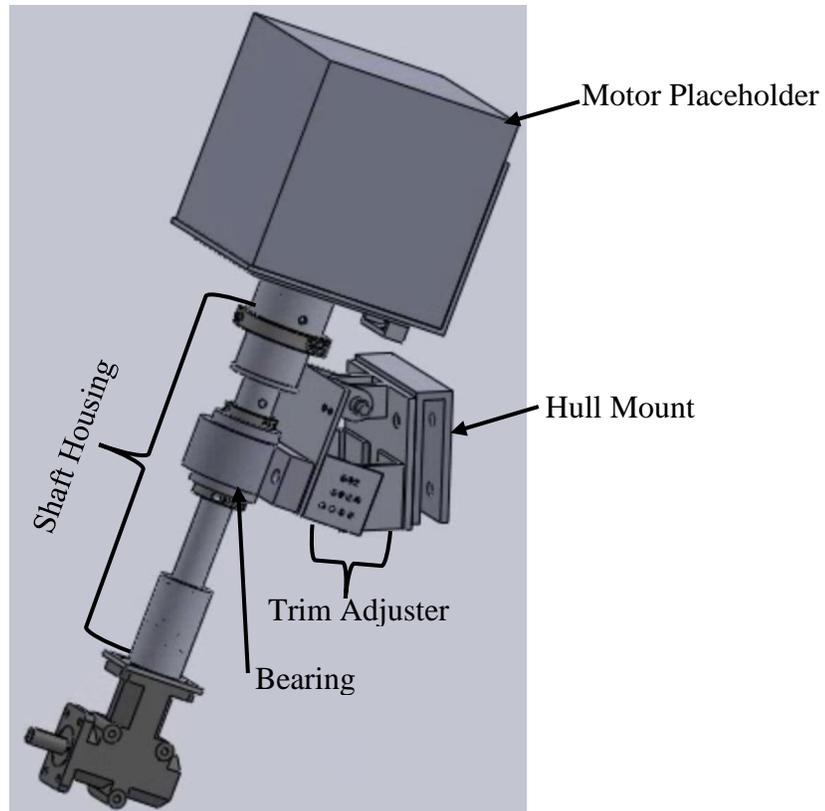
Figure 45: Drive Train Configuration with Hull and Drive Train Mount.

Motorized boats run differently according to the angle of the propeller shaft, and how it sits in relation to the hull and the water [25]. This allows subtle changes that can and will impact how the boat runs, its drag force, speed, fuel efficiency, etc. [25]. The angle in question is between the drive shaft and the transom is called the "trim angle" and is adjusted such that the propeller shaft is parallel to the surface of the water when the boat is moving. Both the force the water exerts on the boat when resting and moving, and the center of mass on the boat will affect this angle. For efficiency and ease, it will be adjustable. Figure 46 below shows the configuration without any adjustable feature, while Figure 47 demonstrates an example of how the propeller would rotate about the hinge while the hull is not moving. Figure 48 shows how the boat and propeller will sit in relation to each other and to the water's surface when the boat is moving. If the trim angle is too large, the boat will nosedive; if it is too small, the boat will angle up out of the water. The situation described occurs naturally if there is no trim angle adjuster present on a boat, causing increased drag and inefficient use of the batteries.
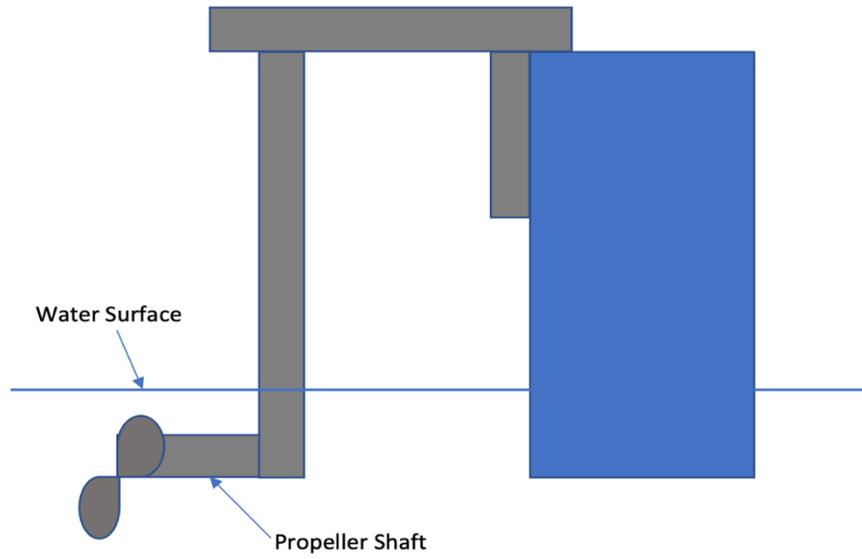
Figure 46: Image of the Propeller Shaft in Relation to The Water Surface with No Initial Rotation.
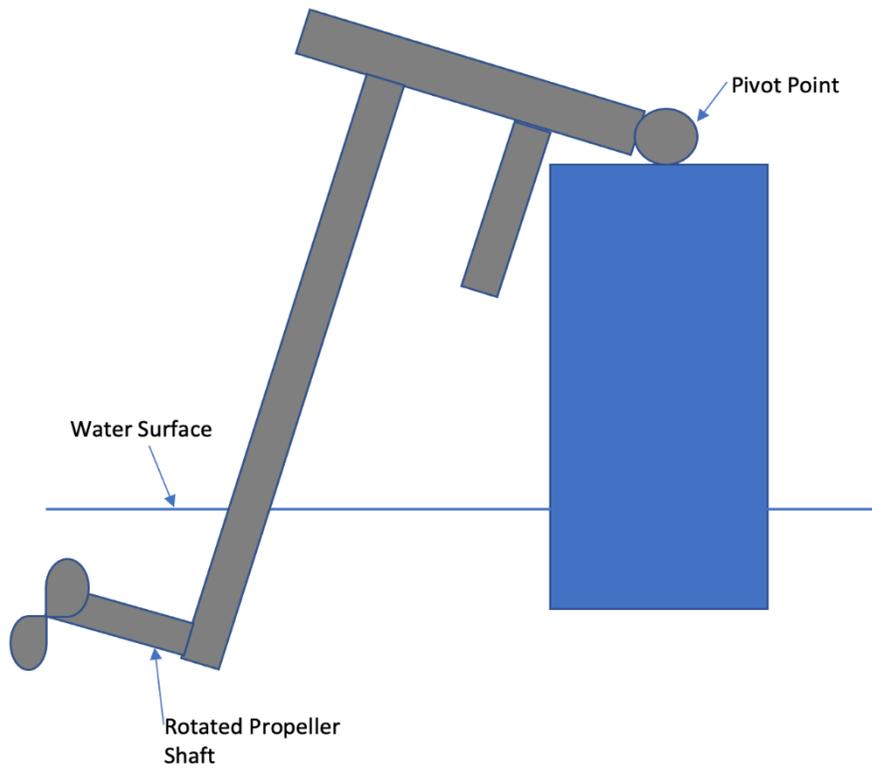
Figure 47: Image of Rotated Propeller Shaft in Relation to the Water Surface.
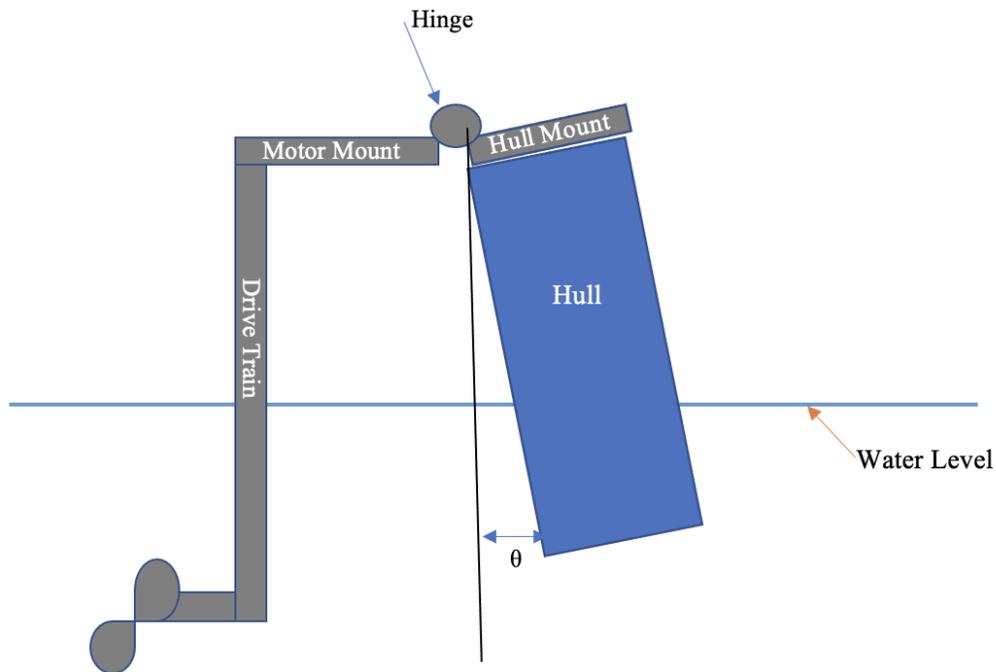
Figure 48: The Hull When Moving in Water.

In Figure 48 the image shows that the propeller shaft is parallel to the water's surface. For reference, θ is the angle the hinge rotates the drive train, also known as the trim angle.

There is front-end and back-end steering on this boat. Front-end steering refers to the steering cable and how the skipper controls the direction of the boat. The steering system selected is a rack and pinion steering system; the front-end steering system was designed by another member on the Solar Splash team.

Back-end steering refers to the actual turning of the propeller. Due to the propeller, drive train, and motor connection, that whole configuration must rotate at the same time. To do this, it was determined by Solar Splash's propulsion team that the shaft housing must be vertically secured, meaning it will not slide up and down, but still able to rotate using a bearing attached to the shaft housing and mounted to the trim angle adjuster. The actual mechanism to connect the front-end and back-end steering is yet to be designed, however, it will attach to the motor mount and the trim angle adjuster via a steering cable.

5.2 Constraints and Engineering Design Choices

The trim angle is ever changing. As the boat's thrust increases, the trim angle will need to be larger, and when the thrust is lower the trim angle decreases. The trim angle also changes when the center of mass changes. Since the components of the Solar Splash boat will all be fixed, the center of mass should only change when there are different skippers aboard, and when the solar panels are removed for different races. This is allowed during the competition, and though velocity changes throughout the duration of each race, the targeted velocity will be considered when selecting the trim angle.

In addition, if the front-end steering is unable to connect with the back-end steering system, the boat will only be able to travel in one direction. To traverse the course of the boat, it must have steering capabilities, or else it will not be functional for any of the races.

Below is Table 7, which shows the technical requirements of this design.

Table 7: Trim Angle Adjuster and Propulsion System Connection Requirements

Trim Adjuster and Propulsion System Connection	
Solar Splash [1]	Use only water safe, non-polluting materials.
	Measure less than 4'-11" from the waterline.
	Propulsion system fasteners must be locking or use Loctite.
	The length of the boat must be less than 19'-8".
OSHA Requirements [27]	Couplers must have a cover.
Additional Technical Requirements	Corrosion Resistance
	Display minimal displacement (no more than 1/16 in) in all simulations
	Have shear resistance (factor of safety = 4).
	Entire system must be lightweight (Less than 30 lbs.).

5.3 Literature Review

Other designs that have been implemented in the past were done by the University of New Jersey [22] and Cornell University [26]. While Cedarville did not have a trim design, they did design their hull to eliminate the need of a trim angle adjustor [28].

The image below in Figure 49 shows Cornell University's drive train mount with the trim angle feature. Below Figure 49 is Figure 50, which shows the entire assembly of the drive train along with their drive train mount. Essentially, the drive train sits in the bushings, as shown in Figure 50 and are denoted by the yellow circles, and the trim angle bracket rotates the system. A plate with multiple holes which represent the angles the system will need to be rotated about is shown in Figure 49, and a pin goes through the hole corresponding to the desired angle to set the trim angle [26].
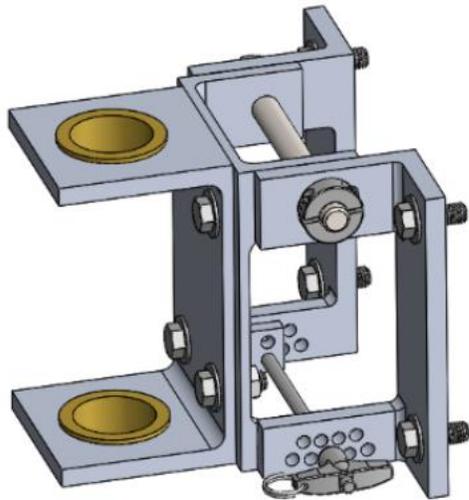
(a) Trim Bracket.   (b) Mounting Clamp.

Figure 49: Cornell University Trim Angle Adjuster [26].
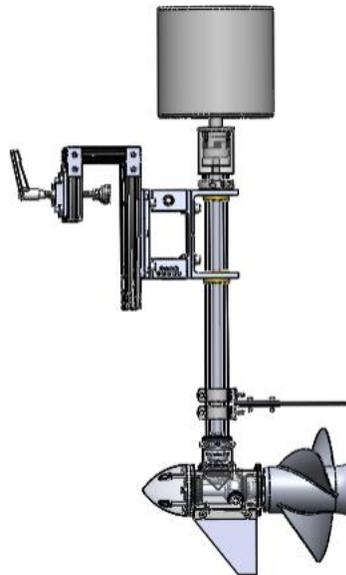
Figure 50: Cornell Full Assembly of Drive Train and Trim Angle Hull Mount [26].

New Jersey's trim angle adjustor, shown in Figure 51 below, had two main components: a triangular wedge that they rotated for specific angles, and a pivot with which the entire drive train and motor mount rotated about. In their report they described that for the different races they had designed the wedge, shown in Figure 52, to correlate with each specific race so they could simply rotate the wedge for optimum performance [22].

Figure 51: University of New Jersey drive train configuration [22].
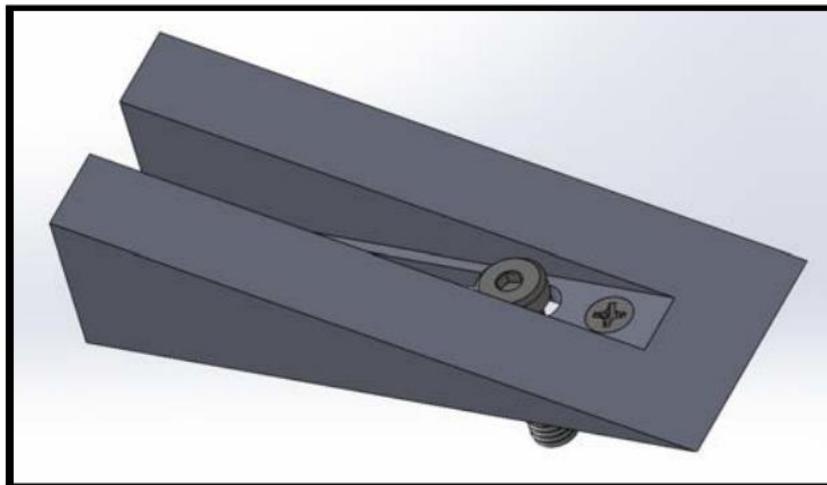
Figure 52: University of New Jersey Trim Angle Wedge [22].

Next, the Solar Splash senior design report for USI's Spring 2021 graduates was studied for steering/propulsion system ideas. The design for the 2021 competition included two bearings

with a shaft that was connected to the motor mount, which was connected to a rack and pinion steering cable via the top steering linkage. When the steering wheel was turned, the steering cable would rotate the motor mount, and the propulsion system would pivot about the pivot shaft. Figure 53 below shows the final design of the 2021 system.
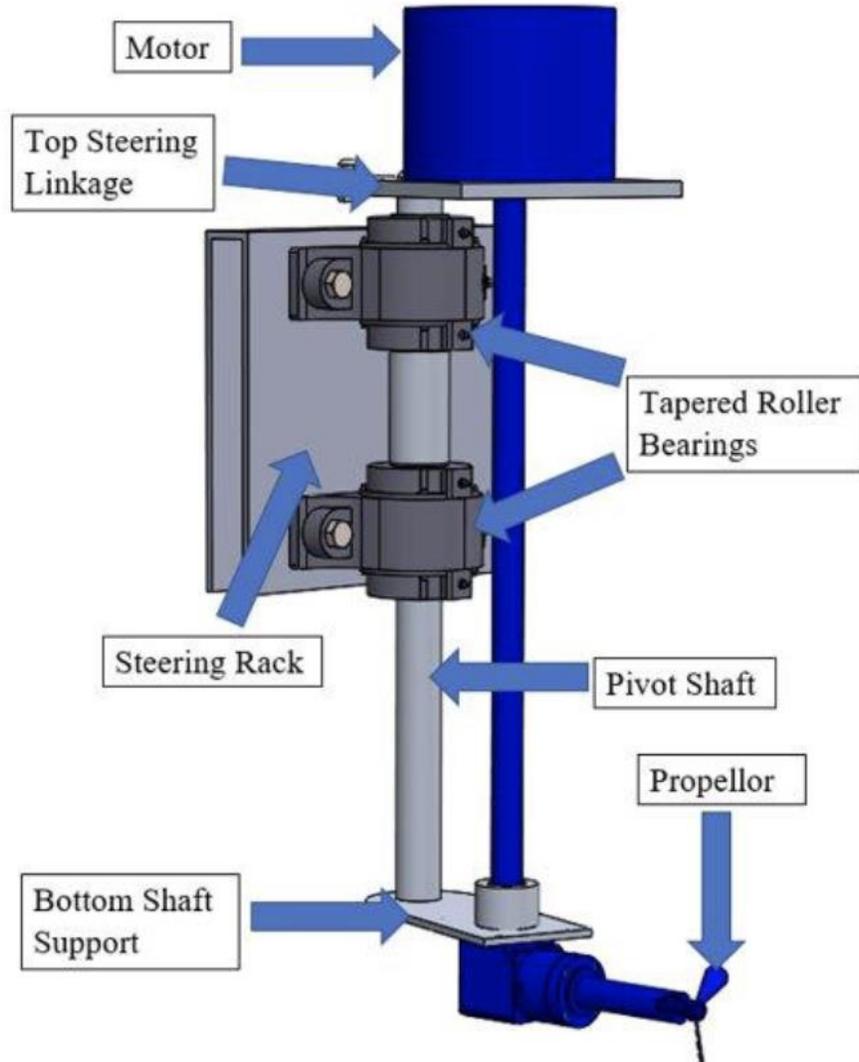
Figure 53: 2021 USI Back-End Steering Design [29].

5.4 Conceptual Designs

For this project, three conceptual designs were made that focus on a trim angle adjuster, with advantages and disadvantages to help decide the best design. Although the focus of these designs is trim adjustment, these systems also deal with connecting the motor and drive train to the hull, which is not shown in the images. All designs were made with the baseline that everything must have either a welded or bolted connection. None of these conceptual designs are drawn to scale.

5.4.1 Conceptual Design #1

This design, in theory, is a crank. There is a large gear that would be welded to the motor mount, which is labeled in Figure 54 below. Connected to that, is an idler gear, which is a smaller gear that is being acted upon by an external force. There is also another cut and welded gear that is connected to the hull mount. This will allow for the idler gear to have two gears to move around, with the gear attached to the hull mount being fixed in rotation and location, and the one attached to the motor mount being used as a track to angle the motor mount as needed. The idler gear will have a crank handle going through the center of it, and there will be plates connected to the hull mount with bearings to enable the crank arm to rotate freely.
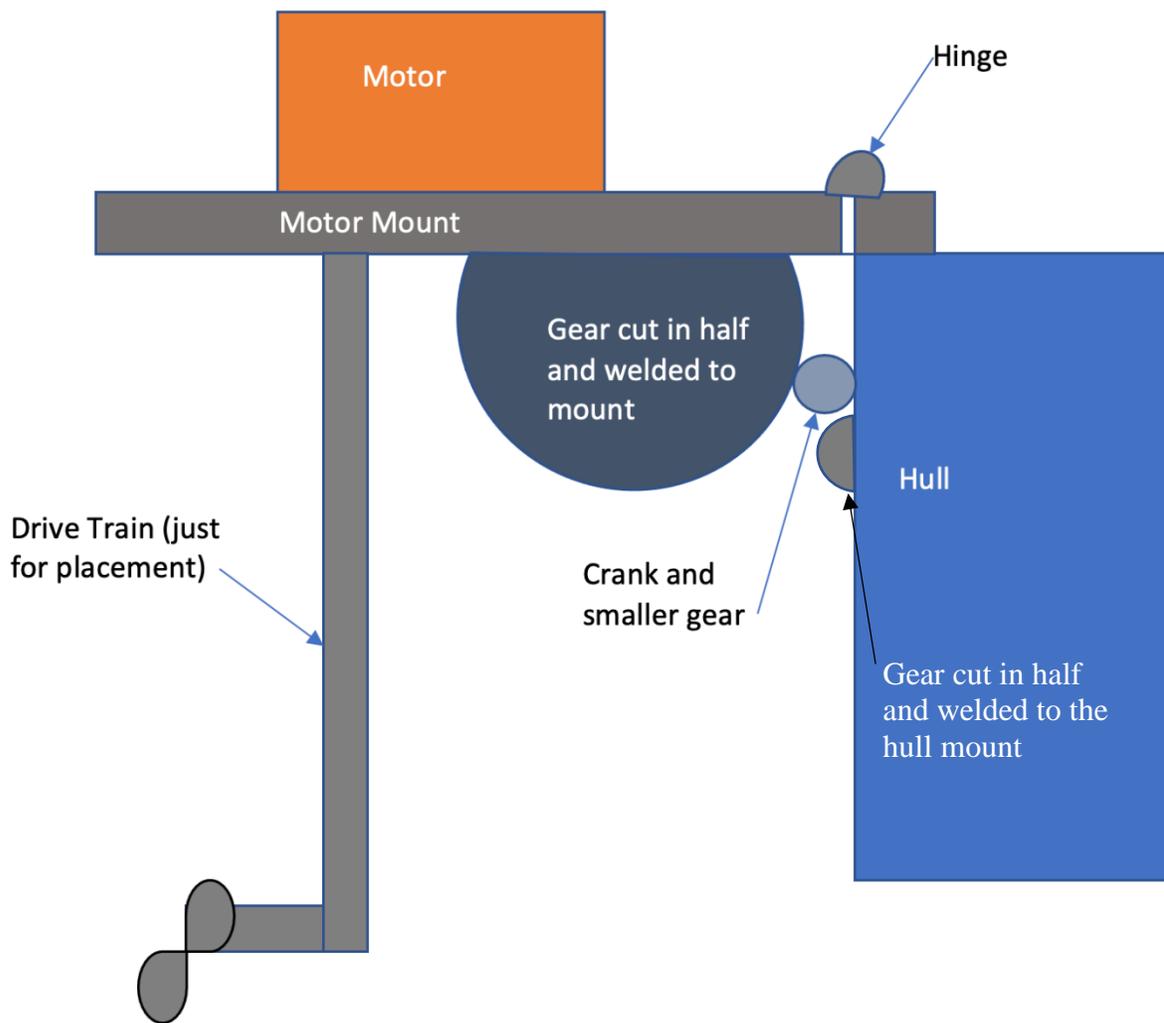
Figure 54: Conceptual Design #1.

The advantages of this design will allow for the trim angle to be easily adjusted via a crank. This is also able to be manufactured with welds and bolts when needed. This would hypothetically allow the system to rotate approximately 90°, which will allow for the position of the drive train

to be perpendicular to the hull. Allowing for this magnitude of rotation will be beneficial when launching the boat to avoid damaging the propeller by scraping it on the ground upon entry and exit of the shore.

There are some significant disadvantages of this design. A large one is the lack of a locking mechanism. If this system is unable to lock rotationally, there is no way it will be able to stay in position during a race. Instead, this will allow for the drive train to rotate up towards the water's surface instead of staying in position to keep the configuration in a good arrangement.

Another con to this design is fact that when the idler gear turns, the entire gear would have to move around the gear that is welded to the hull mount, which is not ideal for this application as the gears would not move in relation to each other in a manner that would work well in this design.

5.4.2 Conceptual Design #2

This design draws inspiration from the design that University of New Jersey described in their report; their final design is shown in Figure 51 and the wedge is shown in Figure 52 [22]. The basic idea in their design is that there is a wedge, dubbed a "support" shown as a trapezoidal shape in Figure 55 below. The shape will be made such that there are a few angles that will be preselected trim angles. The trapezoidal piece in question will have a bar running through it that will connect with two plates enclosing the sides of this trim adjuster. The way this works is when a new angle is needed one plate is removed, someone holds the drive train up, and the trapezoidal shape is slid out of position, rotated, and then inserted back in and the plate is replaced. The way this is supposed to adjust the trim angle is that the shaft housing of the drive train will rest against the rotating piece and be held down by some sort of strap.
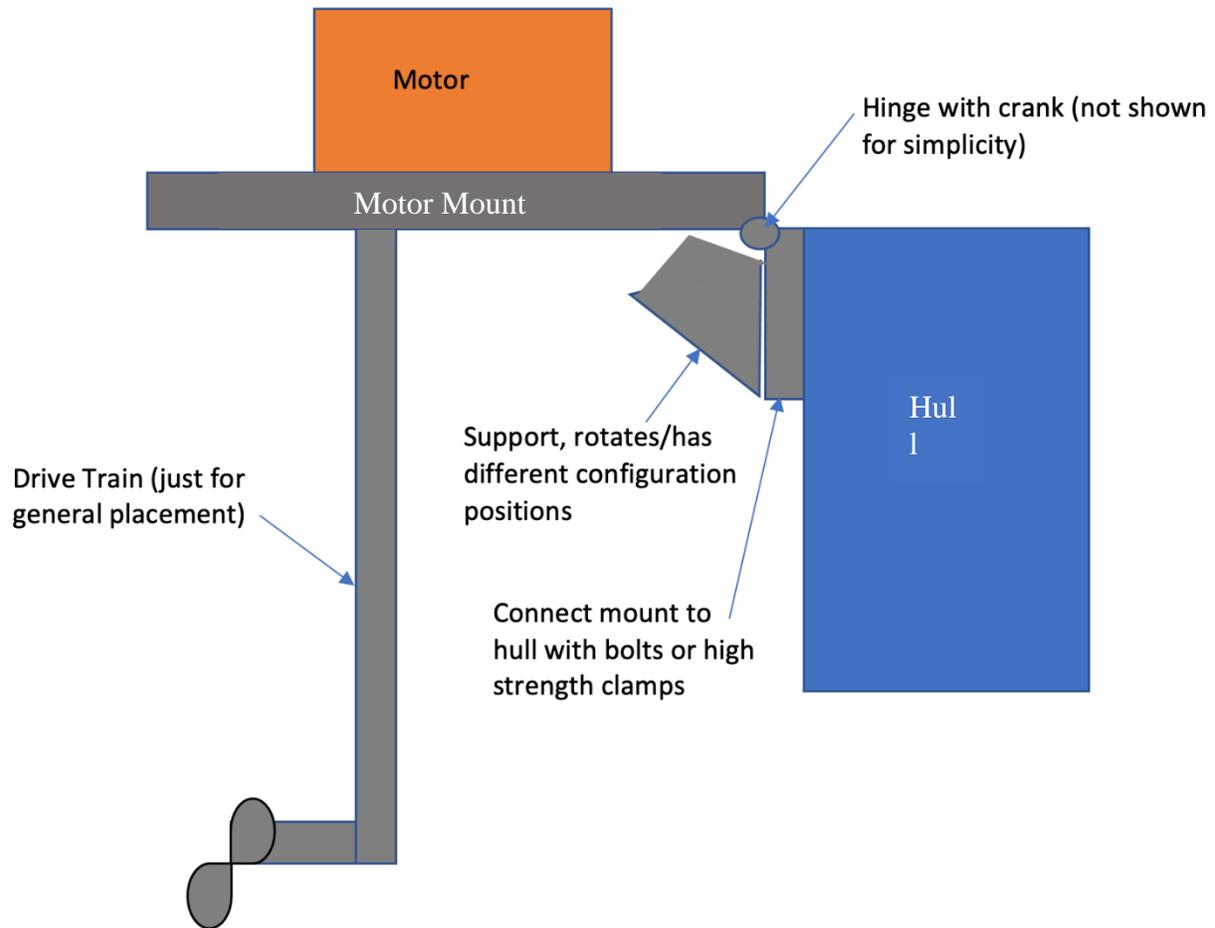
Figure 55: Conceptual Design #2.

There are a couple of advantages to this design. One positive result of this design is the wedge that was described. Since there is only one piece, that means that there will be predesignated angles for this system, which will enable the USI Solar Splash team an ease of changing the desired angle without having to calculate the angles for each race and skipper. With this method, each side can be labelled so the team can select the correct angle simply by rotating the wedge.

With this design comes some difficulty. The layout of this system, shown in Figure 55, shows there is no easy way to rotate the wedge, since there is no crank function. The drive train will have to be physically supported by a team member by rotating that component of the boat along the hinge, which is denoted in Figure 55, and holding the drive train in that elevated position. The next step is to remove the wedge from the system, rotating it such that it is facing the correct way, and then reassembling the mechanism.

Another difficulty with this design would be physical limitations of the students who go to competition. If none of the students in attendance can hold the drive train at an elevation long enough for the locking pin to go through, then this design is not the best idea.

The third drawback of this design is the difficulty of securing the drive train in the correct location for each angle needed. Simple ideas could be used, like securing with zip ties, however, that is not strong enough for this application.

