ME 4182 – Towing Two Jet Skis with a Pontoon

Final Report

Waverunners

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Executive Summary

The increasing demand for safer and more efficient maritime towing systems has highlighted the limitations of current solutions, particularly for dual towing of jet skis and emergency towing of pontoons. Existing products often lack durability, adaptability to dynamic water conditions, and compliance with maritime safety standards. This project aims to develop a robust, user-friendly towing mechanism tailored to these needs, offering a practical solution for recreational and emergency marine towing. The design focuses on meeting stringent engineering specifications and ensuring ease of use, modularity, and durability, all while maintaining compliance with safety and aesthetic requirements. The design objectives were shaped by a detailed analysis of customer requirements and regulatory standards. Key engineering specifications included towing forces of 80 lbf for jet skis and 320 lbf for pontoons, with a factor of safety (FOS) exceeding 2.5 to ensure reliability under worst-case scenarios. Additionally, the design needed to accommodate wave heights up to 5 feet, tilting and rolling angles of ±30°, and dynamic forces induced by towing and wave impacts. Extensive testing and analysis validated these parameters, with Siemens NX simulations confirming the structural integrity and performance of the key components. The square tube, a critical structural element of the towing mechanism, was designed with a width and height of 2 inches and a wall thickness of 0.5 inches. Analytical calculations combined with finite element analysis (FEA) in Siemens NX revealed a maximum deflection of 0.19 inches under a load of 80 lbf, well within the allowable deflection of 0.211 inches (L/360, where L = 76.5 inches. The back mechanism, another vital component, demonstrated a deflection of 0.184 inches under towing forces, providing an 8% safety margin. The design also incorporated flexible joints to handle dynamic forces effectively, preventing stress concentrations that could lead to fatigue over time. Stress analysis of the front hook and curved beam, modelled using combined shear and bending stress equations, further confirmed the robustness of the design. The curved beam demonstrated a total stress of 19,982 psi under combined towing and wave forces, comfortably below the tensile yield strength of 50,800 psi for AISI 1020 steel. These calculations validated that all components are capable of sustaining operational loads without permanent deformation or failure.

The modular nature of the towing mechanism allows for quick assembly and disassembly, taking under 20 minutes for a single operator. This ease of use, combined with corrosion-resistant

materials, ensures long-term durability even in harsh marine environments. The mechanism's aesthetic integration with existing watercraft designs also enhances its appeal to recreational users while maintaining a professional finish. To refine the product further, recommendations include integrating shock absorbers to reduce vibrations and stresses, enabling smoother towing under dynamic conditions. Additionally, modular attachments can increase compatibility with a broader range of pontoons and jet skis, enhancing market versatility. These enhancements, alongside the existing design, create opportunities for scalability, prototyping, and commercialization. In conclusion, this innovative towing mechanism addresses a critical gap in the marine industry, combining safety, reliability, and adaptability to meet the needs of recreational and emergency users. By incorporating rigorous engineering analysis, material selection, and modular design, the solution ensures compliance with industry standards while offering a pathway for future improvements. With its strong foundations in both theory and testing, the design is well-positioned for prototyping, production, and further innovation, providing significant value to marine enthusiasts and the broader market.

Introduction

The purpose of Team Waverunner's Capstone Design project is to create an attachment for a pontoon boat which enables it to tow two jet skis simultaneously, and in the case of an emergency such as engine failure, allow the jet skis to tow the pontoon back to safety. This problem was postulated by team sponsor Ken Wright, the founder and CEO of Novacc. Despite having over thirty years of industry experience and a vast knowledge of product design, fabrication and manufacturing, he recognized