5.4.3 Conceptual Design #3

The conceptual design shown in Figure 56 below draws inspiration from the trim angle design from Cornell University's trim angle adjuster in the 2021 Solar Splash Competition [26]. The hinge is a pivot point, with an L-shaped plate connection to the hull, which is pointed out in the picture. There will be multiple holes on this L-shape plate, and the red rectangle represents a plate of some sort that is intended to have holes as well, such that a pin of some sort can go through the L-shaped plate and the plate represented in red to fix the mechanism in place. This will allow the motor and drive train to be at that designated trim angle and stay in position. Though the conceptual design is very similar to that of Cornell University's design, improvements and changes will be made.
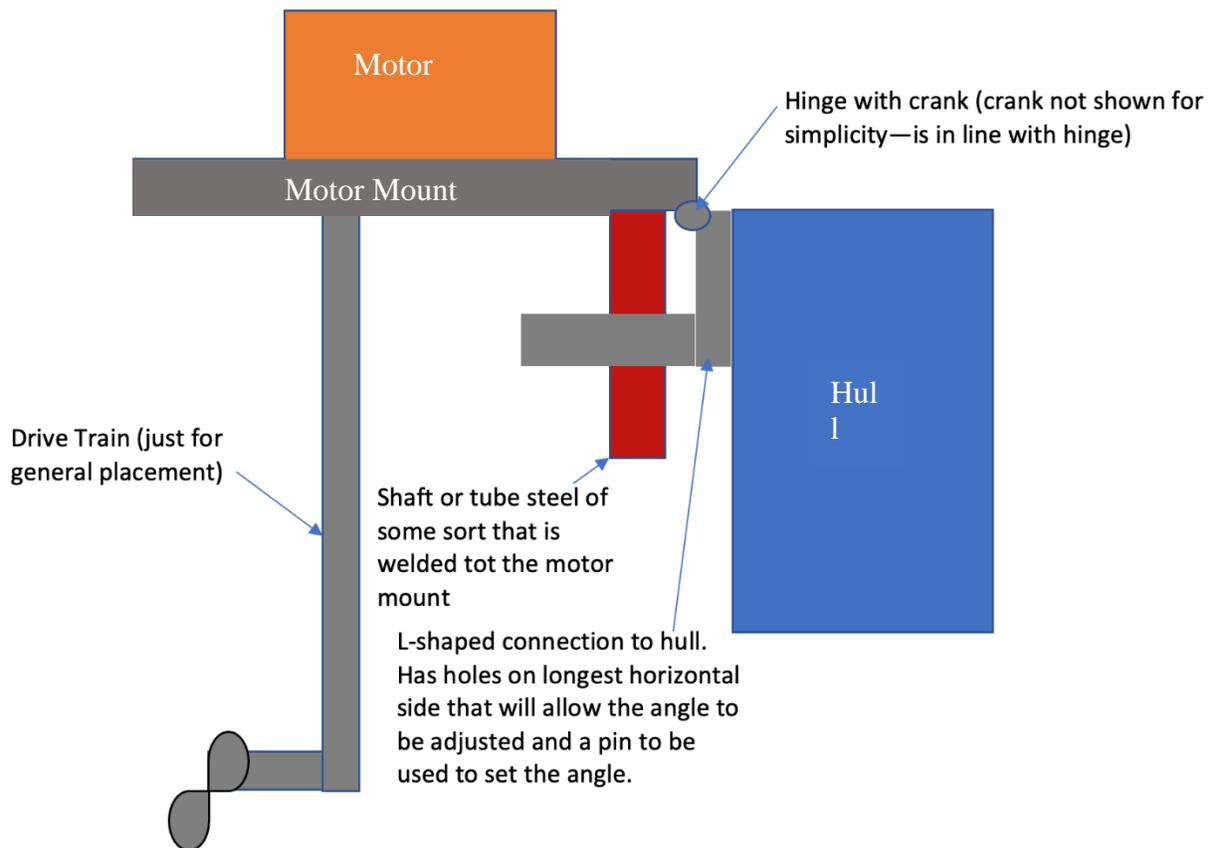
Figure 56: Conceptual Design #3.

There are some advantages to this design. The location is easily secured by inserting a pin or bolt through the holes, creating a locking mechanism. The L-shaped plates will be on both sides of the red plate for stability. This also allows for multiple predesignated angles of rotation, so the

48

angle does not need to be recalculated for each race and a marking can be placed to designate the holes for the angle that will be used most often.

The disadvantages of this design include the fact that it is not easy to adjust since it requires one team member to lift and hold the drive train to pivot the configuration, shown in Figure 56, while another team member places the locking pin or bolt. In addition, a strong hinge will need to be found or designed for this mechanism to not fail.

Another disadvantage to this design is the fact that physical strength will be required since the drive train will need to be held at the required angle as the angle is set. Of course, multiple people can hold up the drive train for the angle to be locked in place, but since the physical strength of the students on the team during competition will vary, it is still a concern.

After comparing the advantages and disadvantages of the three conceptual designs, Conceptual Design #3 was chosen. This design had the most flexibility as to how it can be designed, and allows for a multitude of angles, which is desired for this application.

5.5 Engineering Design

Three designs were made for the trim adjuster and propulsion system connection. The designs include a hull mount, shaft housing and motor mount, and a trim angle adjuster.

5.5.1 Hull Mount

As stated before, the transom is the back wall of the hull of a boat, and a hull mount is necessary for reinforcement of the transom. Since there is a large mass comprised of the drive train and motor and a thrust force acting upon the drive train, both of which are connected to the hull mount, reinforcement of this back wall is necessary. If the transom fails, the boat will sink. The hull was designed by another member of USI's Solar Splash team, so it was not analyzed for these designs.

This reinforcement comes in the form of a "U" shaped metal sleeve, which is shown below in Figure 57. It is made of three plates welded together with the material choice of 6061-T6 Aluminum. 6061-T6 Aluminum was chosen for this design because it is light weight, strong enough for this application (as will be demonstrated), inexpensive, and commercially available in the stock that was analyzed for these systems.

Based on prior Solar Splash projects surrounding the hull reinforcement, a selection process was conducted where the thickness of these plates was chosen. The two thicknesses in question are 1/4-inch plates and 3/8-inch plates. Figure 57 below shows the dimensions of the 1/4-inch option. The only dimension that would change when observing the 3/8-inch plates is the thickness, all other dimensions will be held constant.

HULL MOUNT (SIDE VIEW)   HULL MOUNT (FRONT VIEW)
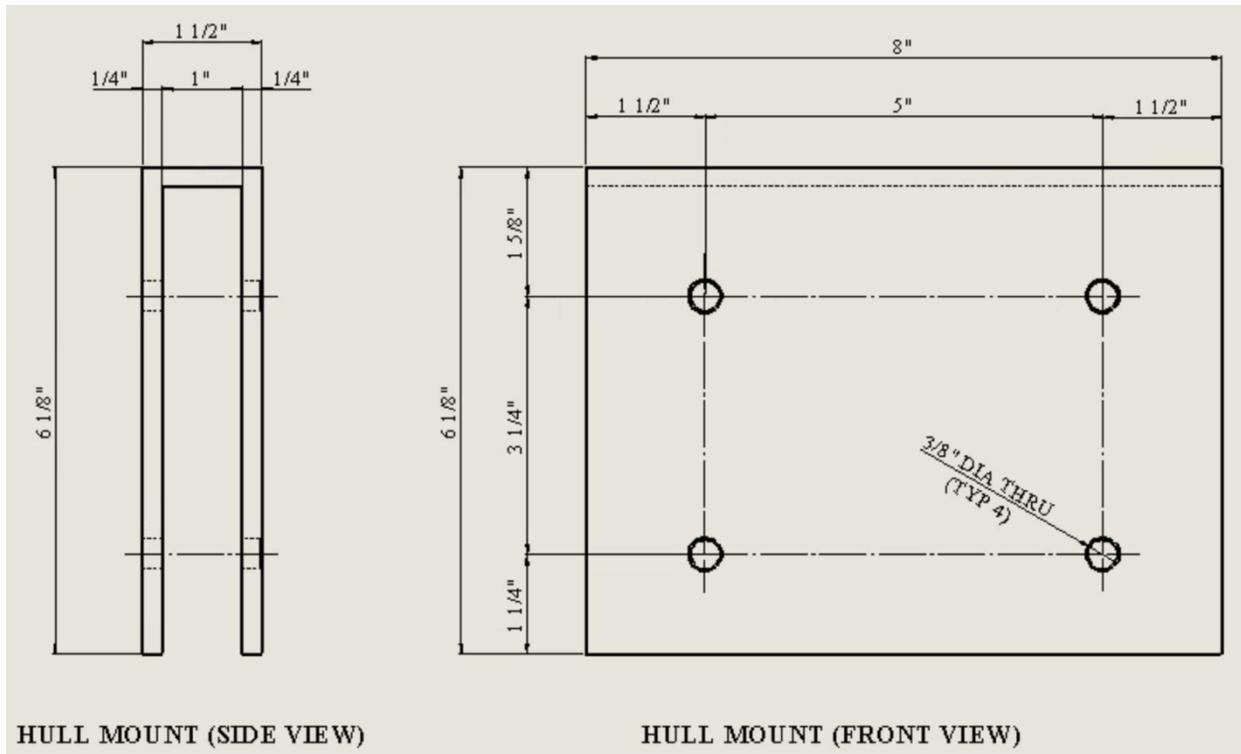
Figure 57: Hull Mount with Dimensions.

There are some clearances that the hull mount must account for. The first is the inside distance between the two larger plates. This distance must be 1 inch, because that is the thickness the transom is designed to be. Of course, once the hull is built this may need to be adjusted and reanalyzed, however, for the sake of this project, 1-inch is the dimension that is known. The other dimension that must be followed is the height of the hull mount. The depth of the transom is 6 1/4 inches, so the hull mount must not exceed, and in fact should be a little shorter, than that dimension in case of manufacturing errors or discrepancies when building the hull.

A Finite Element Analysis (FEA) study was conducted to compare the chosen thicknesses. Before this could occur, the applied forces needed to be calculated. No buoyant forces were considered for this simulation, though they will be present when the boat is in the water. Though this was not considered, it will not make this design weaker, it will just add a little factor of safety since the buoyant force acts in the opposite direction as the weight of the drive train, meaning the simulation will be designed to hold more vertical force than necessary.

One force the hull mount must withstand is the downward force of approximately 150 lbs. that is caused by the combined weights from the drive train configuration and the motor. This is the maximum value that was allotted to the propulsion system in the hull design. A vast amount of the Solar Splash construction is not complete enough for a final analysis; however, since the expected weight capacity of the hull is 500 lbs., the goal for the drive train configuration was for it to be no more than 150 lbs. total.

The next force that was applied was a horizontal force that is caused by the thrust from the propeller, since it causes the boat to move forward. This thrust force was calculated by Melissa Kurz. The maximum thrust value of 134 lbs. was increased to 180 lbs. for this application. Then a factor of safety of 2 was applied, meaning the force used in this simulation for the thrust value is 360 lbs.

There is also a force acting on the face of the plates around each of the holes to simulate the force exerted by the bolts once they have been tightened as much as possible. It was estimated that the average human can exert approximately 150 lbs. Since this is with a wrench, this is not the actual force the bolt is experiencing, just an approximation that will only be used for this simulation.

Figure 58 below shows the magnitude of the final forces applied, their locations, and the directions they were applied in.
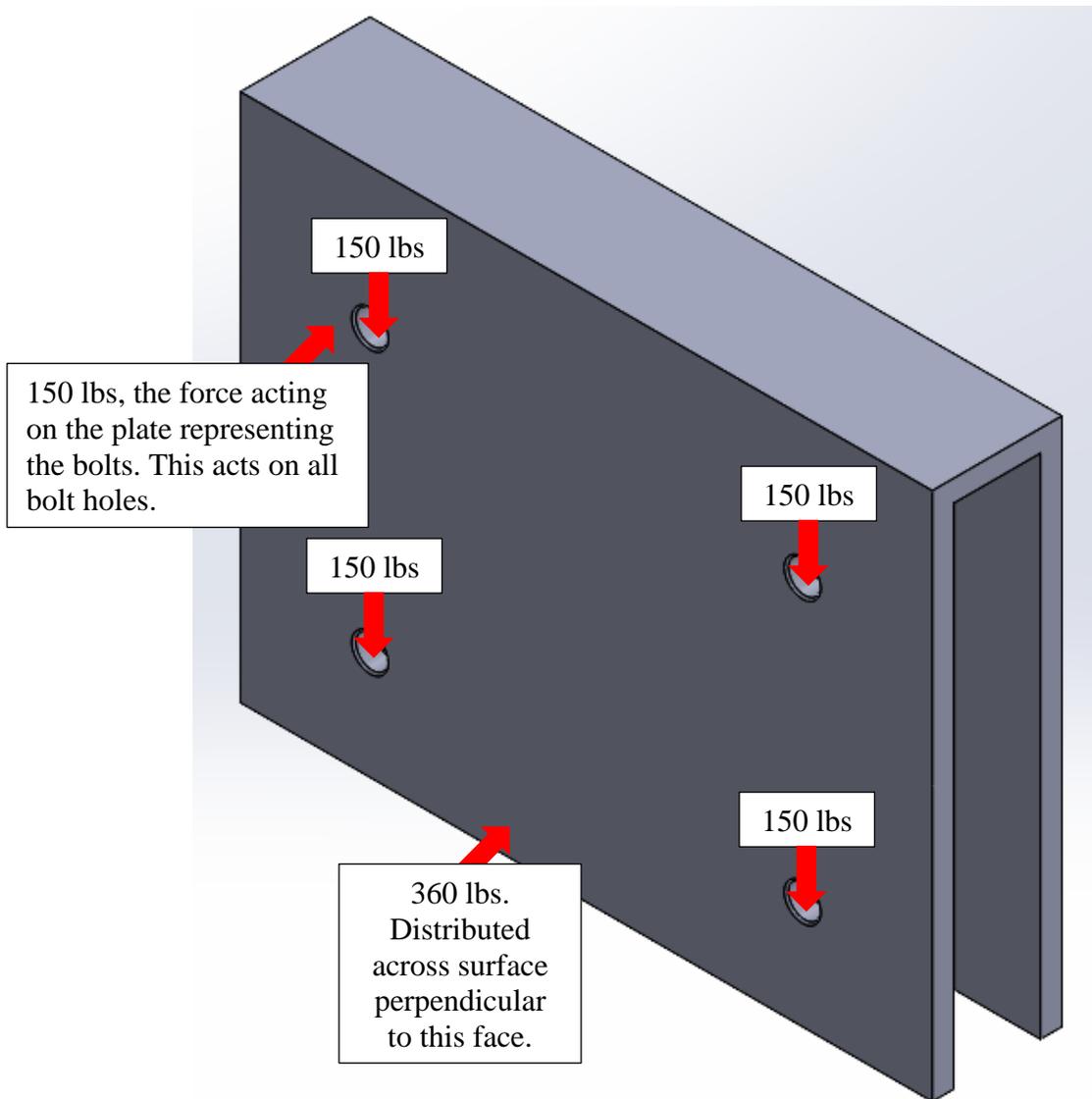
Figure 58: Hull Mount FEA Setup

Two different plate thicknesses were compared, 3/8-inch and 1/4-inch, for the hull mount. The same material and forces were applied. The results of this simulation are below in Table 8*Table 8: SolidWorks Simulation Results.*, and a cost comparison between the stock prices of the different thicknesses are also shown in Table 9. The resulting plots of the simulation are shown in Appendix D if further information is desired.

Table 8: SolidWorks Simulation Results.

Simulation Results		
Thickness	1/4 inch	3/8 inch
Maximum Displacement (in)	9.061×10^{-6}	1.241×10^{-5}
Minimum Displacement (in)	3.927×10^{-32}	3.937×10^{-32}
Maximum Stress (psi)	4.037×10^{2}	3.976×10^{2}
Minimum Stress (psi)	2.947×10^{-5}	6.223×10^{-4}
Weights (lbs.)	2.55	3.82

The thickness selection could go in a couple different ways. Weight optimization is key, and so the 1/4-inch thickness design would be best. Since the maximum stresses and displacements are approximately equal, the decision will be heavily based on the weight and cost of the different thicknesses. The 3/8-inch thickness design has a heavier weight, but it also has lower displacements and stresses. On the other hand, the 1/4-inch design has a lighter weight, which is favorable for this application.

The purpose of Table 9 below is to show how the cost of the 1/4-inch thickness compares to the 3/8-inch thickness. These numbers were taken from McMasterCarr.com for 6061-Aluminum stock. Based on the data, the 3/8-inch thickness costs approximately 1 1/2 times larger than 1/4-inch thickness. Based on this alone, the 1/4-inch thickness would be preferred over the 3/8-inch option.

Table 9: Cost Comparison for Various Stock Sizes.

Cost Comparison		
Size	1/4-inch Thickness	3/8-inch Thickness
8" x 8"	$19.69	$28.08
12" x 48"	$109.41	$155.98

Based off the SolidWorks Simulation results, cost, and the force, the 1/4-inch thickness should be chosen. This decision was made because the resulting forces and stresses were almost equal for both thicknesses, and the designs are approximately 1 lb. difference. Since optimization of the weight on the back end of the hull is key for this system, the 1/4-inch design was chosen.

The hull mount was designed first, so the material used for the rest of the systems was chosen to match. This is because when two different types of metals are in contact, they may

experience corrosion. In addition, other materials were investigated, however the sizes that are needed for the designs that would also meet the weight restriction were not available in other materials. In addition, the thickness of the plates used in the shaft housing, motor mount and trim angle adjuster was kept constant at 1/4-inch.

Once the plate thickness was determined, the next step was to make the hole pattern. A MATLAB program was used for this calculation, shown in Figure 82 in Appendix D. The final bolt size and type chosen was 3/8-inch grade 8 bolt.

5.5.2 Trim Angle Adjuster

Once the hull mount was designed, the trim adjuster design was conceptualized and then refined as information was provided throughout the semester regarding the weights of the components of the hull as the rest of the components of the boat were designed and finalized. For this design, Cornell University's design, as seen in Figure 49, was used as inspiration with the fact that there is a hinge mechanism with a locking pin setting the desired angle. A picture of the final hull mount and trim angle adjuster design is shown below in Figure 59.
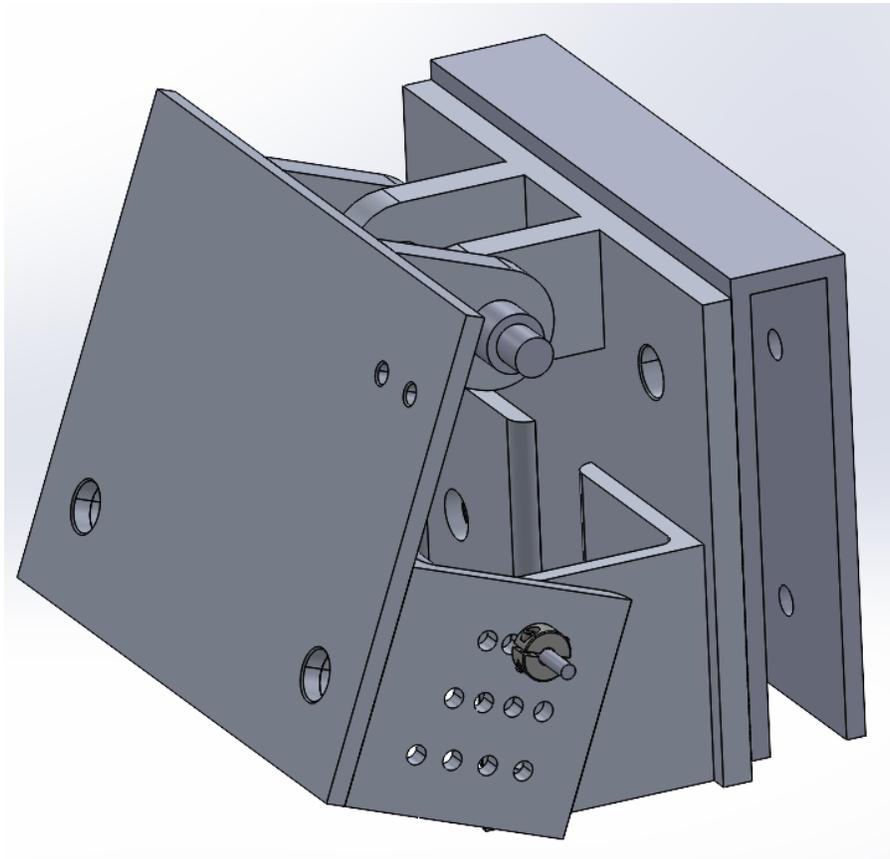
Figure 59: Assembly of the Hull Mount and the Trim Angle Adjuster.

Below in Figure 60 is an isometric view of one half of the trim angel adjuster. This half has all the holes at the different angles and is also what the bearing bolts to. The largest angle

possible with this system is 20º, which is the largest angle trim adjusters typically approach [25]. Figure 61 below shows the hole pattern in more detail.
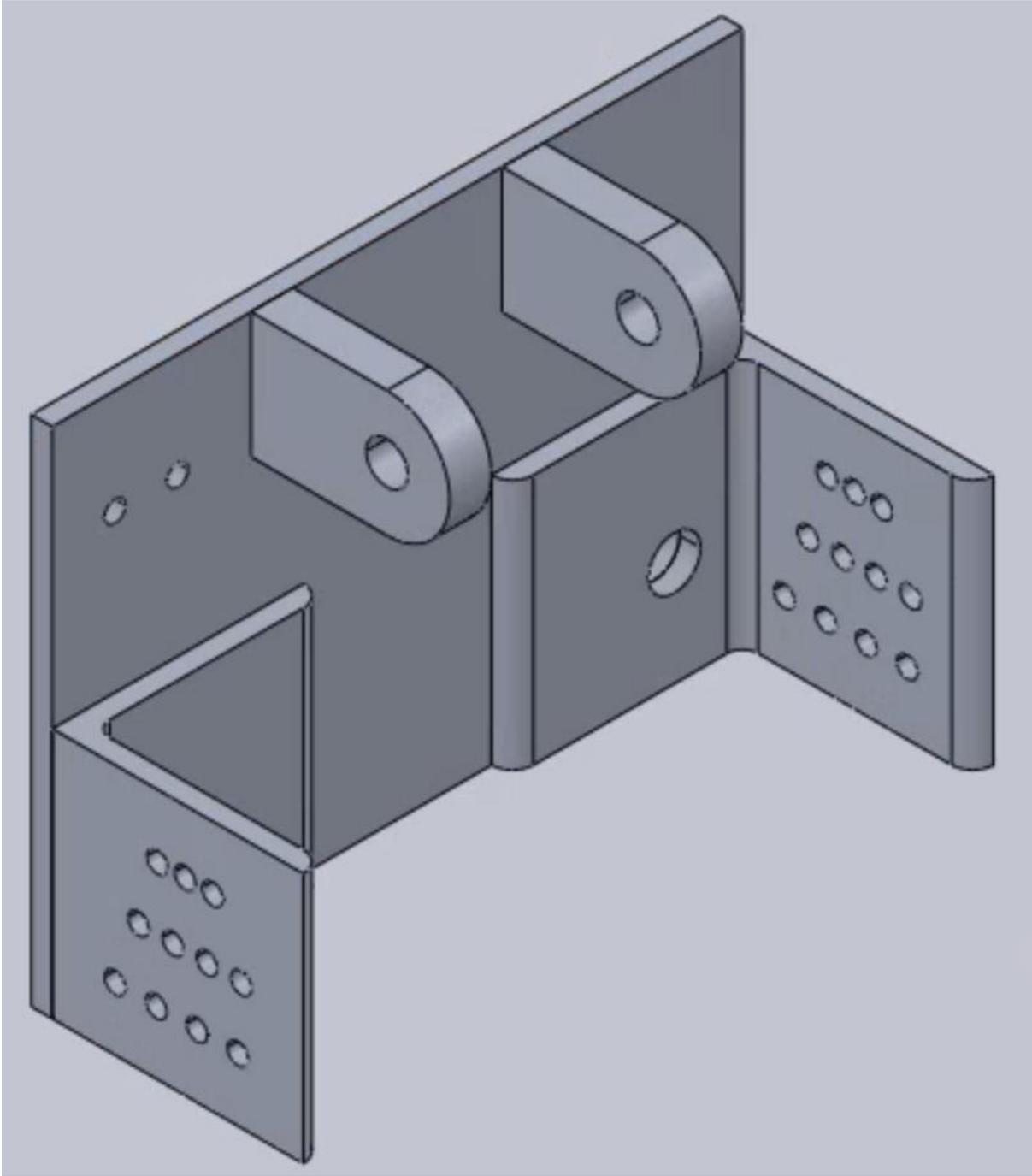
Figure 60: Half of the Trim Angle Adjuster. The side the bearing bolts onto.
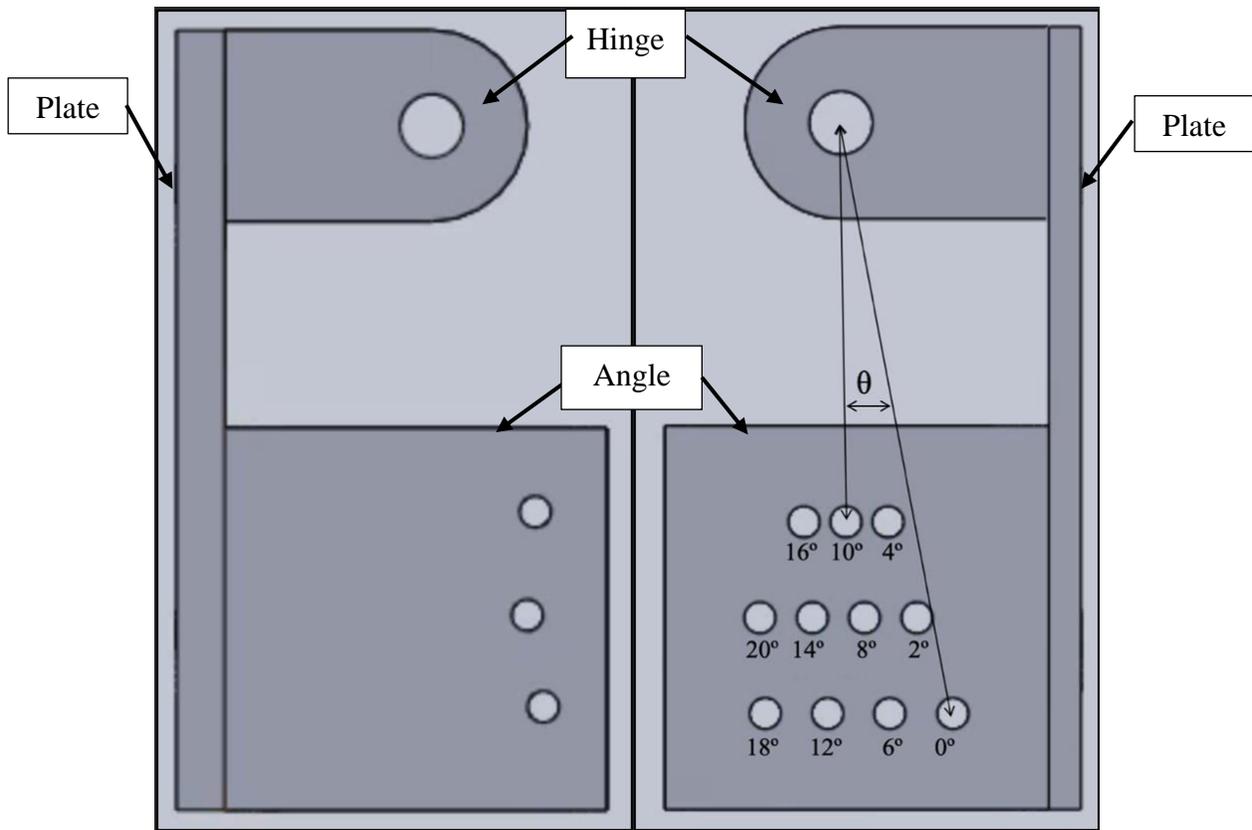
Figure 61: Side View of Trim Angle Adjuster Halves. The right side has a diagram of the hole pattern for the different angles, while the left is the image of the other part of the trim adjuster.

The hole pattern shown in Figure 61 above took some time to refine. Initially, the idea was to have the holes in a horizontal line starting at zero, and increase the angle by 2° horizontally, however, that made the holes overlap which will not work for this application. To combat this, the holes were rearranged until they were in an order that would be easy for a user of this system to follow and to ensure the holes were placed away from each other enough as to not overlap.

The first step was to set the different heights of the rows of the holes. Manufacturing details shown in Figure 89 in the Appendix D, shows more information. Since the angle is only 3" tall, the height was divided by 4 for an even hole distribution. In addition, the angles are arranged to kind of zigzag in order such that when the angle gets changed, if the person changing the angle only knows the zero/neutral point, they only need to be familiar with how the angles change and they will be able to easily select the desired angle. The diagram above will be provided, but should it be lost, a logical order was desired for a worst-case scenario.

When deciding where the holes go, the zero point is the only one that the horizontal distance really matters since the rest of the holes are moving away from the plate. That means, it needed to be far enough away from the inside leg of the angle, where the angle bolts to the plate as to give enough space for a bolt to be in place and still have the locating pin easily insert through the desired hole.
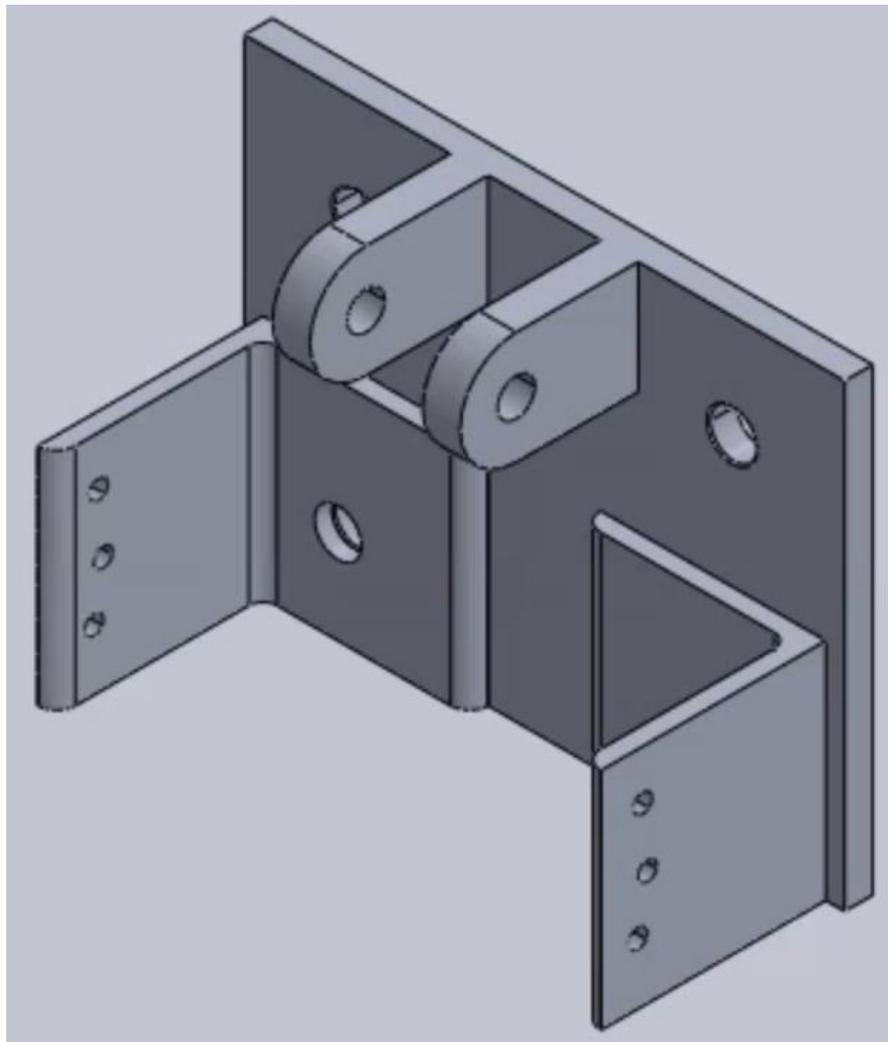
Figure 62: Half of the Trim Angle Adjuster. The half that bolts onto the hull.

Once the angle side of the trim adjuster was finished, the next step was to design the other half of the adjuster. The isometric view of the other half of the trim adjuster, which is the side that bolts into the hull, is shown in Figure 62. This side of the adjuster didn't require as many considerations because it was basically a mirror of the other half of the adjuster. The only things that changed was the hole pattern, the distance the L-angles were from the edge, and the location of the bars (where the hinge is), so the two halves of the adjuster fit together nicely.

The only issue that arose was the location of the holes. Initially the holes were all vertically in line and the zero point was set to locate all the points, however, that did not work because the holes did not line up. To get the holes to line up, some trial and error was required to get the holes as close as possible for every angle in every line. For the most part, the holes were just shifted horizontally, however, the top hole needed to be moved up slightly.

Once the design of the trim angle adjusters was made, the next step was choosing the material, sizing the holes, and choosing the thicknesses of everything used to make the system.

When different metals are in direct contact with each other they start to corrode [24], so the same material as the hull mount was chosen and analyzed to ensure it met the technical requirements. In addition, not all materials had the same commercial availability as 6061-T6 Aluminum, and if they were available, the final system did not meet the technical requirements. For ease of ordering materials and being consistent in a design perspective, the same plate thickness was used for the flat plates in which everything is either welded or bolted as was used in the hull mount design. The results of the FEA simulation are shown in Table 14 in Appendix D, showing the material chosen does meet the deflection requirements.

After choosing the material, the hinge and pin were sized. The following equations were used. Sy is the yield strength of the material, V is the applied shear force, n is the factor of safety, and A is the cross-sectional area of the item in question. Equation 3T was plugged into Equation 2T and solved for r, the radius, to determine the correct size of the rods. The final equation used is Equation 4T, after all substitution is completed.

$$\frac{Sy}{2n} = \frac{4V}{3A} \tag{2T}$$

$$A = \pi r^2 \tag{3T}$$

$$r = \sqrt{\frac{8nV}{3\pi Sy}} \tag{4T}$$

Different forces were applied for each shaft. First the shaft for the hinge was sized. A weight of 300 lbs. was used for V. This is higher than the combined thrust and weight loads of the system the rod will be supporting, however, it accounts for other forces act on it. For example, if a team member is working on the propulsion system, they will be applying extra forces than will be occurring when the system is in use. It would be extremely unfortunate should the system fail during construction or maintenance, so a larger force was applied for this calculation. In addition, a factor of safety of 6 was applied to this calculation for an added precaution. The result was a shaft size of 0.2699 inches, which is a little over 1/4 inches. The shaft size chosen from this calculation was 3/8 inches due to commercial availability.

The values used to size the lower shaft are the same besides the applied force. The value used for V was 225 lbs. and the factor of safety chosen for this calculation was 6. These values were chosen because the hinge will be bearing the weight of the system, so most of the load the lower pin will be experiencing is due to the thrust from the propeller, which is 134 lbs. The diameter value calculated was 0.18106 inches, which is a little less than 3/16 inches, so a 3/16-inch rod was selected.

Once the diameters of the shafts were calculated, the lengths had to be chosen. The length for both the hinge rod and the locating rod needed to be long enough to enclose a locking collar on each end with a little extra room on the end so the collars will be easier install on the system. The chosen length of the hinge shaft is 8 inches, and the chosen length of the locating rod, or pin, was 10 inches.

Once the shafts were sized, the next thing was to calculate the angle the system will need to be, based on the expected thrust/speed of the boat and the weight of the skipper. A Flow Simulation was conducted for these calculations.

The first step before this simulation can be done was to calculate the volume being submerged, which also tells the waterline of the boat with all components using force calculations. Equation 5T below was used to determine the volume submerged, with Equation 6T plugged into Equation 5T simplified into Equation 7T to solve for the volume.

$$W_{components} = F_B \tag{5T}$$

$$F_B = \rho V g \tag{6T}$$

$$V = \frac{W_{components}}{\rho g} \tag{7T}$$

In the equations above, F_B is the buoyant force, the force experienced on the system by the water. The $W_{components}$ is the total weight of all components on the hull, including the skipper. These values were summed together because the forces are all acting in the same direction. Next, constants were plugged in. First, the value of ρ, which is the density of the fluid being displaced. In this case since the fluid is water, the value used was 62.4 lbs./ft^3. The value of g, which is the acceleration due to gravity, was used as 32.2 ft/s^2. The volume displaced for a skipper of 150 lbs. was calculated to be 0.2538 ft^3 or 438.605 in^3, which corresponds to a water line of 10.835 inches below the top of the hull. The submerged volume when there is a 180 lb. skipper was 0.2688 ft^3 or 464.405 in^3, which corresponds to a water line of 10.813 inches below the top of the hull. The waterlines were found by cutting the top of the hull SolidWorks model until the desired volume was left.

Once the volume was found, the hull was cut until the only part left was the volume which would be submerged. Then the SolidWorks flow simulation was conducted for each volume at four different speeds: 6 ft/s, 9 ft/s, 12 ft/s, and 15 ft/s, which were applied as if the water were going along the length of the hull, which is how it will be during competition. The fluid selected was water, and the boundary, which is the area that the simulation will evaluate, was only the enclosed volume of the submerged hull. The next step is to select which values are desired to be recorded from the analysis. For this project the following were selected: average velocity, average force of friction, minimum static pressure, and maximum static pressure. This data was compiled, and then plotted to solve for the C_{fd} value, which is the coefficient of friction.

To plot the C_{fd} value, Equation 8T [8] is needed, shown below. For this application, both the friction due to the surface finish and the friction due to pressure was accounted for and added, since the values have an additive property [8].

$$C_{fd} = \frac{F_D}{0.5 \rho V^2 A} \tag{8T}$$

Table 16 in Appendix D has all the necessary data for the friction calculation. The F_D value used for calculating the C_{fd} due to the pressure was found by multiplying the change in pressure by the surface area of the submerged portion of the hull. To solve for these values, first the F_D values from the simulation were compiled and plotted on the y-axis and then the values in the denominator were multiplied and plotted on the x-axis of the graphs in Appendix D in Figure 103 and Figure 104. Once each respective graph was plotted, the slope of the lines is the C_{fd} value which is used later in the MATLAB code to help calculate the angle necessary to set the system at. The accuracy of this simulation data is unknown since these values are typically taken experimentally when dealing with boats. Since the hull is not complete, these are the closest values available for this calculation through simulation, but for actual competition, the C_{fd} value will need to be updated and all other values (weight, location) based on the hull will need to be updated for maximum accuracy.

The MATLAB code, shown below in Appendix F, is a way to determine which angle the system will need to be set at based on the weight of the skipper and the approximate speed the boat will be travelling, which is determined by the thrust provided by the motor. To use the code, simply input the power the motor is drawing from the batteries and the weight of the skipper. For output angles that are in between the set angles of the trim adjuster, it is best to round down to prevent the hull from angling down. This happens when the angle is larger than the system needs at the specified conditions and is worse than the boat angling up.

The force diagram used to setup the MATLAB code is shown in Appendix D in Figure 105. The weight of the components, apart from the weight of the drive train/back-end propulsion system, was combined, which is the W_{hull}. Buoyant force was taken as F_b, and the friction force was F_f.

Once the design was finished, the next step was to do a SolidWorks Simulation to ensure the requirements were met. Figure 63 below shows the applied forces, and their direction, for the

simulation which were applied to each half of the trim adjuster. Only one half of the system is shown for simplicity since both halves were analyzed with the same forces at the same geometry.
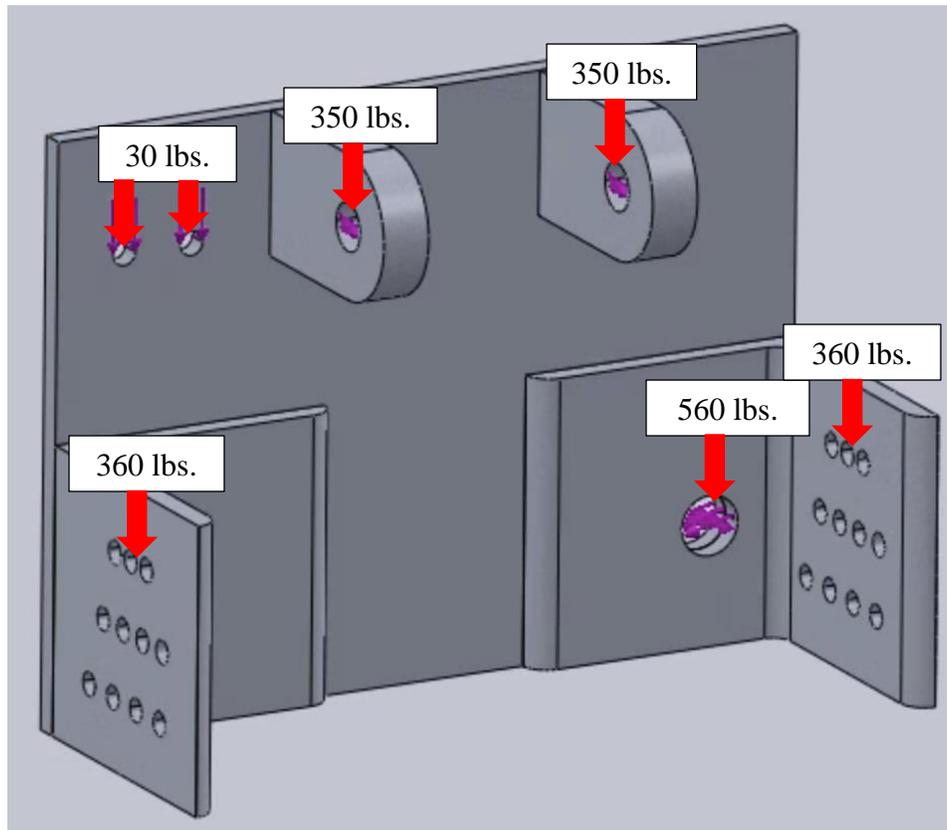
Figure 63: Loading of Trim Adjuster for FEA.

The forces applied were larger than necessary for an added factor of safety. Overall, the forces were at least doubled, if not more. Table 14 in Appendix D shows the minimum and maximum displacements of each piece, in addition to the maximum and minimum stresses. When comparing these values to what was allowed in the requirements, this system displays a minimum displacement of less than 1/16 inch (0.0625 inch). Appendix D also has the FEA stress and displacement results if more information is desired. For further information of the bearing connection side of the trim adjuster, see Figure 84 and Figure 85; and for information on the hull mounted side of the trim adjuster, see Figure 87 and Figure 88.

5.5.3 Propulsion System Connection

The next system that was designed was how the motor and drive train connect to the hull, also called the propulsion system connection. This is comprised of the shaft housing, motor mount, and bearing. The first thing that was decided upon was how the system would turn once connected to the front-end steering, which was designed by another member of the USI Solar Splash Team. Looking at the design from USI's 2021 competition, described above in 5.3 Literature Review, it

was too large of a system to attach to the trim adjuster and the new hull, so a new turning system needed to be designed.

Before anything could be designed for this mechanism, the first step was figuring out how the steering mechanism would connect while allowing the drive train and motor to freely turn without extending too far from the transom. After some brainstorming with a member of the Solar Splash team, it was decided to enclose the drive shaft within shaft housing, which will bolt to the gear box and motor mount. The shaft housing was designed to fit into a side mounting bearing and is how the system will turn for steering. In addition, since the only connection from the drive shaft to the hull is through the bearing and shaft housing, the shaft housing must support the weight of the motor, entire drive train and propeller. It also must keep the couplers in the drive train from supporting any axial loads since they are not designed to support any weight.

Once the general idea was decided, the next step was to design the shaft housing. Initially, the shaft housing was going to be the same diameter and thickness for the entire length of it, but after realizing how the drive train will have to be assembled, it was redesigned to be telescoping. In addition, it was designed to reuse one of the bearings from the 2021 boat. One of the smallest sizes the shaft housing can be while still allowing the drive shaft to freely rotate is 1 ½ inches, which is the diameter of the 2021 bearing. In addition to the bearing being the right size, it was previously calculated to support the weight of the propulsion system, which is slightly larger than the design of the propulsion system for 2022, so the bearings will work for this design [29]. Below, in Figure 64, an image of the entire system, exploded and together, are shown side by side.
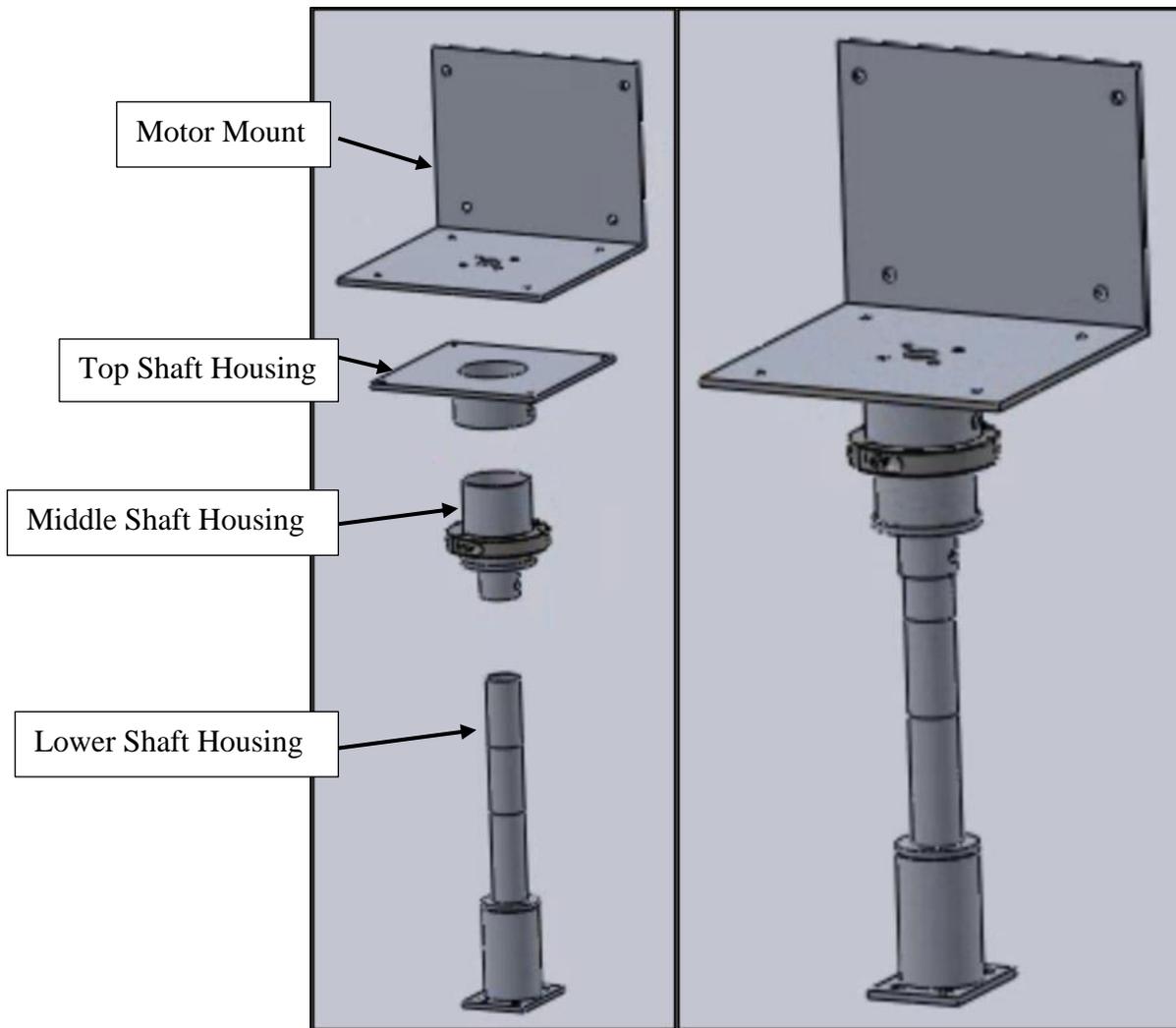
Figure 64: Shaft Housing Assembly. The image on the left is the exploded view, while on the right is the unexploded view.

This design, as stated above, is telescoping. The shaft housing pieces are vertically fixed with locking collars and are kept from rotating with set screws. The bottom shaft housing bolts to the right-angle gear box and has the same hole pattern as the top of the gearbox. In addition, slotted holes were used that are slightly larger than those of the gear box, in case there is a slight discrepancy during manufacturing. Though the holes are slightly slotted, washers can be added to allow for a better fit.

The shaft housing has a few purposes. First, regarding OSHA standards, all couplers must be enclosed [27]. Second, as stated in the drive train section, the couplers are not designed to support any axial loads, so the shaft housing must support the weight of the motor and the components of the drive train. Third, shaft housing is needed to connect the drive train to the hull.

The shaft housing is comprised of three parts, the bottom, which bolts to the gear box, the top, which bolts into the motor mount, and the middle, which connects the top and bottom sections.

Multiple sizes for the tubes making up the shaft housing were used to allow the system to be telescoping, which is necessary for the assembly of the drive train and shaft housing. In addition, to help keep weight down, the smallest side-mounting bearing that could work with the drive train was used, which had a diameter of 1 1/2-inches. This size of bearing was used in the 2021 boat, only weighs 10 lbs., and will support the weight of the propulsion system [29].

5.5.3.1 Lower Shaft Housing

Two different sized tubes are needed in the design for the lower shaft housing. This is because the drive shaft has a diameter about half the size of the couplers chosen for the drive train. To reuse the bearing from a previous competition, the outer diameter of the tube enclosing the drive shaft must be 1 1/2-inches, and the thickness was selected to be 1/8-inch. The thickness was selected due to commercial availability of the tubes with the necessary outer diameter of the smaller tube and the inner diameter of the larger tube.

The larger tube must have a big enough inner diameter for the coupler to freely rotate without any interference, while having a small enough outer diameter to allow for a bolt/nut to be screwed on, which is how the lower housing connects to the gear box.

To connect the tubes, and to be able to connect this lower shaft housing to the gear box, two plates were added. One plate is welded to one side of the largest tube. This has the hole pattern of the gear box, which is how the housing is connected to the drive train (bolted connection). The other plate gets welded to the other side of the larger tube. After those are connected, the last step is to weld the smaller of the tubes to the plate that does not bolt to the gear box. The fabrication instructions/notes of this piece are shown in Appendix D. The thickness of the plates was kept as 1/4-inch to have uniformity with the hull mount and the trim adjuster.

After the sizes were chosen, a SolidWorks FEA (Finite Element Analysis) simulation was conducted, with a horizontal loading acting the plate bolting into the gear box, in the direction of the thrust force that the propeller is making from the motor. The magnitude of the force was taken to be 200 lbs., this is larger than the maximum thrust the motor is supposed to produce, which should be a maximum of 134 lbs., but extra force was applied for an added factor of safety in case the system is being worked on. Next a force of 200 lbs. was placed where the top of the top locking collar will be located on the system, since that is how the load of the motor and drive train will be supported. The applied force was much larger than the weight of the motor, which is 55 lbs., to account for any changes needing to be made at competition if the system is not being removed from the hull. The last force being applied to this system is 100 lbs. The only vertical force this plate should have to experience is the weight of the gear box, which weighs less than 10 lbs., but again, should any changes need to be made on the system without the drive train being disassembled, this is to check that those forces will be supported.

The next step was fixing the location of the lower shaft housing. Since it is being locked in place in a bearing with locking collars, split lines were placed on the system where the bearing will be located, and that is where the system was fixed for the purposes of this simulation. Figure 65 below shows the location and direction of the forces applied.
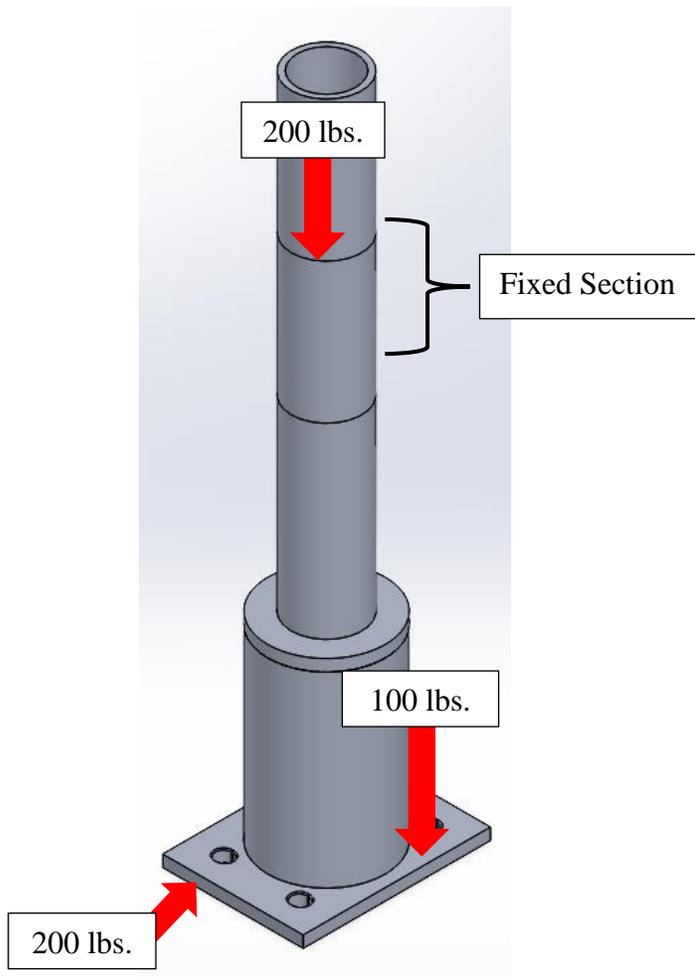
Figure 65: Lower Shaft Housing FEA Setup

The entire lower shaft housing is welded connection, with the tubes connecting via a circular plate, and on the other side of the largest tube, the plate with the hole pattern matching that of the gear box, described above, is welded. This is how and where the shaft housing connects to the gear box, with bolts going through the holes. For the size and material of the set screw, a diameter of 3/8 inch was selected, and to eliminate the worry of bolt shear, grade-8 bolts will be used for this application. Appendix D has the FEA result plots for both displacement and stress if more information is desired. Table 14 in Appendix D shows the final displacements and stresses of all systems analyzed for the shaft housing, and proves the system shows minimum displacement.

The side mounted bearing goes around the small portion of the lower shaft housing, and is located with locking collars, as stated before. The collars chosen for this application are hinged for ease of assembly, that way, when the collars are installed, only one screw needs to be tightened. The alternative, a two-piece locking collar, would be more difficult to install if the amount of people assembling the system is limited. Specific details on the actual dimensions of the bottom shaft housing are shown in the manufacturing instructions, shown in Appendix D, Figure 92. In

addition, Figure 102 in Appendix D locates the bearing and shaft housing to the exact dimensions and has more details on the final assembly. Figure 65 above does not show the collars used in this design that locate the bottom shaft housing, and that is because the bearing is not shown, just the housing.

5.5.3.2 Middle Shaft Housing

Next, is the middle shaft housing, shown below in Figure 66. This piece encloses the top coupler and the top of the drive shaft. In addition, it connects the bottom shaft housing to the top shaft housing. The middle shaft housing is comprised of two different sized tubes, the smaller of the two is just large enough to go around the upper portion of the lower shaft housing, and the larger is wide enough to enclose the top coupler while allowing it to freely rotate. As stated before, this piece rests on a locking collar, and is further fixed with two set screws to prevent this part from twisting/rotating.

Figure 66: Middle Shaft Housing FEA Setup

The inner diameter of the smaller tube on the middle shaft housing needs to be 1 1/2-inches for a snug fit around the lower shaft housing. The thickness selected for that portion of the shaft housing was 1/8-inch due to commercial availability and the fact that the drive shaft needs to be able to freely rotate. This was also chosen due to commercial availability and checked with

SolidWorks FEA Simulation to ensure the desired minimum displacement was achieved. More information on that aspect of this project will be shown later.

The larger tube was selected with the inner diameter of the tube used in the top shaft housing. Since tubes are only commercially offered in specific diameters and thicknesses, it was necessary to make sure the inner diameter of the tube chosen for the top shaft housing was an option for the outer diameter of a tube that had a large enough inner diameter to allow the top coupler to freely rotate. The selected size for this piece was 1 3/4-inch outer diameter with a thickness of 1/8-inch.

To join the two tubes, a 1/4-inch plate was selected. The tubes will be welded to the plate, one on each side. The smaller of the two shafts has two 3/8-inch holes to be used for setscrews once the system is constructed. Manufacturing instructions with more details are shown in Appendix D. The hole diameter for the setscrews was determined by looking at Table 7-4 in the *Shigley's Mechanical Engineering Design* textbook [24]. More details are shown in Appendix D in Figure 95, which are the fabrication instructions of the middle shaft housing. For the size and material of the set screw, a diameter of 3/8 inch was selected, and to eliminate the worry of bolt shear, grade-8 bolts will be used for this application.

A FEA study was conducted in SolidWorks to ensure the design meets the requirements for this project. The bottom of the smaller tube is fixed since it will be resting on the locking collar, and the only force applied was a load of 200 lbs. to the top of the system, which is the weight of the motor, and 145 lbs. more to account for someone working on or leaning on the system. Figure 66 shows how the middle shaft housing was loaded and fixed for the simulation.

The images of the results of this FEA study will be shown in Appendix D in Figure 93 and Figure 94 with the maximum and minimum displacement and stresses resulting in the applied forces of the study. Table 14 in Appendix D has a summary of the results of the entire shaft housing.

5.5.3.3 Top Shaft Housing

Fitting around the middle shaft housing is the top shaft housing. This piece is comprised of one tube and a flat plate. The purpose of the tube is to fit around the middle shaft housing and add height to meet the middle shaft housing to the motor mount. The purpose of the flat plate is to be able to mount the upper shaft housing to the motor mount. This piece locks in place vertically with a locking collar and further with two set screws.

The flat plate needs to bolt to the motor mount, so the hole pattern must be such that the plate can be easily bolted with the motor in place already. This hole pattern may need to be adjusted once the motor mount is built and the motor is installed. For the hole pattern, the hole diameter selected was 1/4 in, and the plate is a 1/4-inch square. If the holes need to be moved away or toward the tube, Equation 9T below is the formula specifying how far away the bolts need to be from the edge of a plate.

$$Distance\ from\ the\ edge = 1.5 * d \hspace{3cm} \text{(9T)}$$

In the equation above, d is the bolt hole diameter. For adjusting the distance from the edge, it should be the same from each edge, assuming the motor and motor mount allow that spatially.

For selecting the size of the tube, a couple factors were taken into consideration. The first thing, as discussed above in the middle shaft housing section, is finding a size that has an inner diameter that fits the requirements of the middle shaft housing. The second thing was the holes around the motor shaft. With how the drive train and shaft housing need to be assembled simultaneously—those directions shown in Appendix E—the plate must be flush with the motor mount. In addition, the tube must be large enough for the necessary bolts that attach the motor to the motor mount to be screwed with some clearance in case the top shaft housing is attached to the motor mount and those bolts needed tightening, a screwdriver or socket wrench can fit in the extra space. The dimension that was critical for the tube was the inner diameter, and following that, the only thickness commercially available that fit the necessary dimension was 1/4-inch thick.

A SolidWorks FEA simulation was conducted on this piece to ensure it met the requirements. To ensure it met the requirements, Figure 67 shows the setup of the simulation. The bottom of the tube is fixed, and the top of the plate has a vertical downward force of 200 lbs. This is to simulate the force of the motor with a small factor of safety for the event of someone applying force from tools, or if someone happens to lean against the propulsion system.

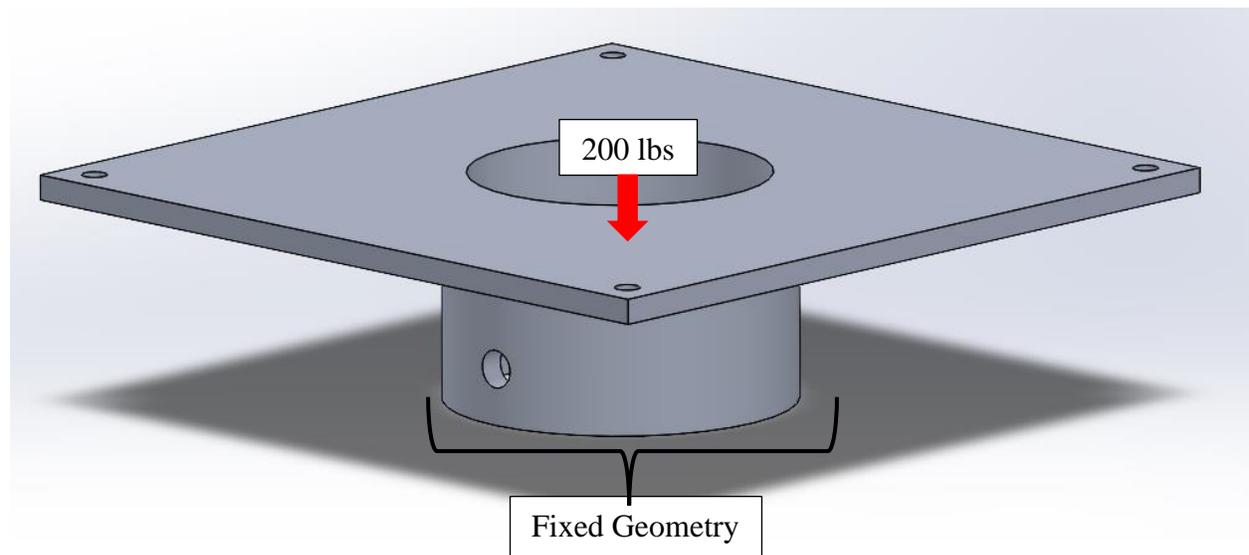*Figure 67: Top Shaft Housing FEA Setup*

Based on the displacement and stress analysis simulation shown below in Figure 96 and Figure 97 in Appendix D, the system meets the requirement of minimal (less than 1/16-inch) displacement. More details about dimensions and fabrication of the top shaft housing are shown in Appendix D in Figure 98. A summary of the FEA results is shown below in Table 14 of Appendix D.

5.5.3.4 Motor Mount

The last step was designing the motor mount. The purpose of this piece is to locate the motor and connect the motor to the shaft housing. This piece is comprised of five flat plates; two larger ones that the motor and shaft housing bolt into, and three small plates that the front-end steering connects to. The front-end steering is still in progress by a student in the 2022 Solar Splash team, so that portion of the motor mount is subject to change.

The motor mount was designed with the same hole pattern that the motor has, centered around the hole for the motor shaft. In addition, there are slotted holes on the bottom of the mount around where the motor shaft is. This is because the bolt pattern is not detailed on the motor detail sheet provided by the company [30]. To combat the missing details of the motor, the slotted holes were necessary for the best mounting possibilities. More details and dimensions are shown below in Figure 101 of Appendix D [30].

The first step in this design was to choose a plate thickness. For uniformity across this system, 1/4-inch plates were chosen. Next the connection from the motor to the motor mount was designed. This was simple as the motor's dimensions were provided with the motor. There are four bolt holes, which screw the bottom of the motor to the motor mount, and a larger hole for the 19 mm motor shaft to freely rotate. For the bolt holes that screw into the motor itself, slotted holes were used, since the hole pattern is not in line parallel or perpendicular to the flat side of the motor. After the holes were added for the motor, four other holes, 1/4-inch diameter, were added an equal distance from the center hole for the motor's shaft. This will allow the top shaft housing to mount to the motor mount.

After the bottom plate of the motor mount was designed, the next step was the vertical plate of the motor mount. This plate bolts to the side of the motor in four places. The bolt pattern dimensions were found with the motor information [30]. Now that the plates the motor bolts to are done, the next step is to design the connection between the motor mount and the front-end steering.

The front-end steering was designed by a member of the Solar Splash team that is not in this Senior Design group. In addition, the specific details of the connection of steering cable selected for the front-end steering were not available online, and since the part has not arrived, the connection may need to be resized.

For the steering connection, three small 1/4-inch plates were welded together in a "U" shape and welded to the bottom front of the motor mount. There is a hole in the horizontal part of the "U" shape which is where the steering connection attaches to the motor mount and bolts the front-end steering connection to the back-end steering connection.

To ensure that the design will hold up under the applied forces, a SolidWorks FEA Study was conducted. The FEA setup is shown below in Figure 68. A distributed force of 100 lbs. was applied perpendicular to the inside face of the vertical plate of the mount, where the motor will be sitting once it is installed. A vertical distributed force of 200 lbs. was applied to the top side of the horizontal plate on the motor mount. Lastly, a force of 200 lbs. was applied to the bolt hole in the "U" shape in the direction of the arrow in

Figure 68 to represent how the system will react when the back-end propulsion system is connected to the front-end steering system and the skipper is turning. The actual force that will be applied is unknown, so to be safe a larger force than should be applied is acting to represent how the bolt will behave in the system. The bottom of the largest horizontal plate, on the other side that the 200 lb. force is being applied, was fixed and the simulation was conducted. Lastly, the side of the motor mount where the top shaft housing will connect/bolt onto the mount was fixed and the simulation was conducted.
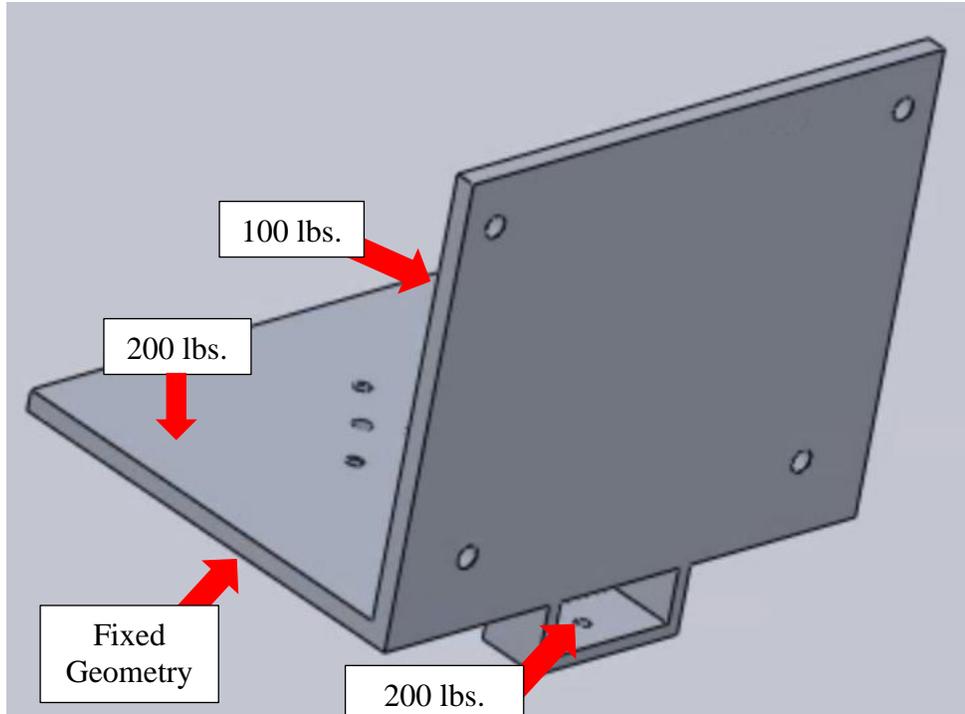
Figure 68: Motor Mount FEA Setup

Table 14 in Appendix D contains all the simulation data for each part of the shaft housing and steering propulsion connection. This shows that all parts of the propulsion system connection had minimal displacement (less than 1/16-inch displacement), and therefore meets all requirements. A more detailed view of the motor mount is shown in Figure 101 of Appendix D.

6. Management and Handoff Plan

A schedule and budget for the project have been identified to determine how the project will be successfully executed. The schedule is shown in Table 10 in Appendix A. It outlines the effective due dates and descriptions of tasks to be completed for senior design and an approximate timeline for tasks due at next year's Solar Splash competition. It can be noted that a boat construction, technical report, and video are to be complete before the 2023 competition and completion dates should allow a margin for possible time for delays and team review. The budget is shown in Table 11 in Appendix A. In this table, the purchased materials necessary for the completion of the solar panel frames and drive train are listed, and a cost is established for each line item. The trim parts have not been ordered but may be purchased by the future team after review of the trim design. A weight budget for USI's Solar Splash team is found in

Table **12** in Appendix A. Here the subsystem weights are defined, and final design weights are listed by the three mechanical subsystem teams.

In addition to the budgets of the project, failure modes and effects analysis (FMEA) tables were put together for the end user operation. The FMEA tables are used to help identify problems that may occur throughout the duration of the competition. The FMEA for the boat and mechanical subsystems may be found in Table 13 in Appendix A. A concept of operations for the boat may be found in Figure 69 in Appendix A detailing how the boat may operate in the competition,

The final constructions of the solar panel frames will be handed off to the 2023 USI Solar Splash teams for use at the next competition. The purchased drive train components will be given to the team to assemble. Manufacturing and assembly instructions for the drive train, propulsion system connection, and trim angle adjustor, located in **Appendix F**, will be given to the USI team to use in order to build and assemble the subsystems. Any disposal of these subsystems will be managed and handled by the future team.

7. Teamwork

The USI Solar Splash team is made up of students from different grade levels and engineering disciplines. There were four main sub-teams within the team: the Hull team, the Electrical team, the Structural team, and the Propulsion team. Students had the opportunity to choose which sub-team to work on at the beginning of the fall semester, in August. Any additional tasks that arose during the year were delegated to students by the team president. The USI team met weekly to discuss updates and changes to the boat, and sub-teams also met outside of the weekly meetings to work. The conflicts that arose within the USI team were mostly time related. It was difficult to schedule the weekly meetings at times that all the group members could attend. Also, some aspects of the senior design projects had to be put on hold due to the Solar Splash team not having their designs completed on time. An example would be the construction of the hull. The solar panels cannot be installed, and the trim angle adjuster design, hull mount and back-end steering cannot be finalized until the hull is completely constructed.

The senior design group was comprised of three mechanical engineering majors: Lily Bittner, Alyssa Dudas, and Melissa Kurz. Lily was part of the Structural team and designed and constructed the solar panel frames. Alyssa was part of the Propulsion team, and designed the hull mount, trim angle adjustor, motor mount, and shaft housing. Melissa was also part of the Propulsion team and designed the drive train. The senior design team had a scheduled weekly meeting but would often meet several times a week as needed throughout the semester. Lily collaborated with Jacob Mills and Mariah Fulton to make decisions when designing the solar panel frames as part of the structural system. Alyssa and Melissa collaborated on their designs, as they are interconnected as part of the propulsion system. This senior design group was very good at communication, there weren't many conflicts that occurred between the three members. One example of a conflict is trying to find meeting times during busy weeks. Another example was when one member's system had to be put on hold until the other members had finished their project

enough to do the aspects that depended on the other's projects. With patience and good communication, the conflicts were quickly resolved.

8. Conclusion

Solar Splash is an annual intercollegiate solar boating competition that demonstrates the use of solar energy to power boats. Due to scoring in previous years, the 2022 USI Solar Splash team is undertaking a complete redesign of the boat with plans to compete in the Solar Splash 2023 competition. The objective of this project was to design three of the mechanical subsystems for USI's Solar Splash boat: the solar panel frames, the trim and motor mount, and the drive train. The solar panel frames provide structure and support to the solar panels, and the frames and fasteners are fully constructed and ready for use. The trim angle adjustor allows the propulsion system to be adjusted for a variety of speeds and conditions. The motor mount, hull mount, and shaft housing provide the integral system connectivity that is necessary for the boat's success. Detailed manufacturing and assembly instructions for the trim angle adjustor and propulsion system connections will be provided to the 2023 USI team. The drive train transmits power from the motor to propeller and allows the boat to move through the water. All components of the drive train have been purchased and detailed assembly instructions will be provided to the 2023 USI team. At the conclusion of this semester, the three mechanical subsystems from this report will be handed off to USI's 2023 Solar Splash team, for success at the 2023 competition.

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Appendix A

This appendix includes additional information that is referenced in the report.

Table 10: Schedule of Project

Task	Due Date
Fall Semester	
First Solar Splash meeting to discuss needs for the 2022 team.	08/30/2021
Solar Splash meeting to review competitions rules and project requirements.	09/13/2021
Establishment of a Solar Splash System Hierarchy with team.	09/27/2021
Determine three concept designs.	10/04/2021
First draft of Senior Design Proposal.	10/08/2021
Final Senior Design Proposal.	10/18/2021
Senior Design Proposal Oral Presentation.	10/25/2021
Discuss three concepts and chose a final concept design.	11/01/2021
Build SolidWorks models and start simulations.	11/08/2021
Senior Design Concept Oral Presentation.	11/15/2021
Pre-Senior Design Report.	12/08/2021
Spring Semester	
Arrange weekly meetings with Senior Design faculty adviser.	01/10/2022
First meeting with Senior Design faculty adviser.	01/24/2022
All materials ordered.	03/04/2021
Critical Design Review.	03/31/2022
Design Presentation Reviews complete.	04/04/2022
Draft of Senior Design Report due to adviser.	04/08/2022
Technical Report complete. Team review to correct as needed.	04/15/2022
Final Presentation PowerPoints.	04/22/2022
Senior Design Poster Session.	04/28/2022
Final Report due to adviser.	04/29/2022
Technical Report due to Solar Splash.	05/02/2022
Final Report submitted to SOAR.	05/06/2022

Table 11: Project Cost Budget

System	Item Number (if applicable)	Item Description	Quantity (units)	Manufacturer	Unit Cost ($)	Total Cost ($)
Solar Panel Frames	-	Aluminum Flat Bar 1/8" x 1-1/4" x 12'	4	West Metals	13.00	52.00
	-	Aluminum Square Tube 1" x 1/8" x 3.5'	4	West Metals	20.00	80.00
	2313N11	Corner Machine Bracket 1-1/4" x 1" x 1"	8	McMasterCarr	7.24	57.92
	92385A035	Locking Quick-Release Pin 1/4" Diameter, 1.25" Long	16	McMasterCarr	33.54	536.64
	90592A012	Hex Nut M5 (100-pack)	1	McMasterCarr	2.36	2.36
	91290A232	Socket Head Screw M5 x 0.8 mm Thread, 16 mm (100-Pack)	1	McMasterCarr	16.23	16.23
	91950A029	SAE Washers (50-Pack)	1	McMasterCarr	8.52	8.52
Trim	9246K425	6061-T6 Plate-1/4" X 12" X 48"	1	McMasterCarr	109.41	109.41
	9246K13	6061-T6 Plate-1/4" X 12" X 12"	1	McMasterCarr	35.61	35.61
	8982K61	6061-T6 L0.25x3x3 X 1'-0" LG	1	McMasterCarr	24.04	24.04
	9246K13	6061-T6 Bar-1/2" x 6" x 12"	1	McMasterCarr	11.61	11.61
	9056K38	6061-T6 Tube-1 1/2"OD x 1/8" thickness x 1'-0" LG	1	McMasterCarr	10.44	10.44
	9056K39	6061-T6 Tube-1 3/4"OD x 1/8" thickness x 6" LG	1	McMasterCarr	11.80	11.80

System	Item Number (if applicable)	Item Description	Quantity (units)	Manufacturer	Unit Cost ($)	Total Cost ($)
		Table 11: Continued				
Trim	9056K41	6061-T6 Tube-3"OD x 1/8" thickness x 6" LG	1	McMasterCarr	17.81	17.81
	9056K24	6061-T6 Tube-3 1/2"OD x 1/4" thickness x 6" LG	1	McMasterCarr	38.54	38.54
	57145K83	1 1/2" ID Locking Hinged Collar	2	McMasterCarr	25.12	50.24
	57145K91	3" ID Locking Hinged Collar	1	McMasterCarr	79.15	79.15
	6436K6	3/16" ID Two-Piece Shaft Collar	2	McMasterCarr	7.81	15.62
Drive Train	7398K221	Drive Shaft	1	McMasterCarr	37.26	37.26
	135891	MITRPAK R-121-C1-M Right Angle Gearbox	1	Lampin	467.00	467.00
	9845T61	Coupler Hub-19mm bore	2	McMasterCarr	104.30	208.60
	9845T5	Coupler Hub-20mm bore	2	McMasterCarr	82.85	165.70
	9845T24	Coupler Spider-Durometer 98A	2	McMasterCarr	32.92	65.84
Total	-	-	-	-	-	2086.72

Table 12: Weight Budget of Solar Splash Boat

System	Weight Allocation (lbs.)	Item Description	Quantity (units)	Unit Weight (lbs.)	Total Weight (lbs.)
Skipper	200	Skipper up to 200 lbs.	1	200.00	200.00
Hull	50	New Hull	1	50.00	50.00
Solar Panel Frames	50	Frame	2	3.14	6.28
		Rail	4	1.73	6.92
		Mounting Bracket	8	0`.06	0.48
		Quick Release Pin	16	0.06	0.96
		Nuts/Bolts/Washers	20	0.03	0.60
		Solar Panel	2	6.20	12.40
Trim	30	Trim Angle Adjuster Assembly	1	5.23	5.23
		Bearing	1	10.00	10.00
		Shaft Housing Assembly	1	4.00	4.00
		Motor Mount	1	4.73	4.73
		Hull Mount	1	1.40	1.40
Drive Train	25	Drive Shaft	1	8.90	8.90
		MITRPAK R-121-C1-M Right Angle Gearbox	1	8.00	8.00
		Coupler Hub- 19mm bore	2	0.27	0.54
		Coupler Hub- 20mm bore	2	0.27	0.54
		Coupler Spider- Durometer 98A	2	0.05	0.10
Motor	55	Lynch LEM 200-95 2x2	1	55.00	55.00
Back End	40	Additional subsystems including steering and propeller.	1	45.00	45.00
Front End	100	Additional subsystems including electrical.	1	100.00	100.00
Total	550	-	-	-	536.37

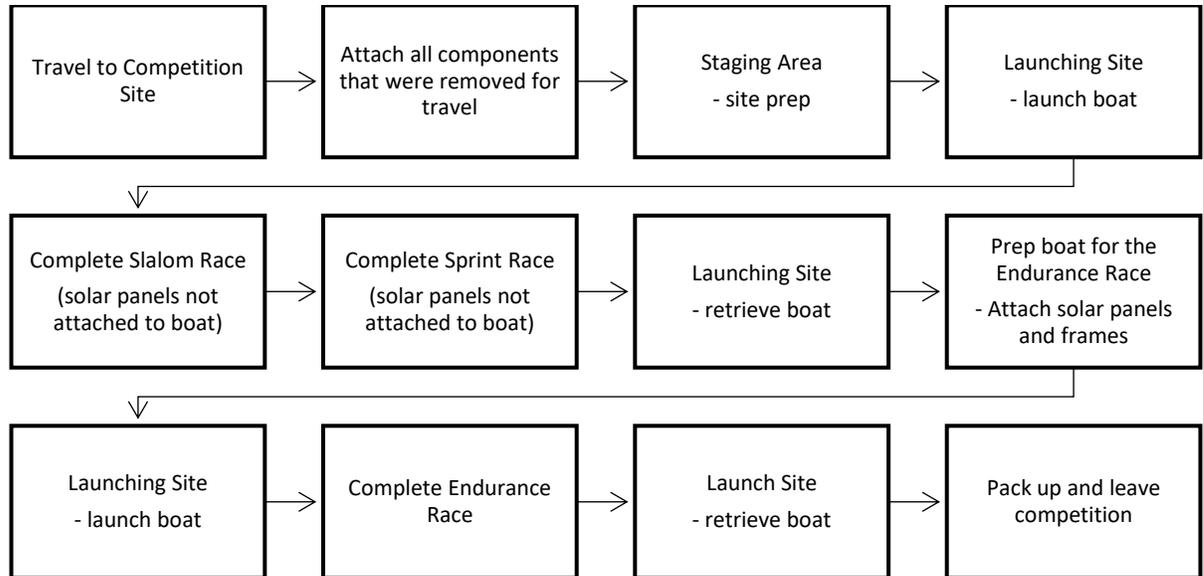Below is a concept of operations for end user boat operation at the Solar Splash competition.

Travel to Competition Site	Attach all components that were removed for travel	Staging Area - site prep	Launching Site - launch boat
Complete Slalom Race (solar panels not attached to boat)	Complete Sprint Race (solar panels not attached to boat)	Launching Site - retrieve boat	Prep boat for the Endurance Race - Attach solar panels and frames
Launching Site - launch boat	Complete Endurance Race	Launch Site - retrieve boat	Pack up and leave competition

Figure 69: Concept of Operations

The System Hierarchy for USI's 2022 boat includes structural, mechanical, and electrical subsystems. In Figure 70 below, level one items are in black, level two items are in blue, and level three items are in green.
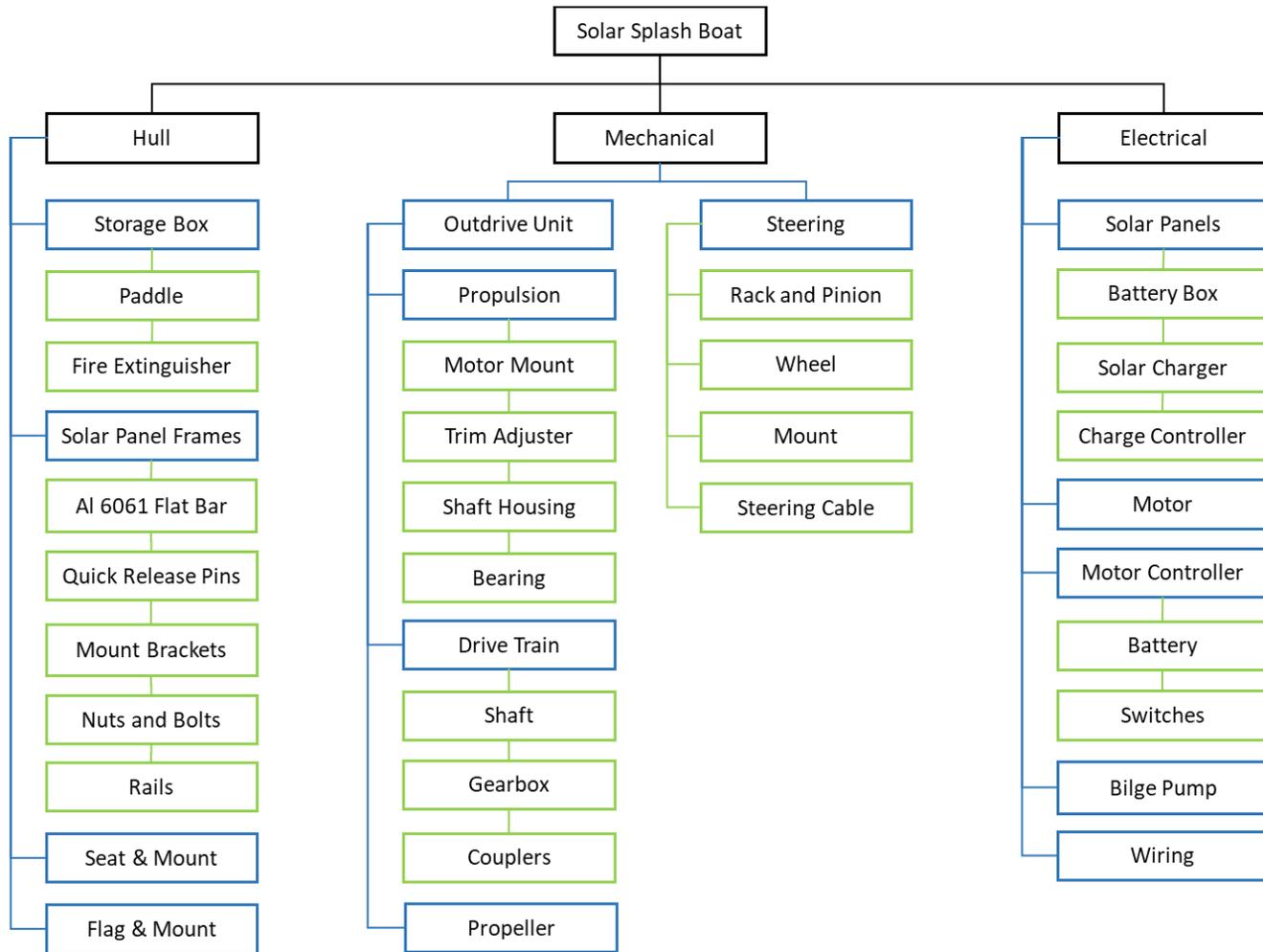
Figure 70: System Hierarchy for Solar Splash Boat

Table 13: FMEA for Competition

System	Item	Failure Modes	Cause of Failure	Possible Effects	Probability	Level of Criticality	Possible Action to Reduce Failure Rate or Effects
Solar Splash Boat	Boat	System Breakage	System Integration issues	Boat Failure	High	Very High	Have all team members assist in troubleshooting and providing input on solutions. Invite minority opinion. Prepare to fix component breakage. Prepare to purchase new components if necessary. Have plans on each subsystem failure to mitigate issues and problems.
			Damage In Use				
Solar Panel Frames	Frames	Breakage	Weld Cracking	Failure of Subsystem	Medium	Very High	For welding cracks or failures having welding machine on hand could be a consideration. Be cautious when transporting and handling the frames. Be prepared to attach the solar panels directly to the hull if failure occurs and there is no way to fix them.
			Excessing Flexure				
			Damage During Transportation				
			Damage During Handling				
	Mounts	Not Functional/ Breakage	Misalignment	Misalignment Between Boat and Mounts	Medium	High	Be prepared to adjust the mount locations if misalignment occurs. Be cautious when transporting and handling the frames. Prepare to purchase alternative components if breakage occurs at competition. Buy backups if the budget allows.
			Damage During Transportation				
			Damage During Handling				

System	Item	Failure Modes	Cause of Failure	Possible Effects	Probability	Level of Criticality	Possible Action to Reduce Failure Rate or Effects
				Table 13: Continued			
Trim	Shafts	Breakage	Shearing off	Subsystem Failure	Low	High	Be cautious when transporting and assembling. Avoid overloading the system with unnecessary external forces. Be prepared to purchase backup parts should failure occur during competition.
	Bolts	Breakage	Shearing off	Subsystem Failure	Low	High	Be cautious when transporting and assembling. Too much stress can cause the bolts to shear off. Prepare to purchase new bolts at competition if necessary.
	Welds	Breakage	Weld Cracking	Subsystem Failure	Low	Very High	Be cautious when transporting the systems. For best results, remove the entire propulsion system from the hull, so the welds aren't under any external stress before competition begins.
	Collars	Not Functional/ Breakage	Damage During Transportation	Subsystem Failure	Low	Very High	Be cautious when transporting and assembling. Be prepared to purchase backup parts should failure occur during transportation.

System	Item	Failure Modes	Cause of Failure	Possible Effects	Probability	Level of Criticality	Possible Action to Reduce Failure Rate or Effects
							Table 13: Continued

System	Item	Failure Modes	Cause of Failure	Possible Effects	Probability	Level of Criticality	Possible Action to Reduce Failure Rate or Effects
Drive Train	Drive Shaft	Misalignment	Damage During Assembly/ Transportation/ Operation	Subsystem Failure / Coupler Jaw and Spider Failure	Medium	High	Be cautious while assembling and transporting the drive train. If damage occurs during operation check shaft straightness and check couplers. Prepare to purchase additional components if failure occurs at competition.
	Coupler Hubs	Breakage	Damage During Assembly/ Transportation / Overload	Subsystem Failure	Low	High	Be cautious while assembling and transporting the drive train. Do not allow the couplers to experience axial loads. Avoid overloading from the motor. Prepare to purchase additional components if failure occurs at competition.
	Coupler Spiders	Breakage	Damage During Assembly/ Transportation / Normal Wear / Overload / Misalignment / Vibration	Low Drive Train Performance	High	Medium	Be cautious while assembling and transporting the drive train. Do not allow the couplers to experience axial loads. Avoid overloading from the motor. Regularly check the spiders for signs of wear. Prepare to purchase backups if the budget allows.

Appendix B

This appendix includes additional information for the design of the solar panel frames. These may be referenced in the report.

Calculations for the solar panel frames included static calculations and drag calculations. Results of the static calculation using the free body diagram in Figure 71 are shown below in Figure 72 and Figure 73. This free body diagram was produced by a beam simulation site SkyCiv and the result of the 2D analysis is shown in Figure 72 and Figure 73. The reaction forces at the pin connections using a basic 2D calculation are approximately 3.1 lbs. each and the maximum moment experienced is approximately 27.9 lbs.-in [18]. These values are not excessive and are reasonable for this project.
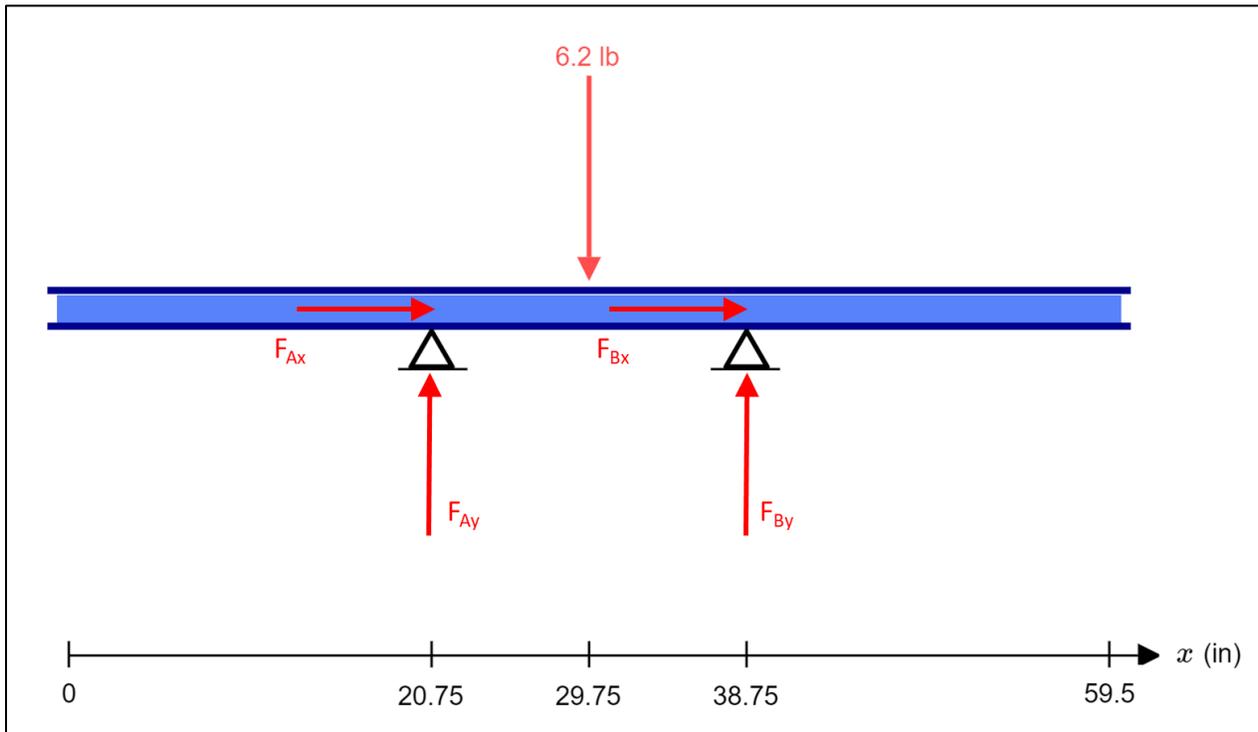
Figure 71: Free Body Diagram of Solar Panel Frames [18]
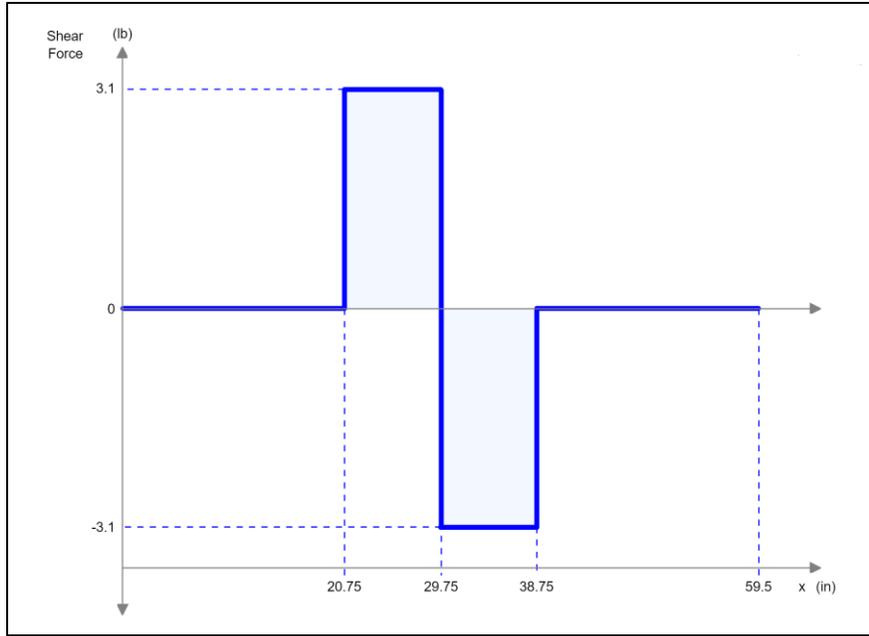
Figure 72: Shear Force Diagram for Solar Panel Frames [18]
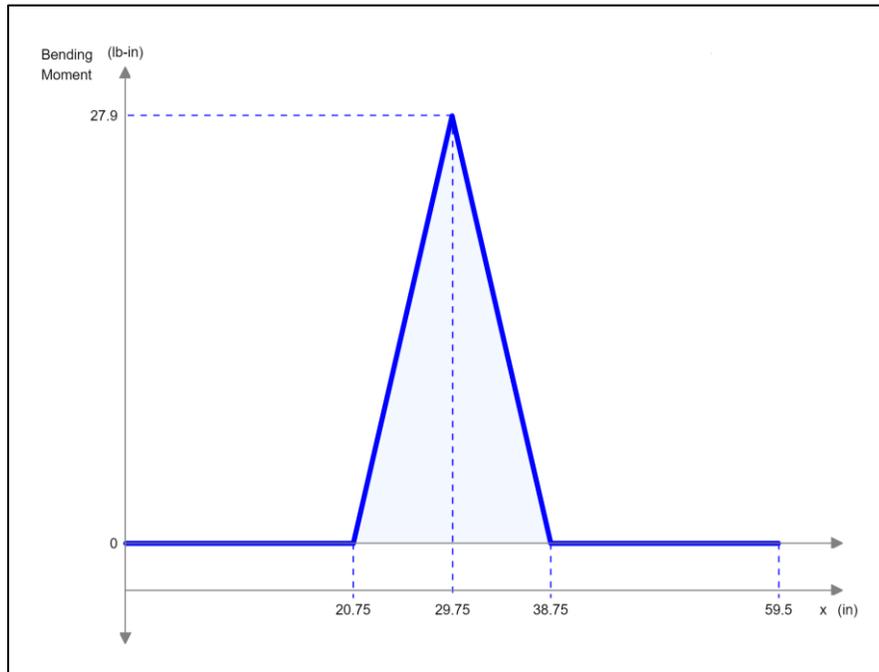
Figure 73: Bending Moment Diagram for Solar Panel Frames [18]

For drag on the solar panels and frames, the drag coefficient is equal to the friction coefficient for a parallel flat plate and can be found by calculating Reynold's number over the length of a plate, Re_L. The equation for average friction coefficient over the length of the plate may be found in Equation 3 for laminar flow when $Re_L \leq 5 \times 10^5$ *[8]*.

Laminar: $$C_{D, \text{ parallel flat plate}} = C_f = 1.33 / (Re_L)^{1/2} \tag{1}$$

Turbulent: $$C_{D, \text{ parallel flat plate}} = C_f = 0.074 / (Re_L)^{1/5} \tag{2}$$

$$Re_L = V * L / v \tag{3}$$

$$Re_L = (22 \text{ ft/s}) * (2.23 \text{ ft}) / (1.697 \text{ x } 10^{-4} \text{ ft}^2/s) \tag{4}$$

$$Re_L = 289,100 = 2.891 \text{ x } 10^5 \tag{5}$$

Laminar: $$C_{D, \text{ parallel flat plate}} = C_f = 1.33 / (2.891 \text{ x } 10^5)^{1/2} \tag{6}$$

$$C_f = 0.002474 \tag{7}$$

Reynold's number for the length of the plate can be found in Equation 4 where V represents velocity, L represents the length of the plate where flow is acting across, and v represents kinematic viscosity [8]. For this calculation, Reynold's number will be determined with a velocity of 22 ft/s to represent possible velocity conditions on the boat, a length of 2.23 ft to represent the width of a plate where flow occurs across the width, and a kinematic viscosity of 1.697 x 10^{-4} ft^2/s for air assumed at 80°F and 1 atm.

Performing this calculation confirms that solar panels in a flat plat configuration result in a negligible coefficient of friction. Drag related to the frames being attached to the hull are analyzed in simulation due to being too complex for a simple hand calculation and simulation results are discussed in the report.

The mechanical block diagram for the solar panel frames including blocks of the solar panels, solar panel frames, boat hull, mounts, and bolting is shown in Figure 74 below.
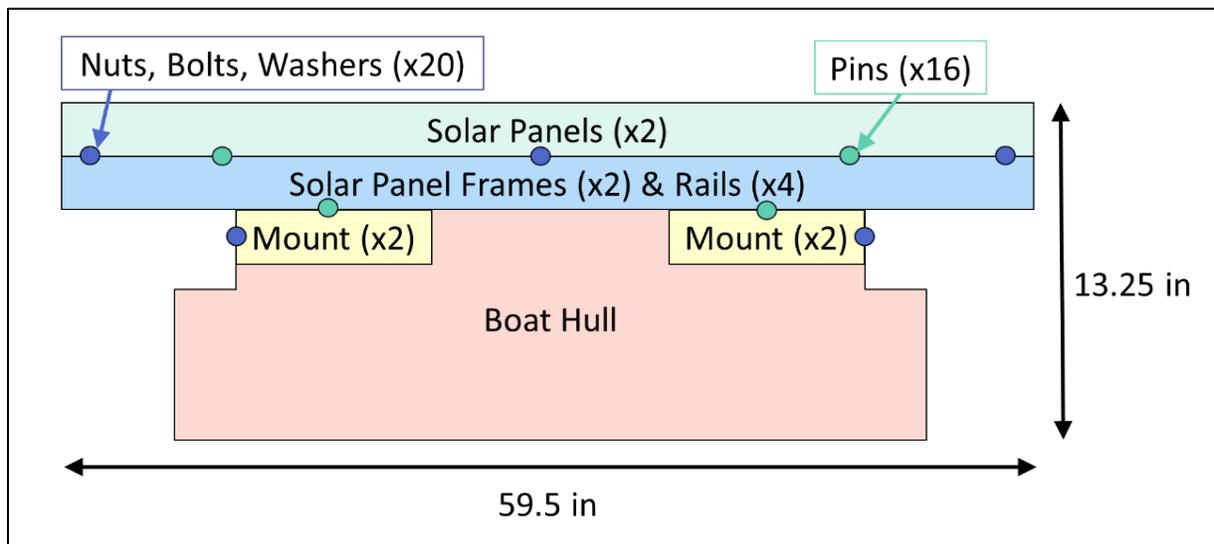
Figure 74: Mechanical Block Diagram of Solar Panel Frames

Appendix C

This appendix includes additional information for the design of the drive train. These may be referenced in the report.

Drive Shaft Design Fatigue Analysis

Instructions: Provide numbers for values in green, and input a diameter

diameter	18 mm	
r	9 mm	
I	5152.99735 mm^4	Solid Shaft
J	10305.9947 mm^4	Solid Shaft
notch radius	0.36 mm	estimate notch radius/diameter = 0.02 for an end mill keyseat
Ma	0 N-m	Alternating
Tm	56 N-m	Mean
nom bend stress	0.00 MPa	
nom shear stress	48.90 MPa	
Kt (bending)	2.14	
q	0.596292447	
Kf	1.67977339	
Kts (torsion)	3	
qs	0.661505925	
Kfs	2.323011849	
max bend stress	0.00E+00 MPa	
max shear stress	1.14E+02 MPa	
static vm stress (yield)	8.47E+01 MPa	ductile assumed
vm alt stress	0.00E+00 MPa	
vm mean stress	1.97E+02 MPa	
Sut	564 MPa	
Sy	210 MPa	
Se'	282 MPa	
Finish	Ground	

k factors

ka	0.9027	
	a	1.38 *Reads ka sheet*
	b	-0.067 *Reads ka sheet*
kb	0.9101	*Calculates kb based on size of rotating diameter*
kc	1	*kc=1 when VM equivalent stresses used*
kd	1	temperature
ke	0.868	reliability 95.00%
kf	1	
Se	201.1 MPa	

Goodman FS	2.866	
Gerber FS	#DIV/0!	
Static Yield FS	2.479	

*Note, if Gerber FS is undefined in this spreadsheet, it is equal to the Goodman FS

Figure 75: Fatigue Analysis for 18 mm diameter drive shaft

Drive Shaft Design Fatigue Analysis

Instructions: Provide numbers for values in green, and input a diameter

diameter	20	mm	
r	10	mm	
I	7853.981634	mm^4	Solid Shaft
J	15707.96327	mm^4	Solid Shaft
notch radius	0.4	mm	estimate notch radius/diameter = 0.02 for an end mill keyseat
Ma	0	N-m	Alternating
Tm	56	N-m	Mean
nom bend stress	0.00	MPa	
nom shear stress	35.65	MPa	
Kt (bending)	2.14		
q	0.60890714		
Kf	1.69415414		
Kts (torsion)	3		
qs	0.673199672		
Kfs	2.346399344		
max bend stress	0.00E+00	MPa	
max shear stress	8.37E+01	MPa	
static vm stress (yield)	6.17E+01	MPa	ductile assumed
vm alt stress	0.00E+00	MPa	
vm mean stress	1.45E+02	MPa	
Sut	564	MPa	
Sy	210	MPa	
Se'	282	MPa	
Finish	Ground		

k factors

ka	0.9027		
	a	1.38	*Reads ka sheet*
	b	-0.067	*Reads ka sheet*
kb	0.8999		*Calculates kb based on size of rotating diameter*
kc	1		*kc=1 when VM equivalent stresses used*
kd	1		temperature
ke	0.868		reliability 95.00%
kf	1		
Se	198.8	MPa	

Goodman FS	3.893	
Gerber FS	#DIV/0!	
Static Yield FS	3.401	

*Note, if Gerber FS is undefined in this spreadsheet, it is equal to the Goodman FS

Figure 76: Fatigue Analysis for 20 mm diameter drive shaft

Drive Shaft Design Fatigue Analysis

Instructions: Provide numbers for values in green, and input a diameter

diameter	22	mm	
r	11	mm	
I	11499.01451	mm^4	Solid Shaft
J	22998.02902	mm^4	Solid Shaft
notch radius	0.44	mm	estimate notch radius/diameter = 0.02 for an end mill keyseat
Ma	0	N-m	Alternating
Tm	56	N-m	Mean
nom bend stress	0.00	MPa	
nom shear stress	26.78	MPa	
Kt (bending)	2.14		
q	0.620194967		
Kf	1.707022263		
Kts (torsion)	3		
qs	0.683596103		
Kfs	2.367192206		
max bend stress	0.00E+00	MPa	
max shear stress	6.34E+01	MPa	
static vm stress (yield)	4.64E+01	MPa	ductile assumed
vm alt stress	0.00E+00	MPa	
vm mean stress	1.10E+02	MPa	
Sut	564	MPa	
Sy	210	MPa	
Se'	282	MPa	
Finish	Ground		

k factors

ka	0.9027			
		a	1.38	*Reads ka sheet*
		b	-0.067	*Reads ka sheet*
kb		0.8908	*Calculates kb based on size of rotating diameter*	
kc		1	*kc=1 when VM equivalent stresses used*	
kd		1	temperature	
ke		0.868	reliability	95.00%
kf		1		
Se	196.8	MPa		

Goodman FS	5.136
Gerber FS	#DIV/0!
Static Yield FS	4.527

*Note, if Gerber FS is undefined in this spreadsheet, it is equal to the Goodman FS

Figure 77: Fatigue Analysis for 22 mm diameter drive shaft

Appendix D

      This appendix includes additional information for the design of the trim angle adjustor and propulsion system connections including the FEA analysis followed by its fabrication instructions. These may be referenced in the report.
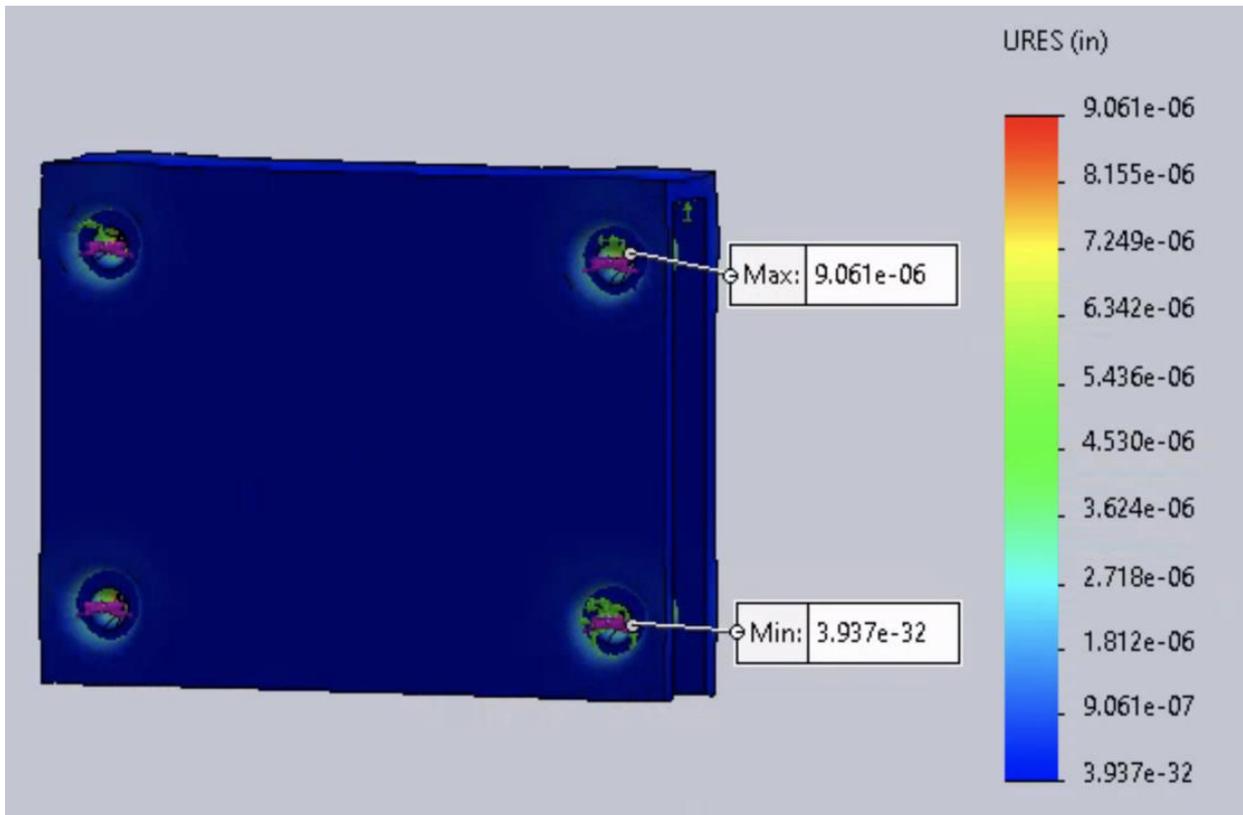
Figure 78: Maximum and Minimum Displacement Plot for 1/4-inch Plate Design.
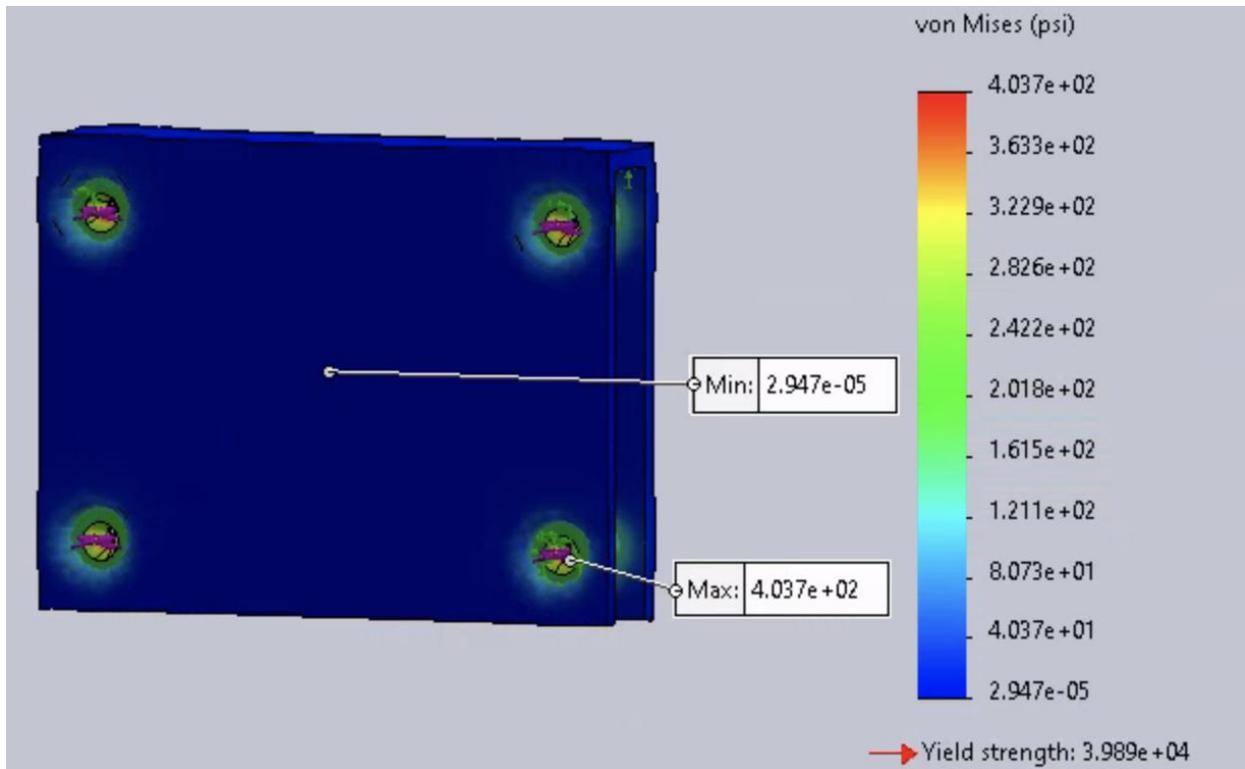
Figure 79: Maximum and Minimum von Mises Plot for 1/4-inch Plate Design.
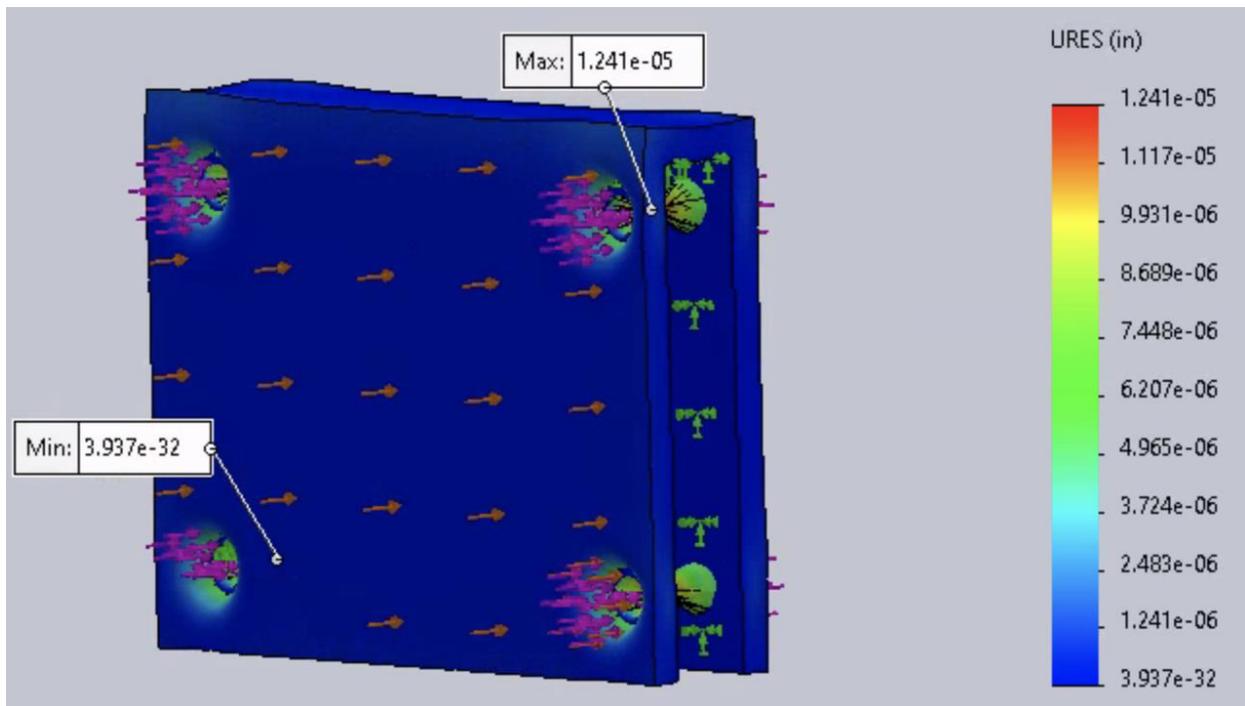
Figure 80: Maximum and Minimum Displacement Plot for 3/8-inch Plate Design.
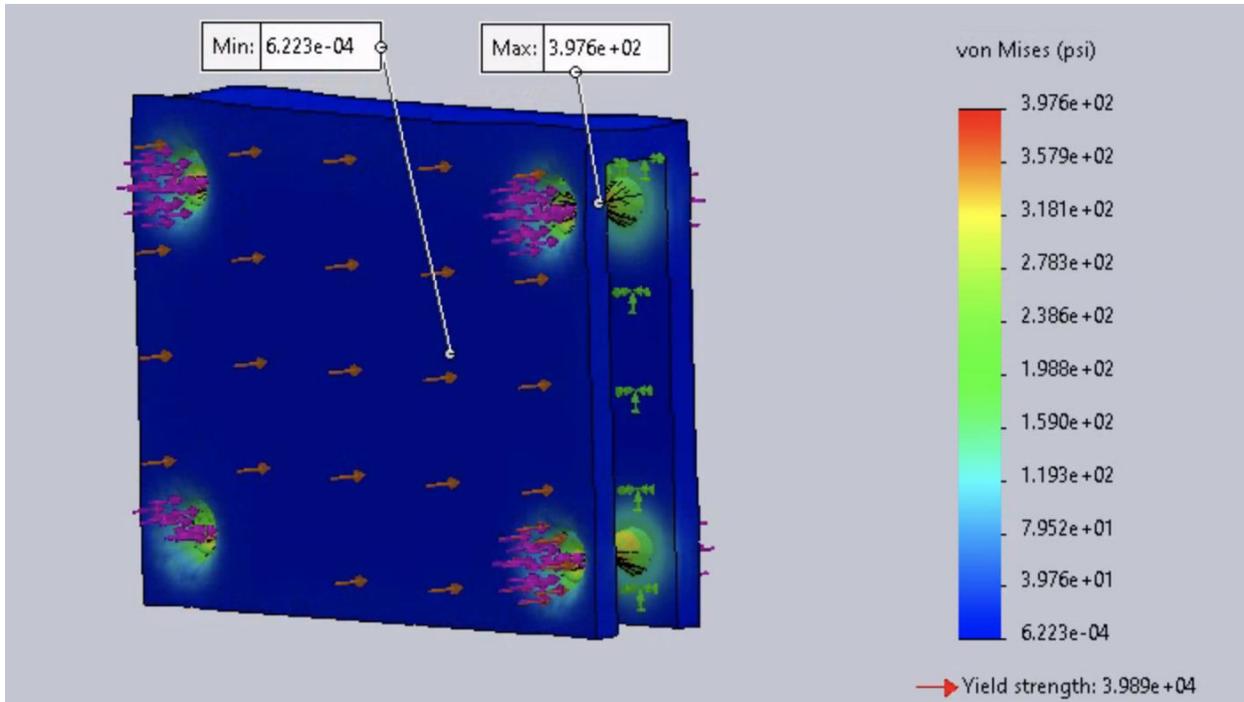
Figure 81: Maximum and Minimum von Mises Plot for 3/8-inch Plate Design.

```matlab
%% Material Properties and Dimensions
%Hull Mount: 6061 Aluminum (matweb.com 'Aluminum 6061-T6; 6061-T651')
Sy_Al = 40000; % psi, yield strength
t_mount = 1/4; % in, thickness of the hull mount

%Bolts: Grade 2
n = 2; % number of bolts
d = 1/4; % in, diameter of bolts
Ac = (1/4)*pi*d^2;  % in^2, cross-sectional area of the bolt
Sy_bolt = 57*10^3; % psi, yield strength (Table 8-9 page 444 in book)
%   Grade 2: 57kpsi, Grade 5: 92kpsi, Grade 8: 130kpsi

%Hull: Marine Plywood coated in Fiberglass (matweb.com 'Plywood')
t_hull = 1; %in, thickness of the hull
Sy_hull = 4000; %psi, tensile strength

%Applied force due to weight of the motor and drive train
F = 300*2; %lbf, with a Factor of Safety of 2

%% Factors of Safety for Shear Failure Modes
%Failure Mode: Bearing on Member (Hull)
sigma_bh = F/(n*t_hull*d);
eta_HullBearing = Sy_hull/sigma_bh;

%Failure Mode: Bolt Shear
tau = F/(n*Ac);
eta_BoltShear = 0.577*Sy_bolt/tau;
```

Figure 82: Bolt Sizing MATLAB Code for Hull Mount
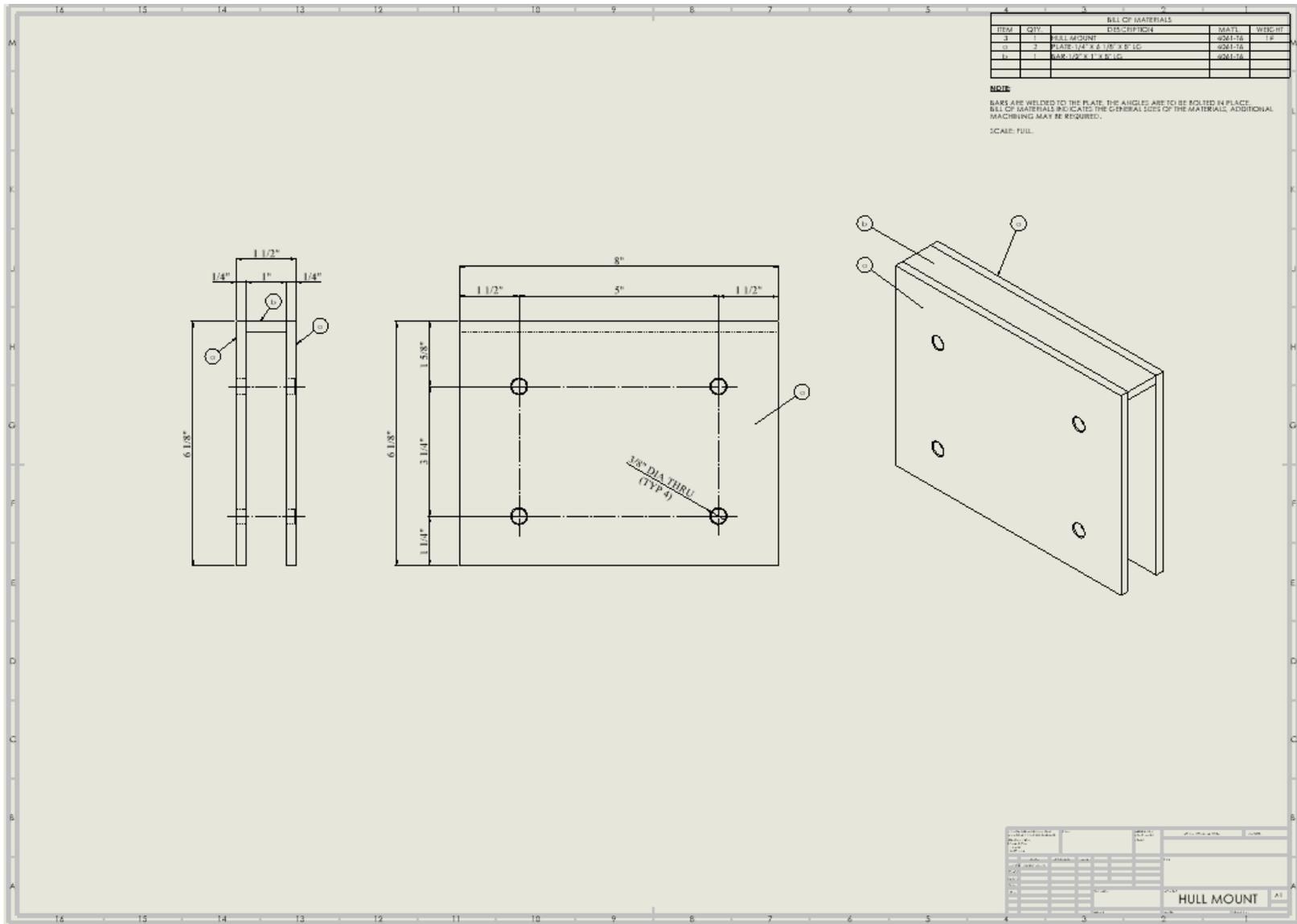
Figure 83: Fabrication Instructions for the Hull Mount
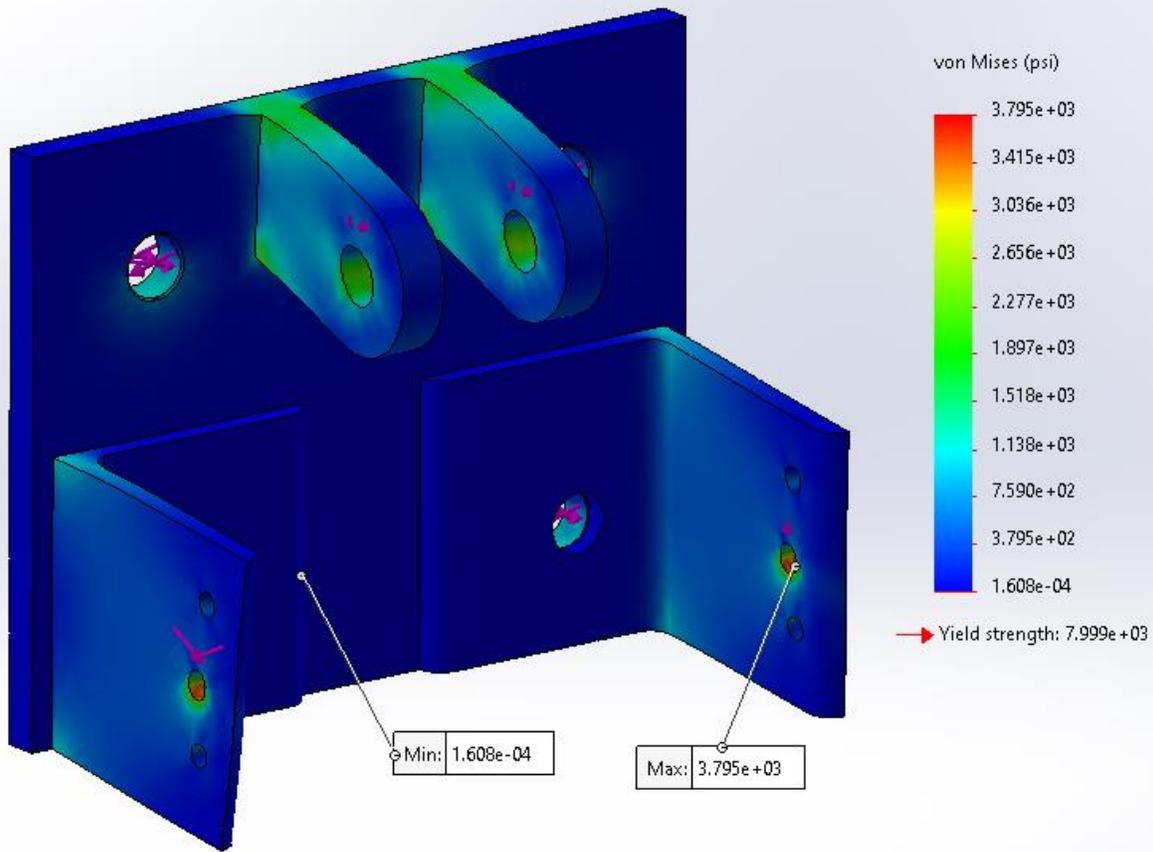
Figure 84: Trim Angle Adjuster Half That Bolts to the Hull Stress Analysis
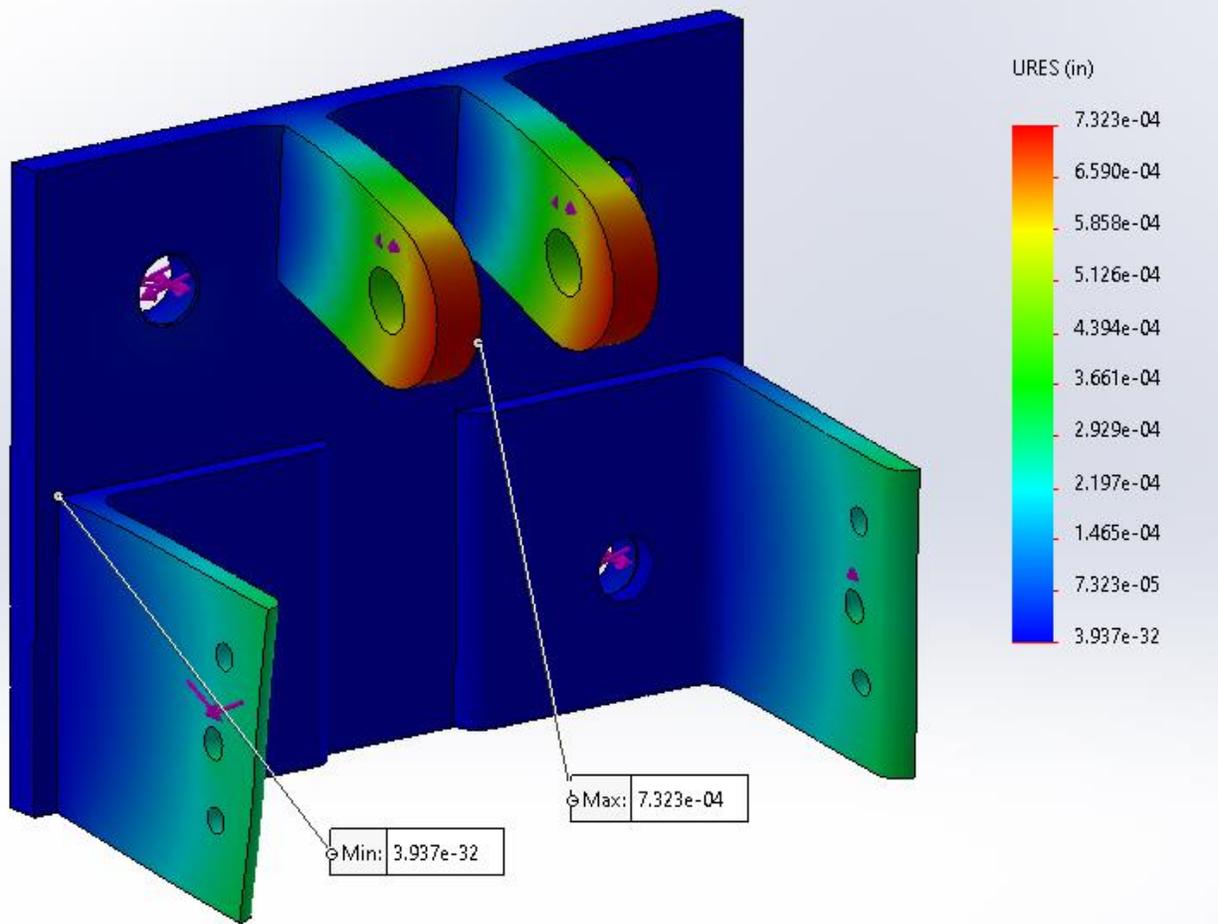
Figure 85: Trim Angle Adjuster Half That Bolts to the Hull Displacement Analysis
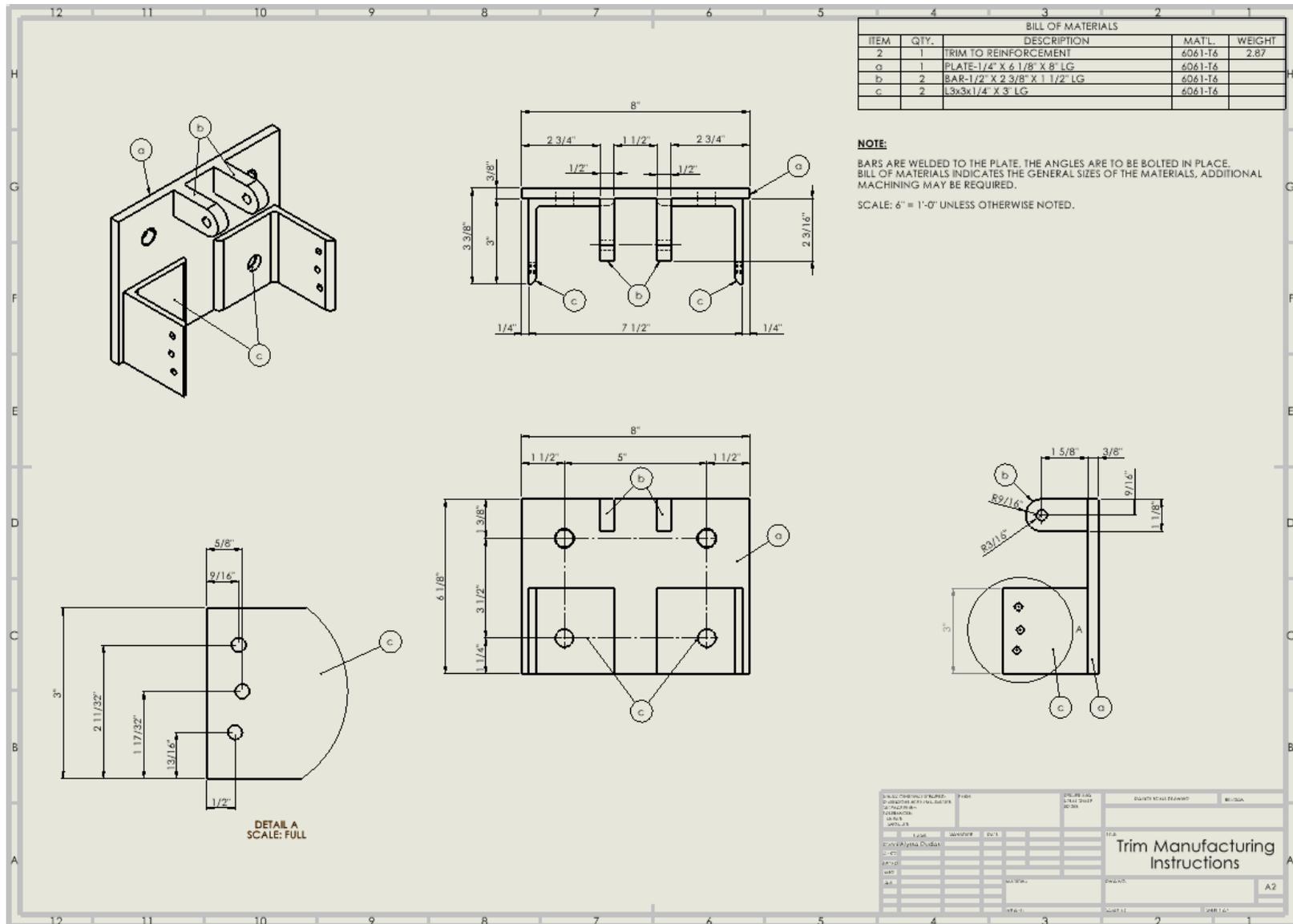
Figure 86: Fabrication Instruction Sheet for Trim Angle Adjuster That Bolts onto Hull
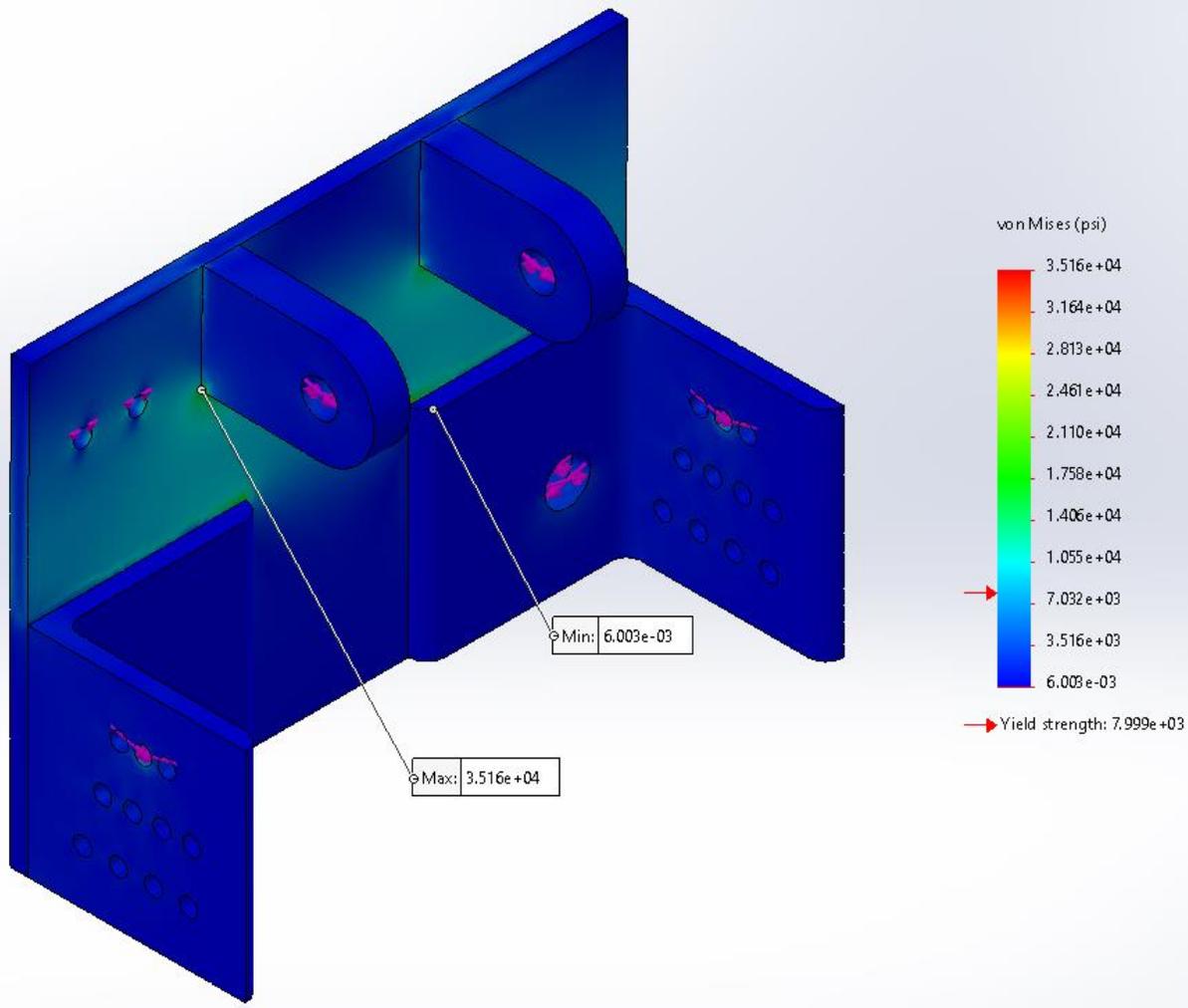
Figure 87: Trim Angle Adjuster Half That Bolts onto the Bearing Stress Analysis
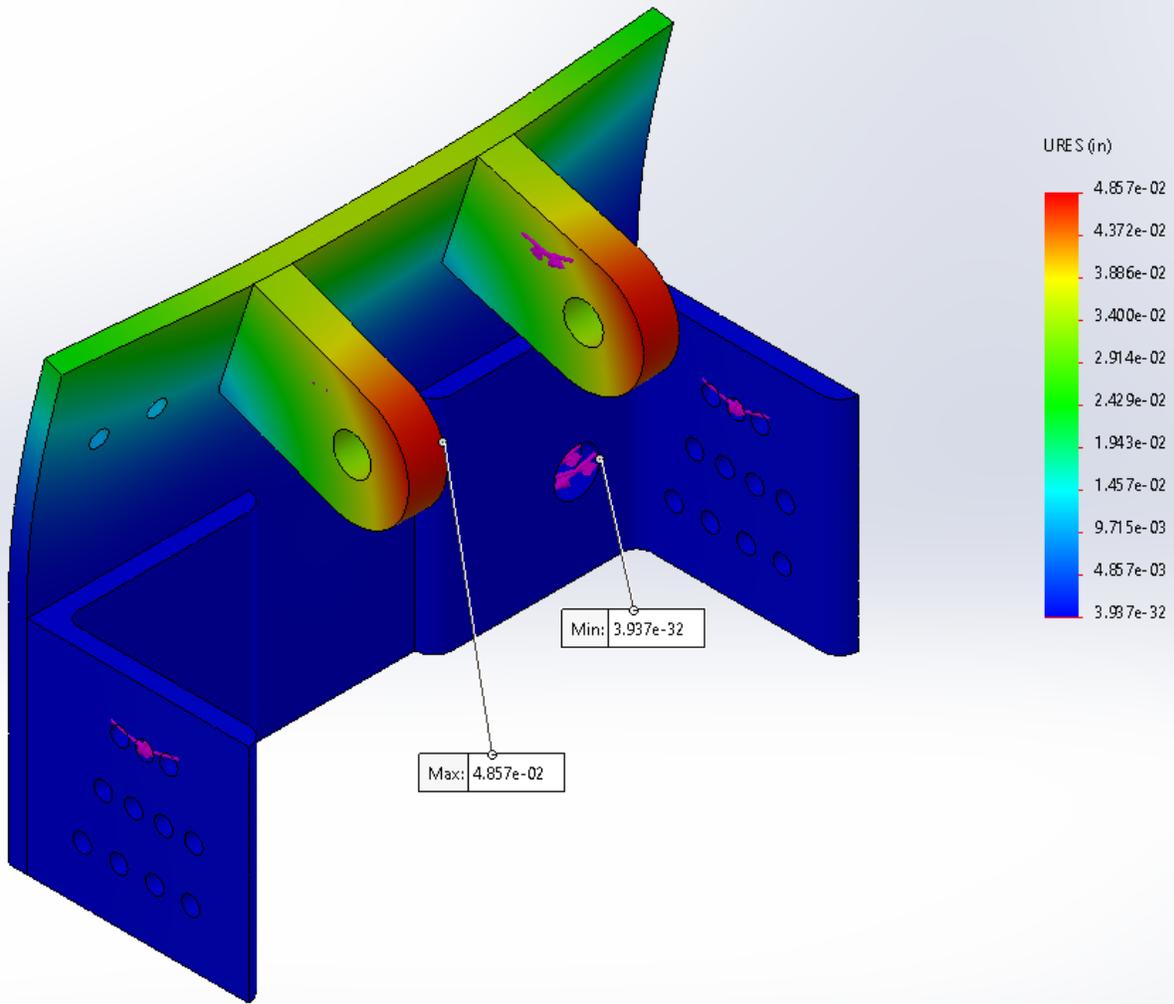
Figure 88: Trim Angle Adjuster Half That Bolts onto the Bearing Displacement Analysis
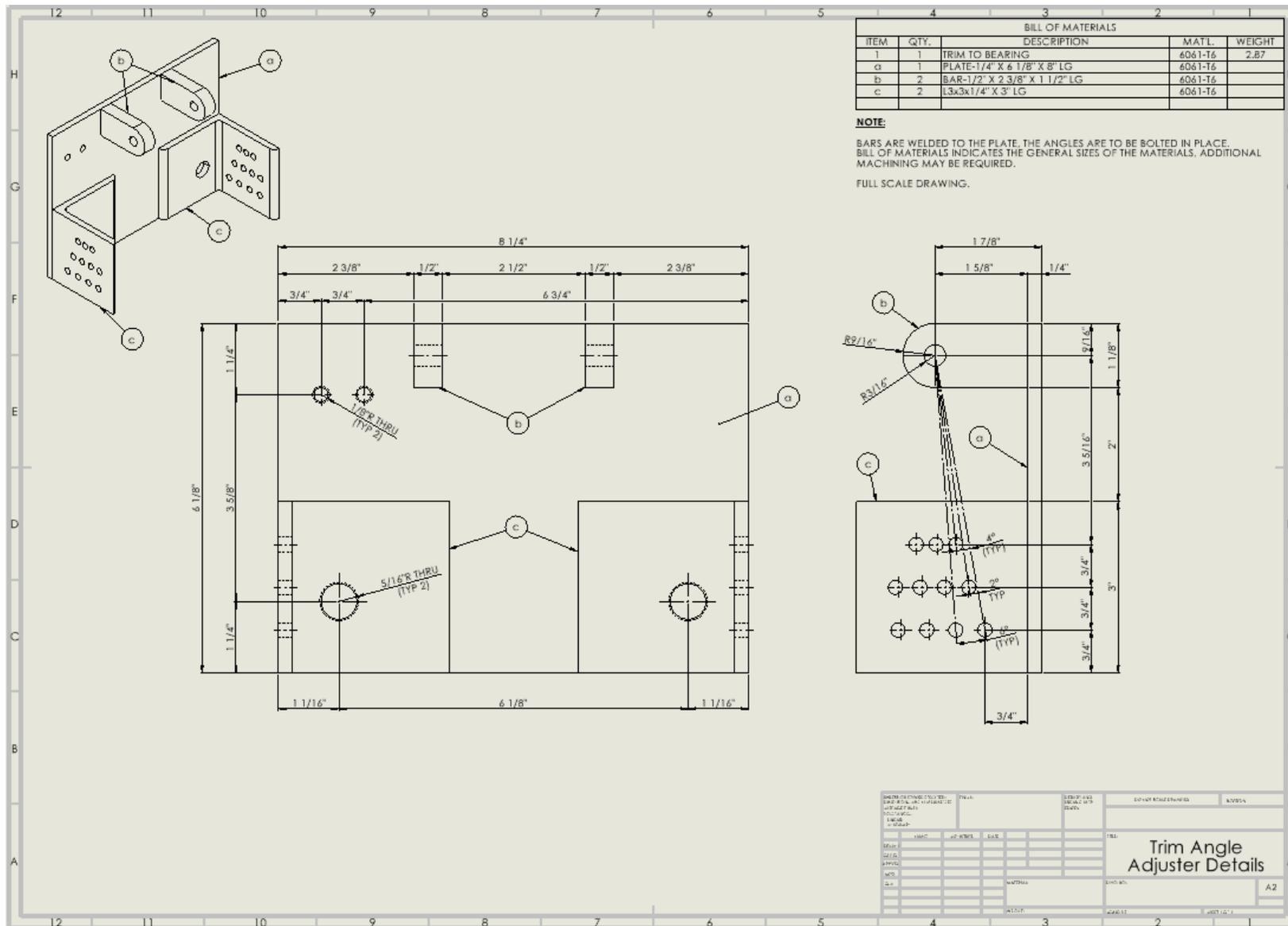
Figure 89: Fabrication Instruction Sheet for Trim Angle Adjuster Half that Bolts onto the Bearing.
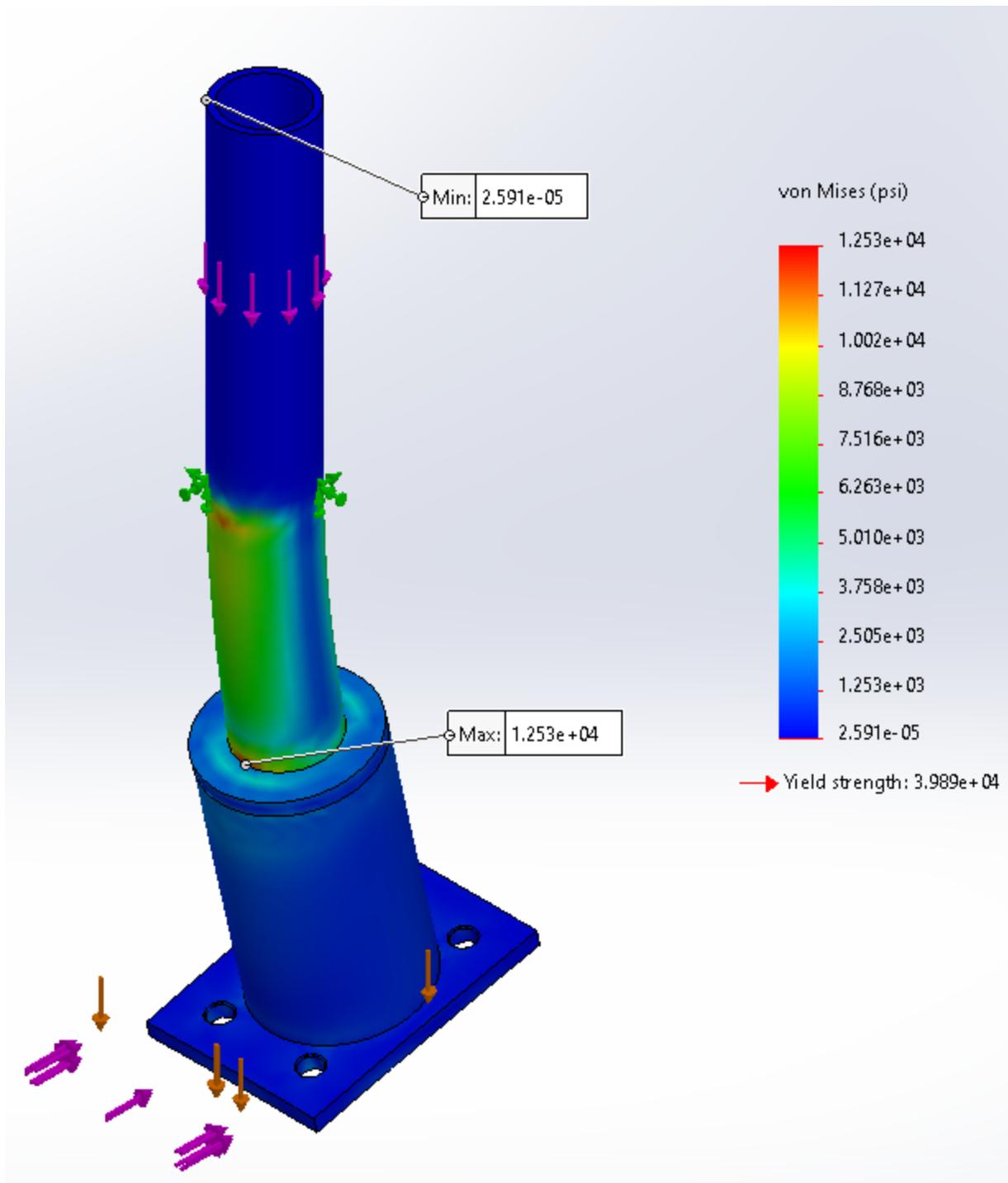
Figure 90: Lower Shaft Housing Stress Analysis
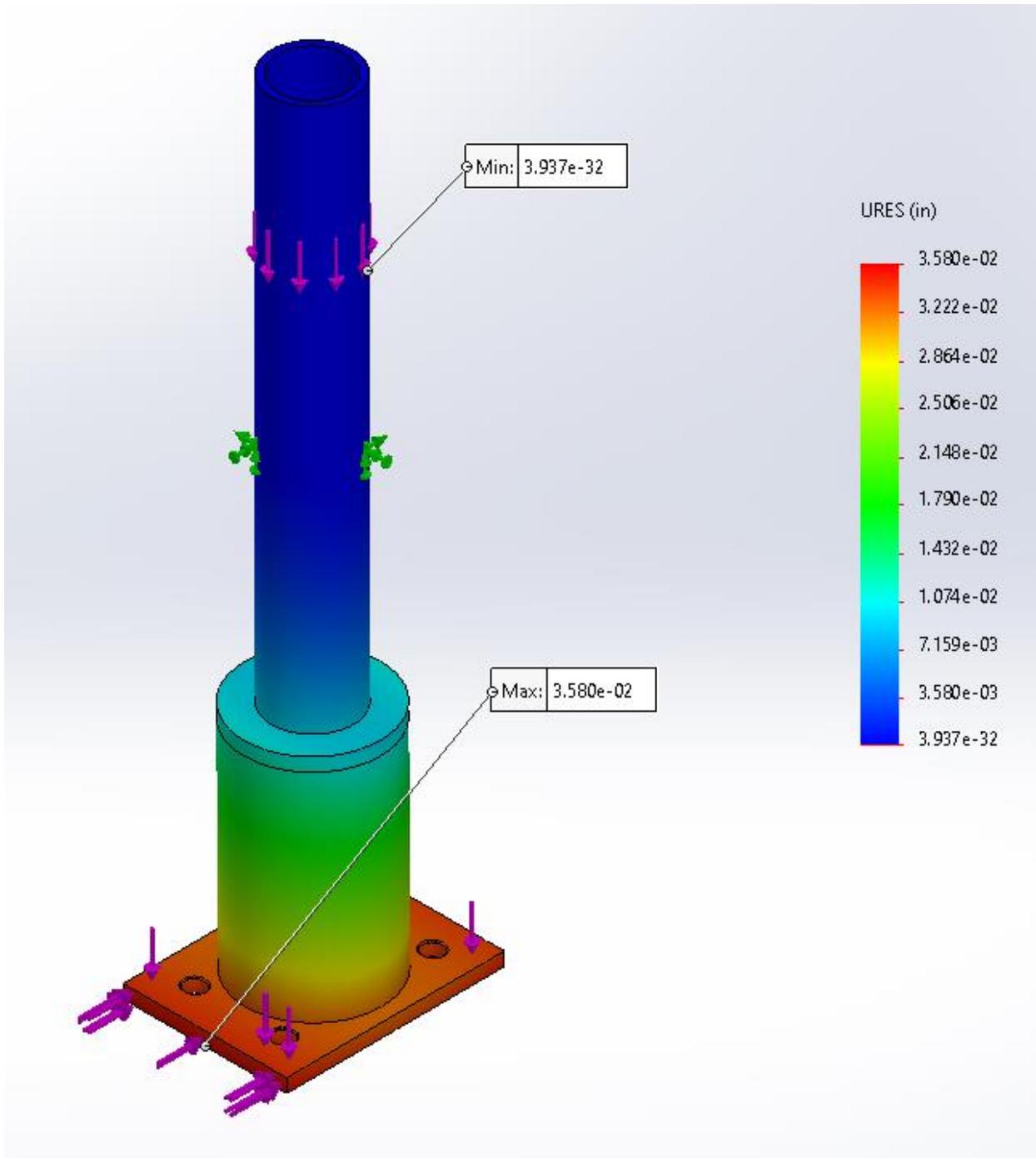
Figure 91: Lower Shaft Housing Displacement Analysis
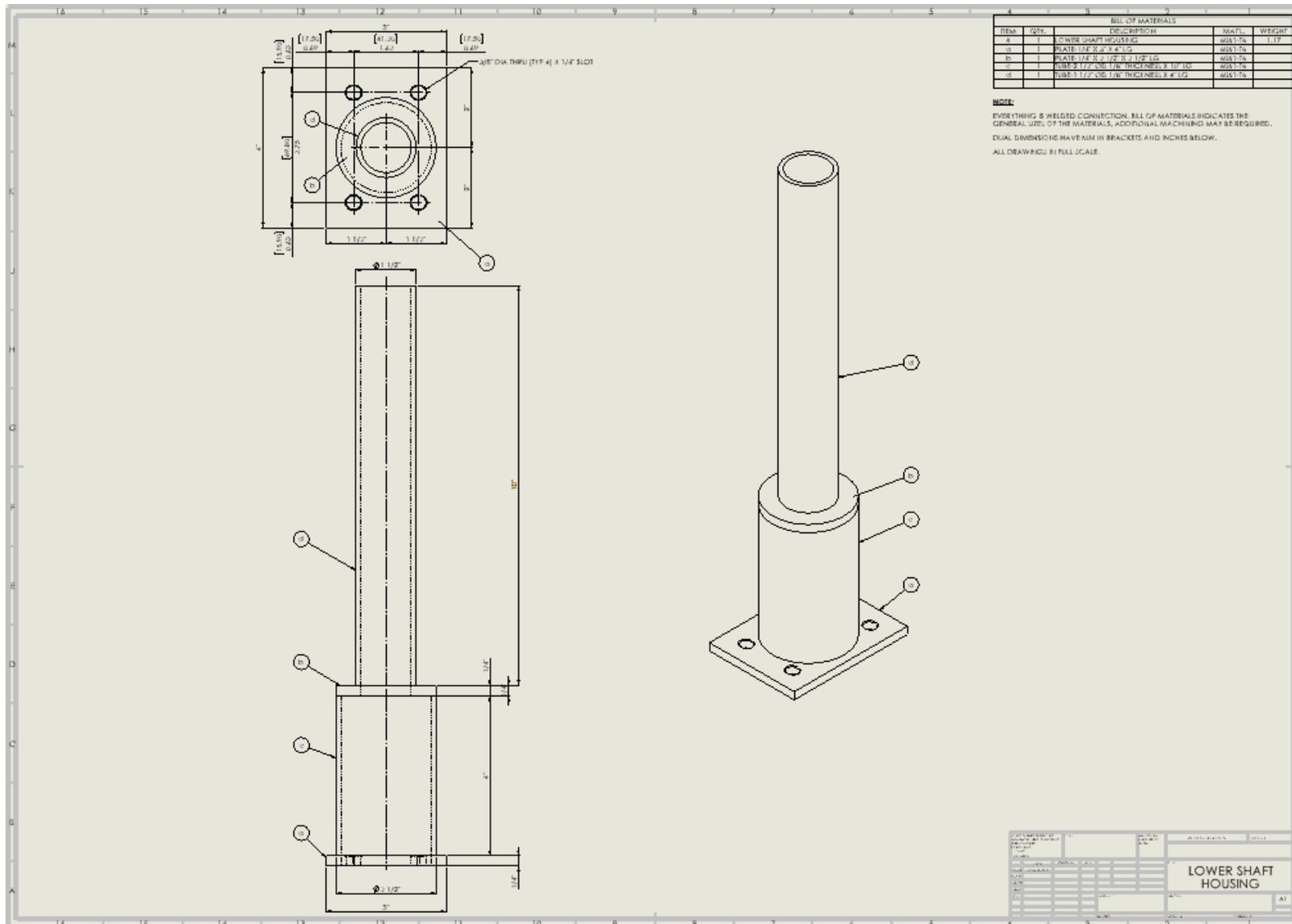
Figure 92: Fabrication Instruction Sheet for Lower Shaft Housing
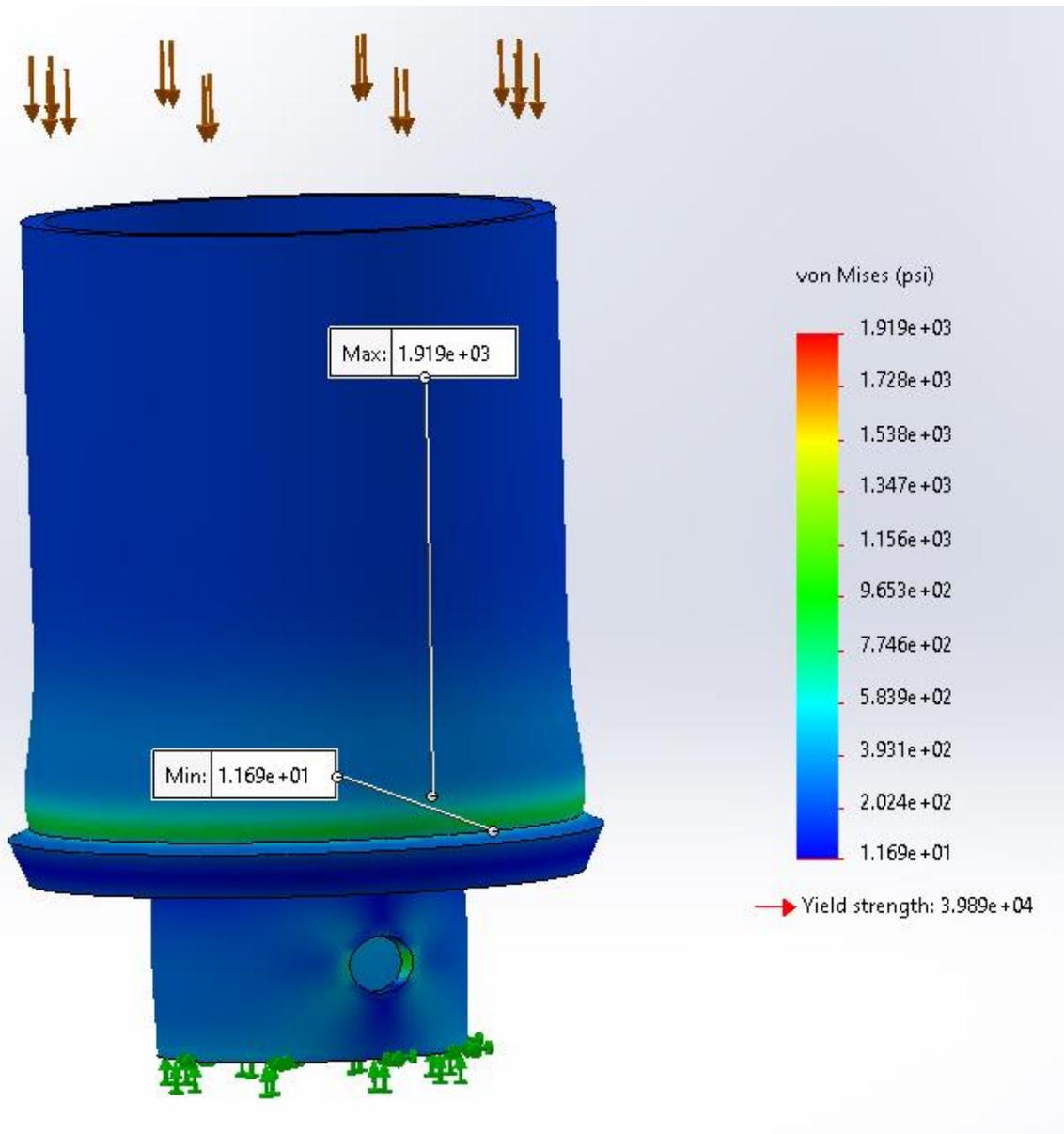
Figure 93: Middle Shaft Housing Stress Analysis
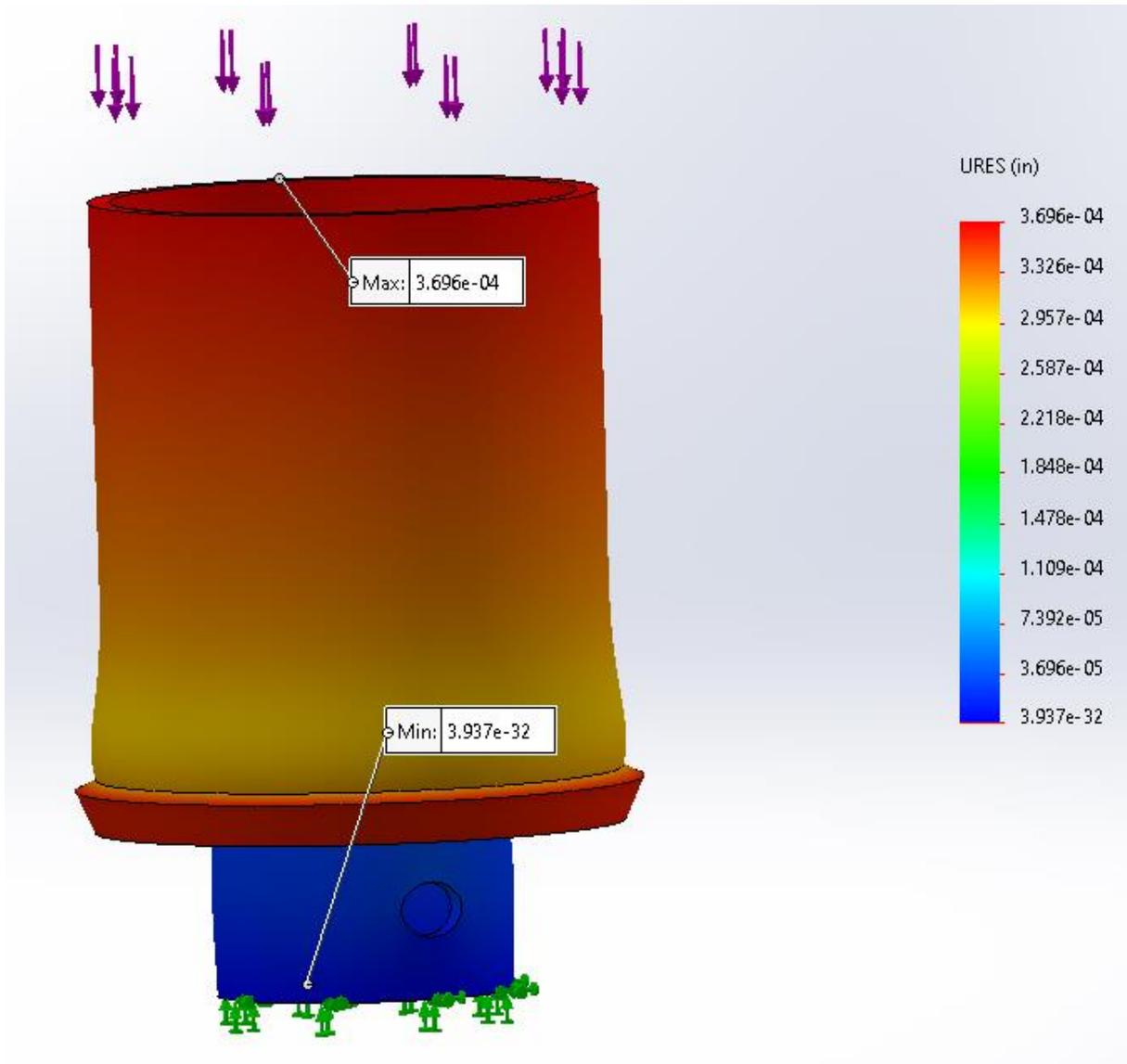
Figure 94: Middle Shaft Housing Displacement Analysis
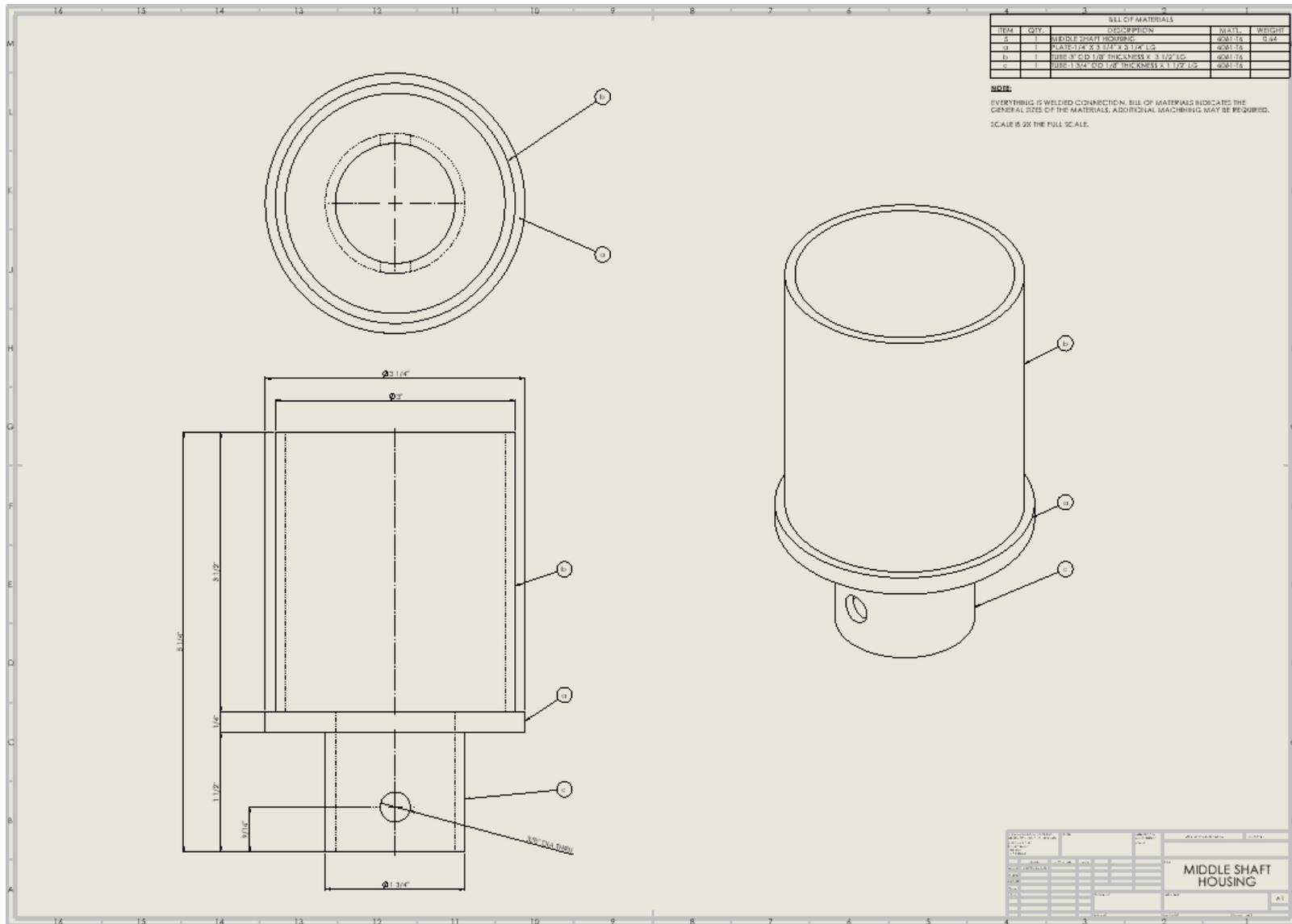
Figure 95: Fabrication Instruction Sheet for Middle Shaft Housing
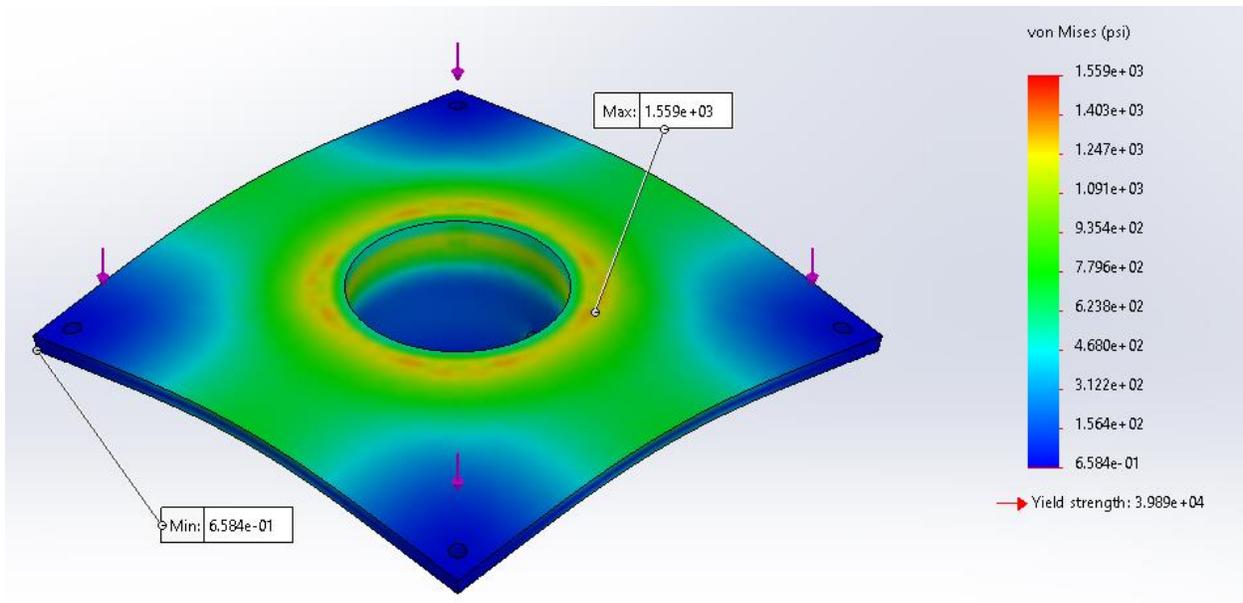
Figure 96: Top Shaft Housing Stress Analysis
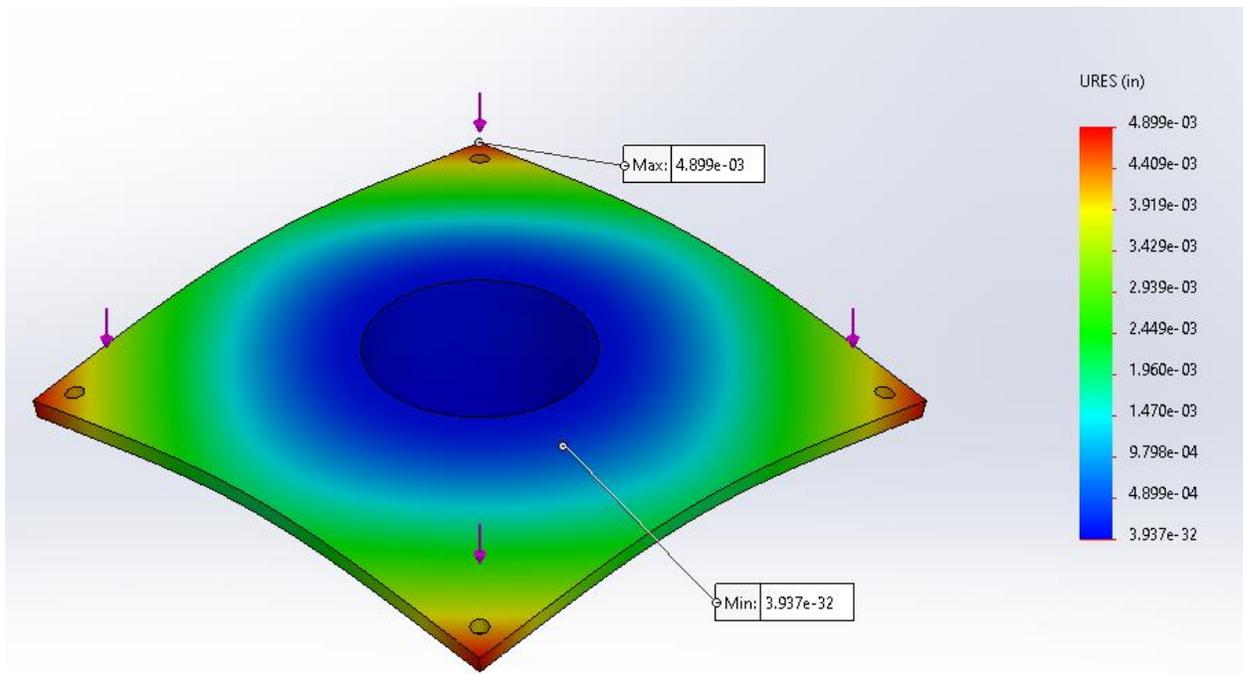
Figure 97: Top Shaft Housing Displacement Analysis
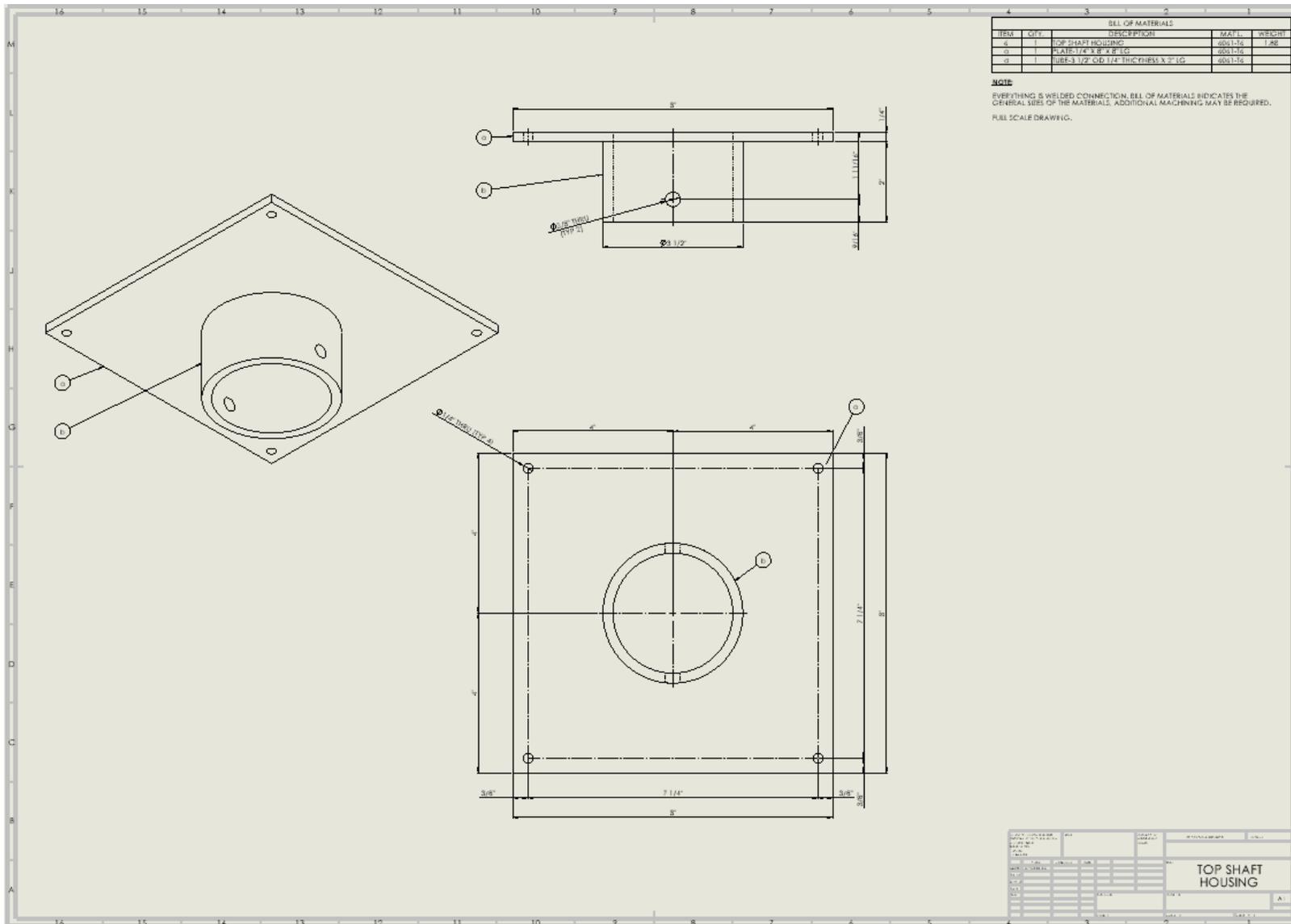
Figure 98: Fabrication Instruction Sheet for Top Shaft Housing
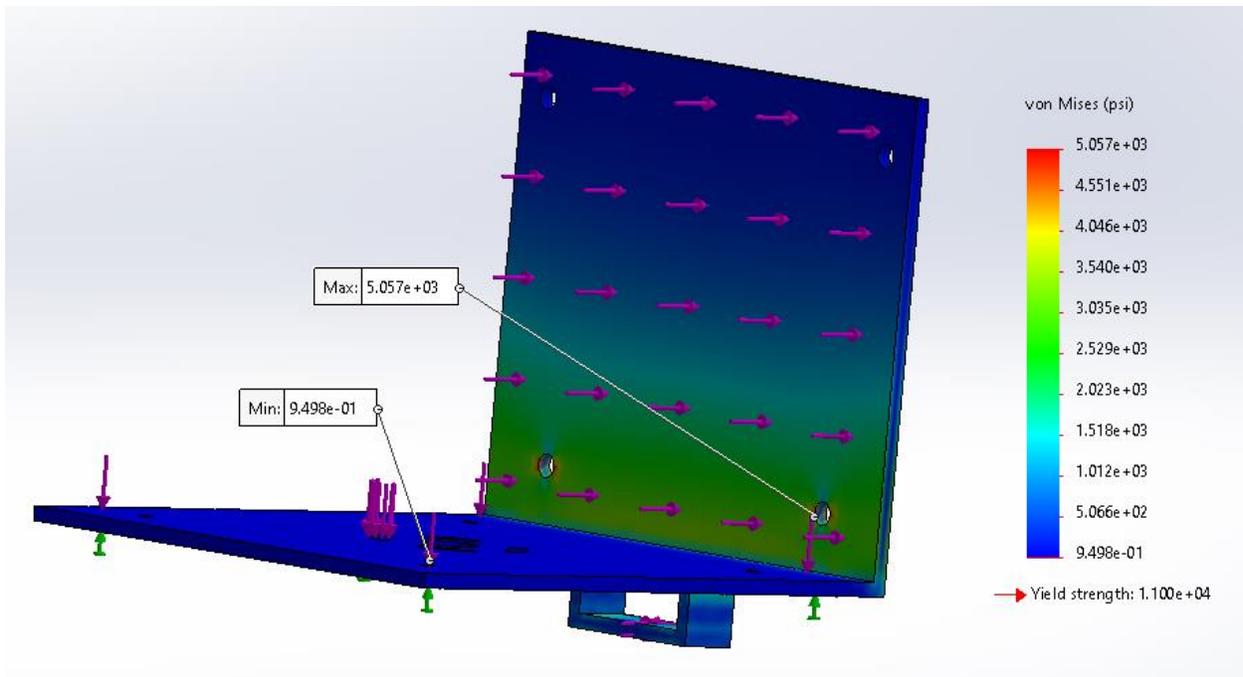
Figure 99: Motor Mount Stress Analysis
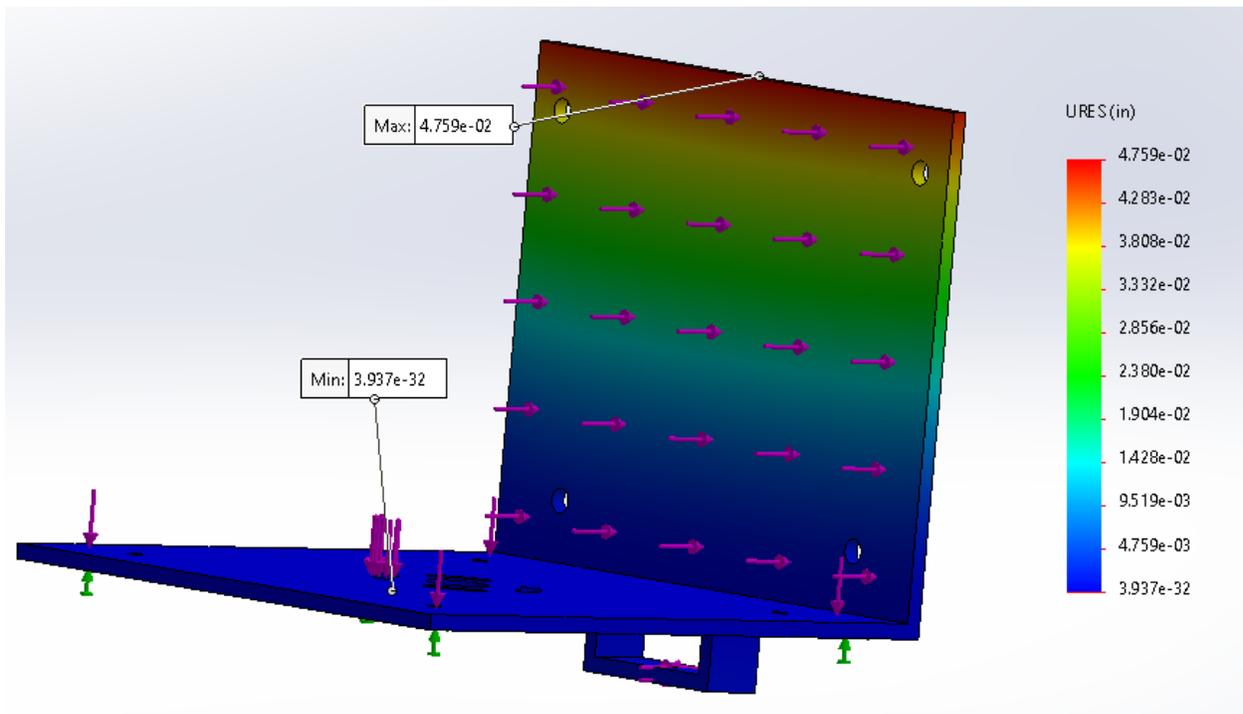
Figure 100: Motor Mount Displacement Analysis

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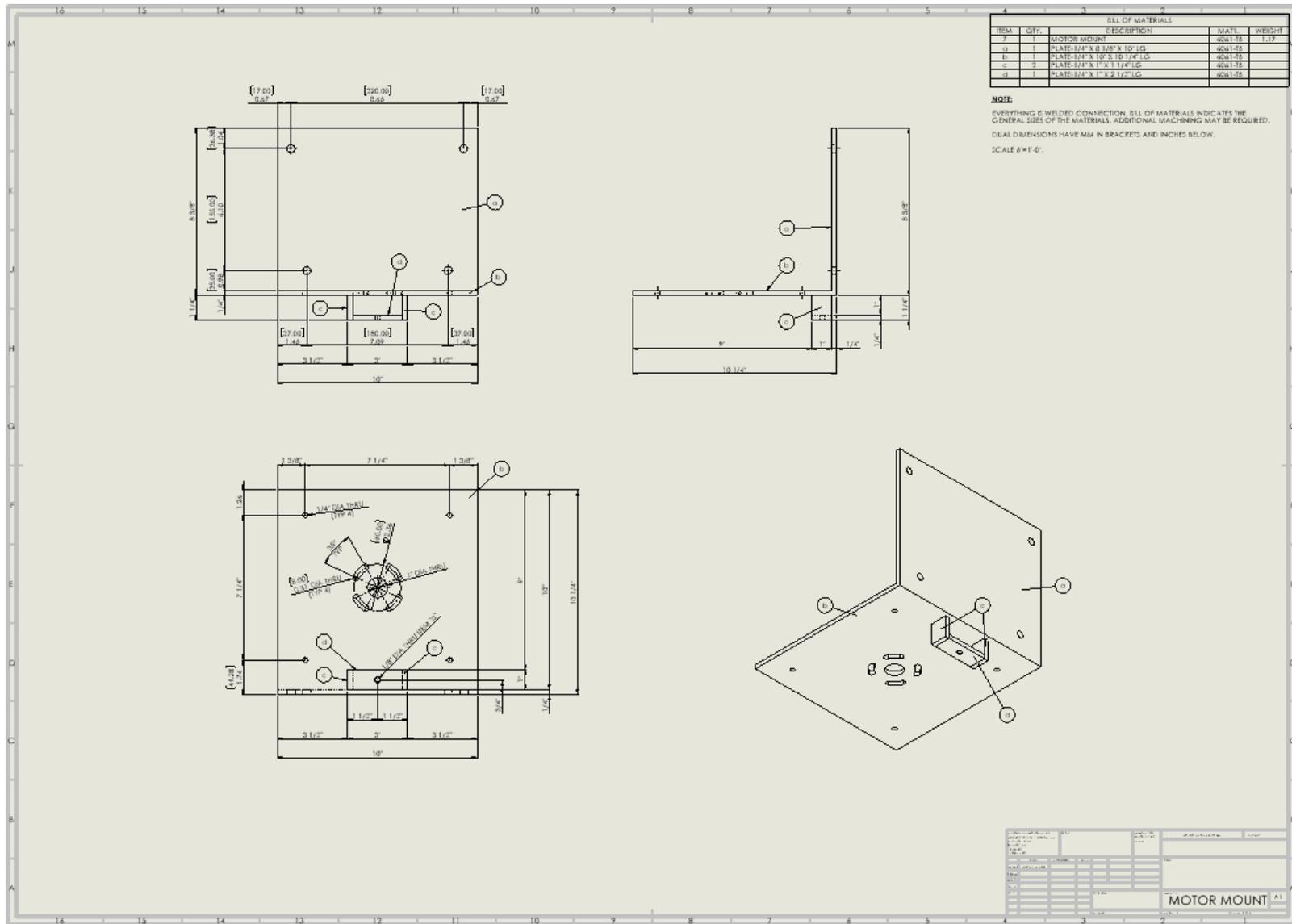
Figure 101: Fabrication Instructions for Motor Mount
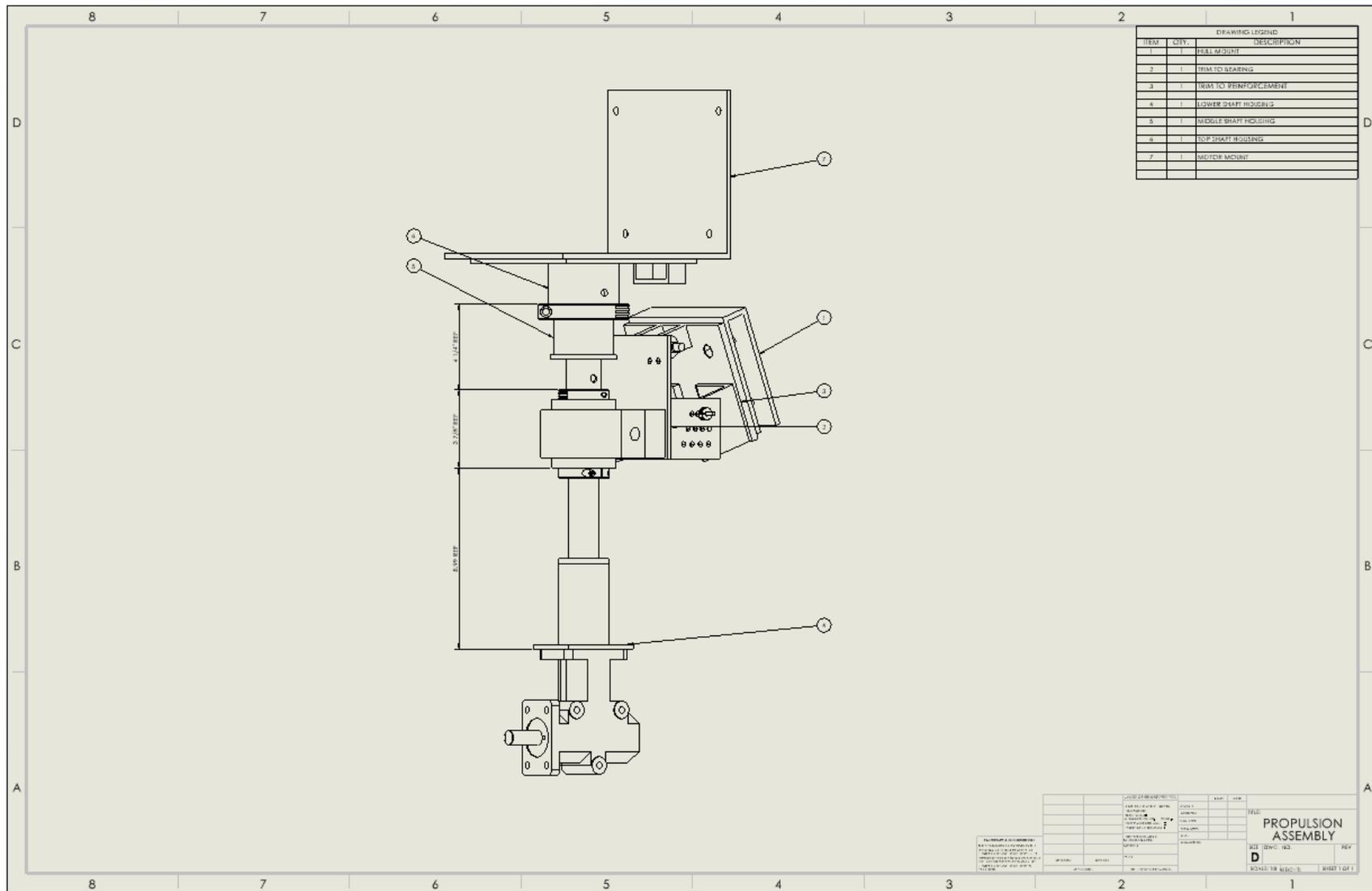
Figure 102: Propulsion System Assembly

Table 14: FEA Results for Trim Angle Adjuster and Propulsion System Connection Components.

Trim Angle Adjuster and Propulsion System Connection FEA Results				
Item	Von Mises (psi)		Displacement (in)	
	Maximum	Minimum	Maximum	Minimum
Hull Mounted Trim	3.80E+03	1.61E-04	7.32E-04	3.937E-32
Bearing Mounted Trim	3.52E+04	6.00E-03	4.86E-02	3.937E-32
Lower Shaft Housing	1.25E+04	2.59E-05	3.58E-02	3.937E-32
Middle Shaft Housing	1.92E+03	1.17E+01	3.70E-04	3.937E-32
Top Shaft Housing	1.56E+03	6.58E-02	4.90E-03	3.937E-32
Motor Mount	5.06E+03	9.50E-01	4.76E-02	3.937E-32

Table 15: Weights Used for Trim Simulation

Table Of Masses	
Object	Weight (lbs.)
Seat	5
Front Steering	10
Steering Cable	5
Entire Propulsion System Connection	30
Drive Train and Propeller	23
Solar Panel Frames	35
Solar Panels	40
Batteries	81
Electrical Components	35
Motor	56
Skipper A	150
Skipper B	180
Hull	50

Table 16: SolidWorks Flow Simulation Data for Trim Angle Adjuster

SolidWorks Flow Simulation Data							
Weight of Skipper (lbs.)	Surface Area (in^2)	Surface Area (ft^2)	Volume (ft^3)	Input Velocity (ft/s)	Actual Velocity (ft/s)	Friction Force (lbs.)	Change in Pressure (lb/in^2)
150	856.92	6.0940278	0.25382226	6	5.0868	0.021678	-5.131167
				9	7.8666	0.230818	-8.125254
				12	11.321	0.065598	-10.6402
				15	14.018	0.058221	-13.95988
180	877.54	6.0940278	0.26875299	6	5.1513	0.020163	-5.065072
				9	8.1490	0.023727	-7.635792
				12	10.676	0.035695	-11.25570
				15	14.018	0.058221	-13.95988
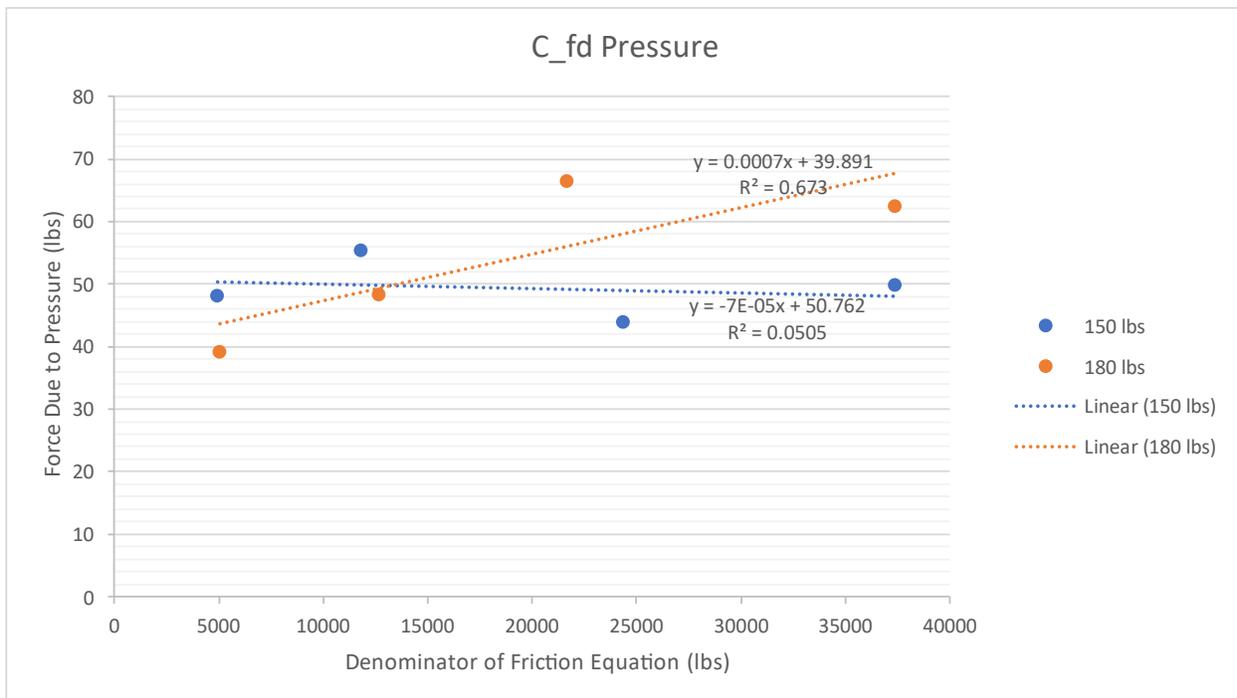
Figure 103: Pressure Coefficient of Friction Plot.
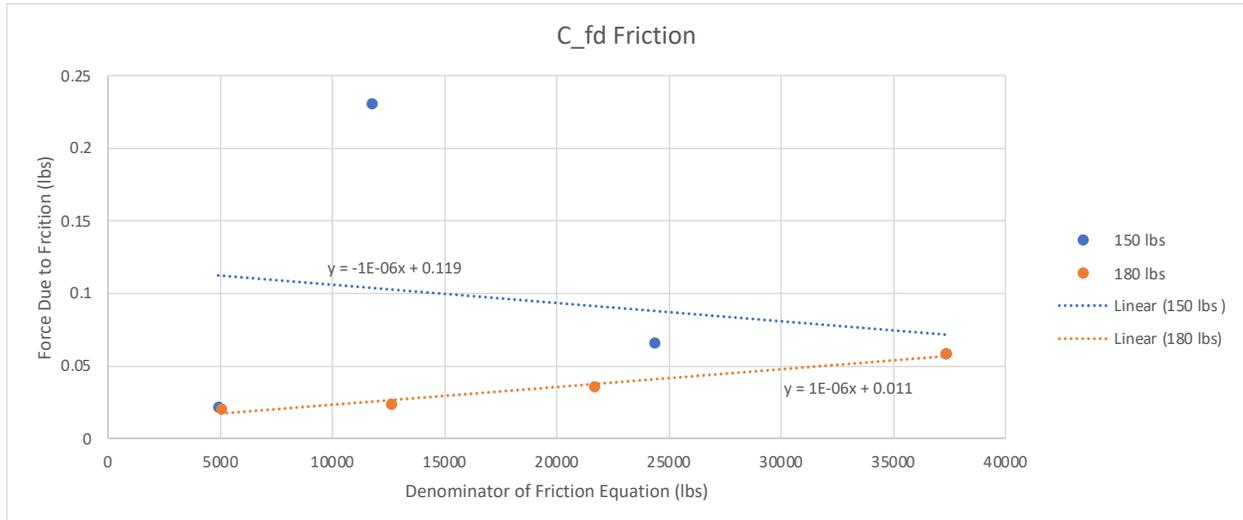
Figure 104: Surface Coefficient of Friction Plot.
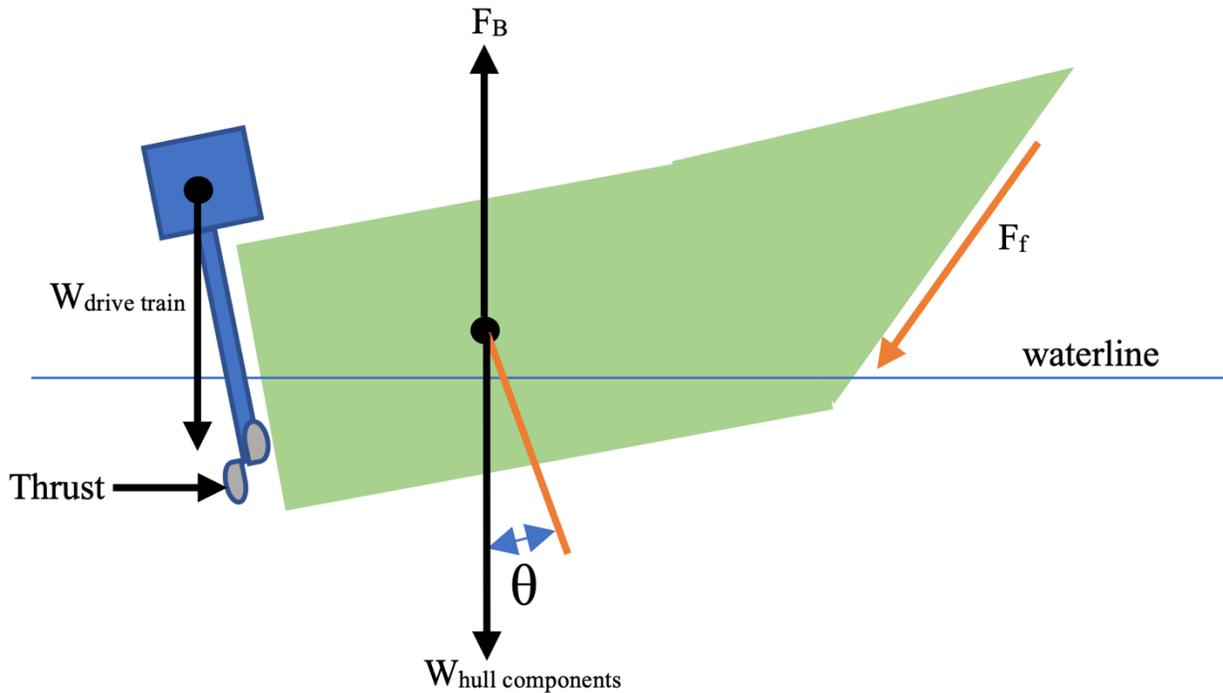
Figure 105: Force Diagram for Trim Angle Calculations.

The vertical forces were summed, set equal to zero and solved for theta. Equation 10T is the force balance, and Equation 11T is the final form that was plugged into the MATLAB code, shown in Appendix F. In the following equations, W$_{components}$ is the combined mass of the hull and all components, including the skipper and drive train.

$$\Sigma F_y = F_B - W_{components} - F_f \sin(\theta) = 0 \tag{10T}$$

$$\theta = \sin^{-1}(\frac{F_B - W_{components}}{F_f}) \tag{11T}$$

Appendix E

This appendix includes the detailed manufacturing and assembly instructions for the propulsion system that will be given to the USI 2023 Solar Splash team.

1) Construct shaft housing (top, middle and lower) and motor mount using the manufacturing drawings attached. See below in Figure 106.
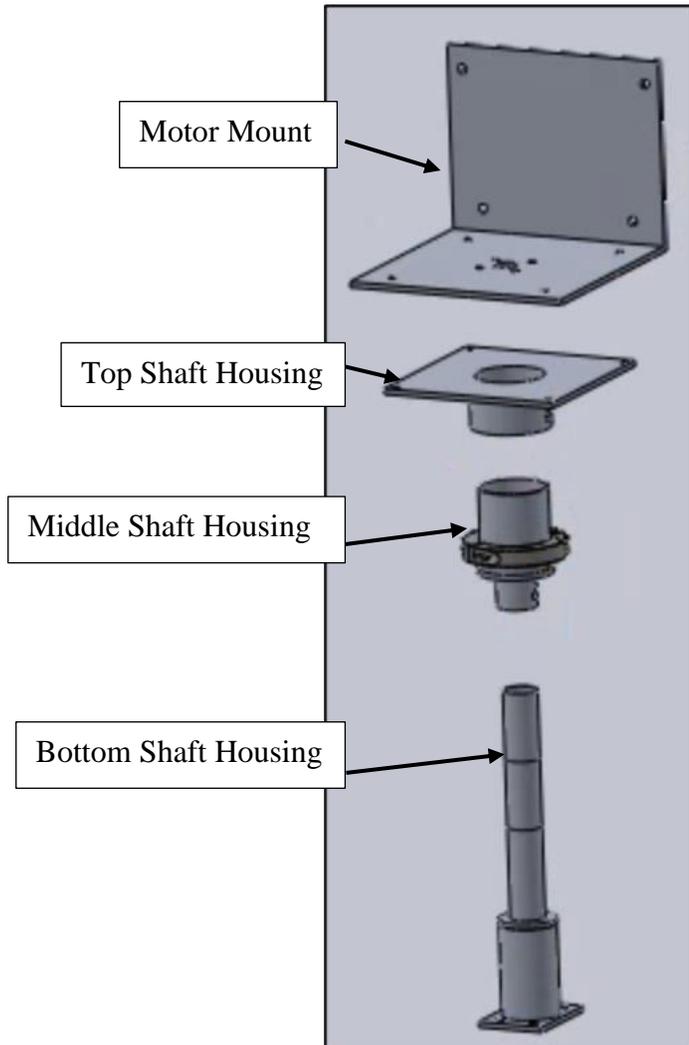
Figure 106: Exploded View of Shaft Housing

2) Assemble couplers [1 coupler = (1) 19mm hub + 1 spider + (1) 20mm hub]. See assembly below in Figure 107.
    a. The following video tutorial of similar couplers may be helpful:
       https://www.youtube.com/watch?v=1nngogpFjB0
    b. These couplers are not designed to be weight bearing! Assemble the system horizontally to avoid axial loads. Do not pick up the system vertically until the entire system is assembled.
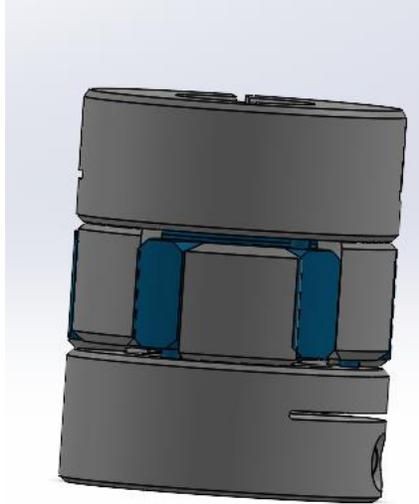
Figure 107: Assembled Coupler

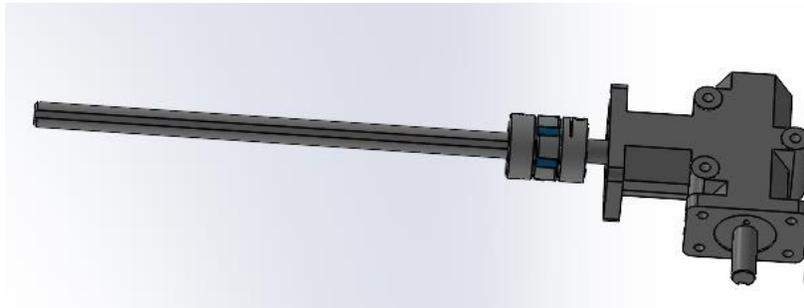3) Attach drive shaft to gear box via (1) coupler. See Figure 108.

Figure 108: Drive Shaft, Coupler, and Gearbox Connection

4) Attach lower shaft housing onto gear box with (4) 1/4" grade 8 bolts w/ (1) locking nut & (1) washer. See Figure 109.
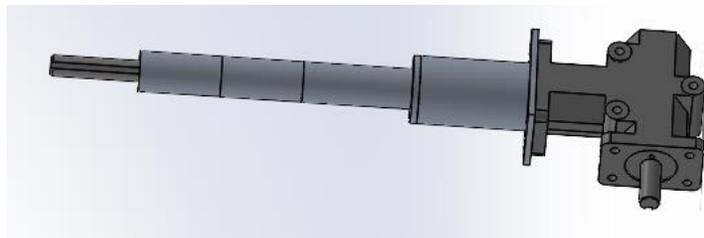

Figure 109: Lower Shaft Housing Placement

5) Slide the 1 1/2" bearing onto the lower shaft housing (to be properly located at the end of assembly). See Figure 110.
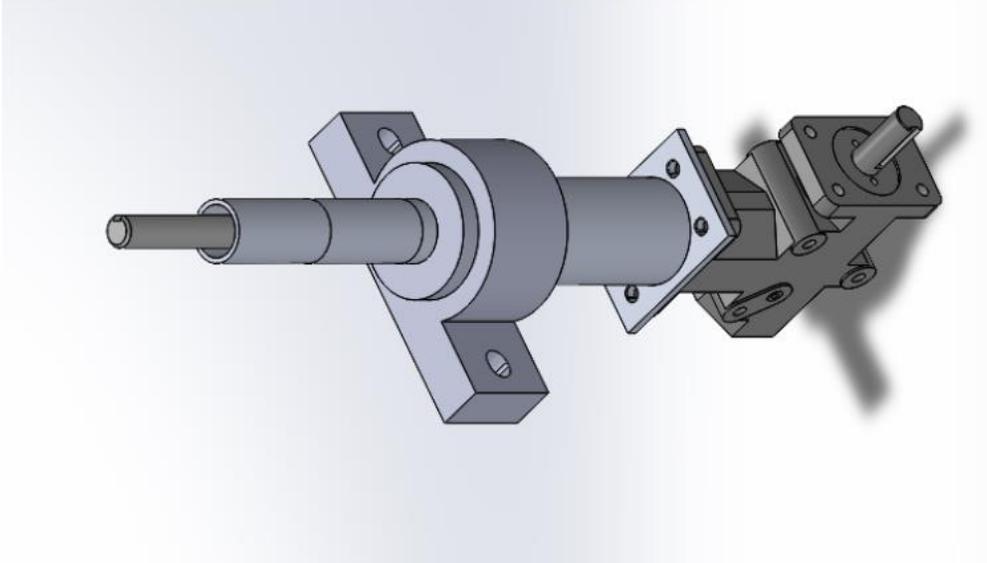
Figure 110: Place Bearing on Lower Shaft Housing.

6) Slide the middle shaft housing onto the lower shaft housing (to be properly located at the end of assembly). See Figure 111.
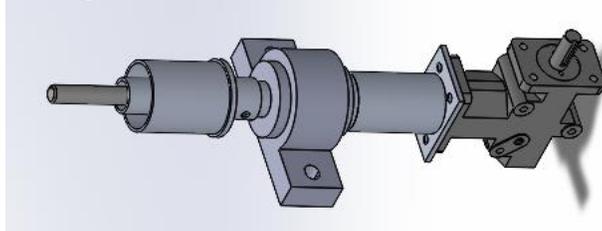

Figure 111: Middle Shaft Housing Installation

7) Slide the top shaft housing onto the middle shaft housing (to be properly located at the end of assembly). See Figure 112.
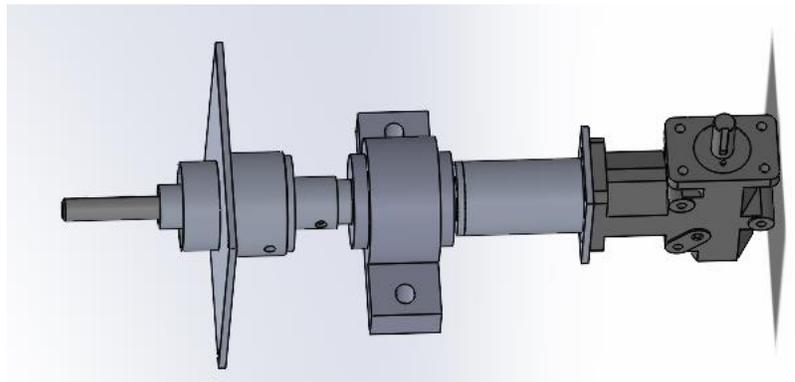

Figure 112: Top Shaft Housing Installation.

8) Screw the motor mount onto the motor: first screw bottom of motor (where the shaft is) w/ (4) M8x1.25 bolts. Then screw side plate of the motor to the motor mount w/ (4) 3/8" bolts w/ (1) washer & (1) locking nut.
9) Attach motor to drive shaft via (1) coupler, attaching the motor to the coupler first. See Figure 113.
    a. Make sure the motor and the rest of assembly is horizontal because couplers are not designed to be weight bearing.
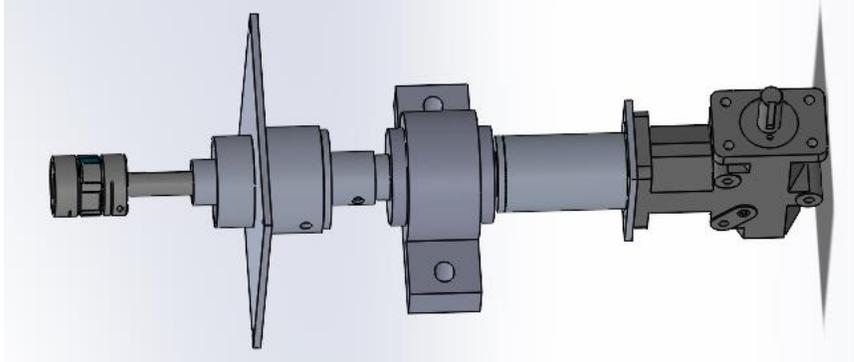   b. Note: the motor is not shown here, because no SolidWorks version exists from the manufacturer.

Figure 113: Second Coupler Installation

10) Slide upper shaft housing up and bolt to motor mount with (4) 1/4" bolts, (1) locking nut, and (1) washer. See Figure 114.
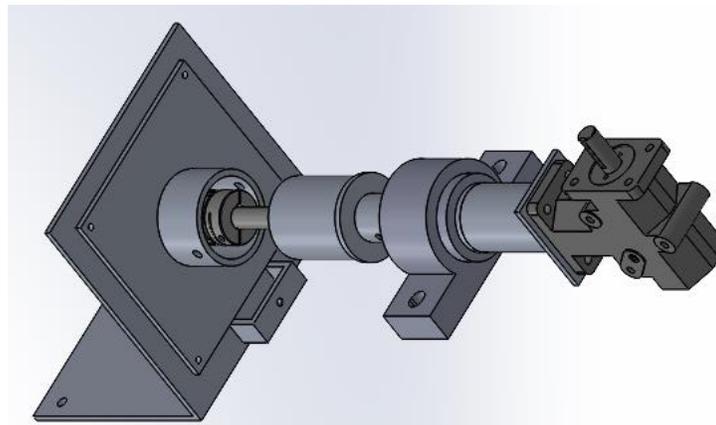

Figure 114: Motor Mount Installation

11) Slide middle shaft housing up and lock in place vertically with (1) locking collar and (2) 3/8- inch set screws. See Figure 115.
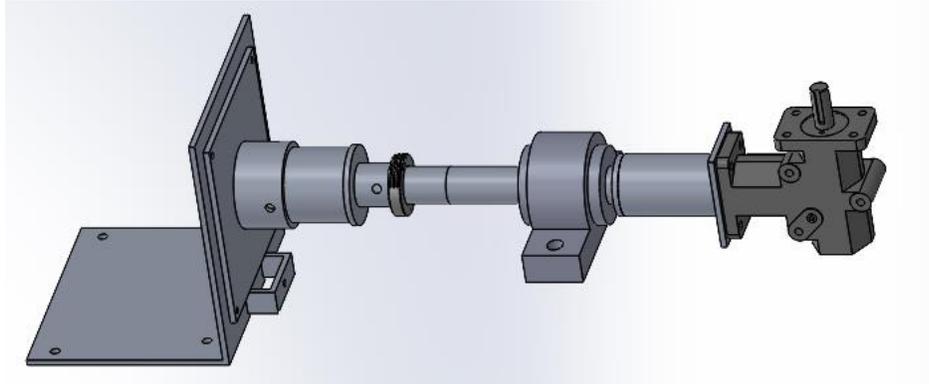

Figure 115: Shaft Housing Assembly, Middle Shaft Housing Locked in Place

12) Lock top shaft housing in place with (1) locking collar and screw in (2) set screws. Note: set screws not shown in image for simplicity. See Figure 116.
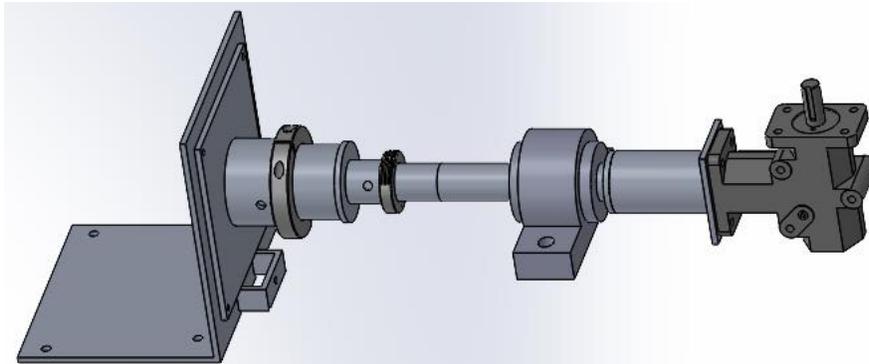

Figure 116: Shaft Housing Assembly, Top Shaft Housing Locked in Place.

13) Slide bearing up to designated location, locking in place with (2) locking collars: (1) locking collar below bearing, and (1) above, for this use the locating collar from the middle shaft housing. See Figure 117.
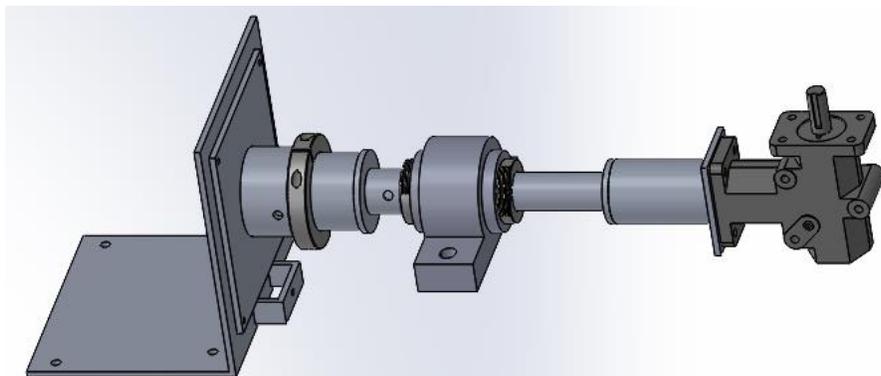

Figure 117: Locking the Bearing in Place.

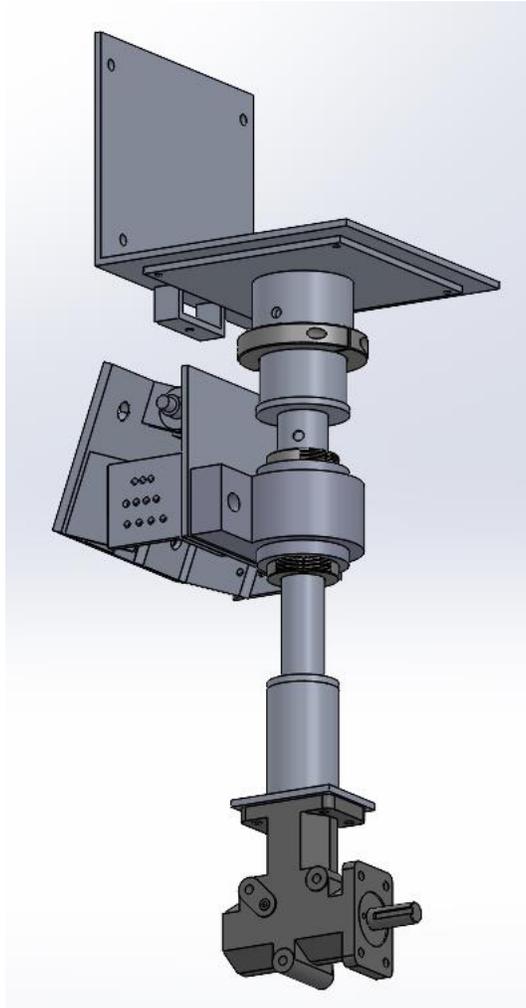14) Bolt bearing onto trim adjuster with (2) 5/16-inch bolts w/ (1) locking nut and (1) washer. See Figure 118.

Figure 118: Propulsion System Assembly

Appendix F

```
clear all
clc

%% center of mass calculations
% the origin is the hinge of the trim angle adjuster
% assumed uniform mass distribution along the width of the boat

%%%% skipper (change this mass for each skipper)
% weight
m_sk = 175; % lbs.
% COM location
x_sk = 120; % in

% solar panels and frames
%%% change to zero when they are removed from the hull for the races this
% applies to
% weight
m_sp = 35; % lbs.
% COM location
x_sp = x_sk; % in

% seat
% weight
m_se = 5; % lbs.
% COM location
x_se = 30; % in

% Batteries
% weight
m_ba = 30; % lbs.
% COM location
x_ba = 130; % in

% steering wheel mount stuff
% weight
m_fs = 20; % lbs.
% COM location
x_fs = 40; % in

% drivetrain
% weight
m_dt = 150; % lbs
% COM location
x_dt = -4; % in
% distance from the propeller to the pivot point
y_dt = -16; % in

% resulting moments
b_sp = m_sp*x_sp; % lb*in, solar panels
b_se = m_se*x_se; % lb*in, seat
b_sk = m_sk*x_sk; % lb*in, skipper
b_ba = m_ba*x_ba; % lb*in, batteries
b_fs = m_fs*x_fs; % lb*in, wheel mount
```

```matlab
b_dt = m_dt*x_dt; % lb*in, drive train

%% take masses and plug into COM EQN
COM = (b_sp + b_se + b_sk + b_ba + b_fs + b_dt)/(m_sp + m_se + m_sk + m_ba +
m_fs + m_dt); % center of mass calculator along the length of the hull with
the trim hinge being (0,0) in inches, not including the drive train or motor
m_tot = m_sp + m_se + m_sk + m_ba + m_fs + m_dt; % total mass of entire
system

% center of mass and total mass not including the drive train
m_h = m_tot - m_dt; % lbs, weight of the items in the hull (located on), not
including the drive train
COM_h = (b_sp + b_se + b_sk + b_ba + b_fs)/(m_sp + m_se + m_sk + m_ba +
m_fs); % center of mass calculator along the length of the hull with the trim
hinge being (0,0) in inches, not including the drive train or motor

%% plug in velocity to calculate the force
v_ft = 49.7; % ft/s, maximum velocity
v_k = v_ft*0.592484; % knots, velocity
w_f = 0.98*0.83*(v_k^0.047); % wake factor
v_a = v_k*w_f; % speed in knots
hp = 20.11; % max horsepower (max power possible with this motor). When
new/different horsepower value, multiply by the % of energy that is likely to
be used in this application, or simply redefine this value to the amount of
energy that the motor is projected to be used for each speed.
thrust = 326*hp*0.6/v_a; % lbs



%% plug in skipper weight to approximate the cfd value

% cfd values are the coefficient of friction, found using the data from
% solidworks flow simulation.

if m_sk < 180
    cfd = 0.0007 + 0.000001; % first number is the cfd of the pressure and
the second is that of the actual friction. These values are additive, so they
were combined for this application.
    SA = 6.09422078; % ft^2, SA submerged. This value changes, but based on
hte calculations, the difference isn't that significant
    V_sub = 0.253822265; % volume of submerged fluid in ft^3
    w_l = -10.835+9/16; % waterline, inches, below pivot point
else
    cfd = 0.00007 + 0.000001; % coefficient of friction
    SA = 6.09422078; % ft^2, SA submerged. This value changes, but based on
the calculations, the difference isn't that significant
    V_sub = 0.268752986; % volume of submerged fluid in ft^3
    w_l = -10.813+9/16; % waterline, inches, below pivot point
end



%% use COM calcs to find the angle, or a halfway point to get something
useful for the calculations

g = 32.2*12; % acceleration due to gravity, in in/s^2 (that's what the *12 is
for)
```

```matlab
rho = 62.4; % lbs/ft^3, the density of water (the fluid the object is going
through)

% start the calculations
F_f = 0.5*cfd*v_ft^2*SA; % friction force, surface and pressure
F_b = V_sub*rho*g; % buoyant force, lbs

% solve for theta
% theta_im = acos(((F_b - m_h)*COM + thrust*(y_dt))/w_l) % results in an
imaginary number, so need to find the angle between the two values
theta_im = asin((F_b - m_tot/F_f)) %

%%%% here is the final trim angle--this is the angle you use when adjusting
%%%% the system
theta_deg = abs(atan(real(theta_im)/imag(theta_im))*180/pi) % take the atan
of the real number over the imaginary and convert to degrees
```

Appendix G

This appendix includes design factors considered in the development of the project.

Design Factor	Page Number or Reason Not Applicable
Public health, safety, and welfare	N/A: Solar Splash is an intercollegiate competition of educational purpose; it is not publicly sponsored and is intended to be a collegiate learning experience.
Global	pg. 2: Intercollegiate – International Teams Welcome
Cultural	N/A: Solar Splash is an intercollegiate competition of educational purpose; it is not culturally driven and is intended to be a collegiate learning experience.
Social	pg. 2: Communication and Learning with Peers pg. 71: Teamwork
Environmental	pg. 16: Material Selection for Solar Panel Frames pg. 32: Drive Train Material Requirement
Economic	pg. 16: Material Selection for Solar Panel Frames pg. 60: Hull Mount plate thickness selection
Professional Standards	pg. 4: IEC 60529 – IP67 and IP 68 pg. 41: ISO/R773 pg. 49: 29 CFR 1910 Subpart O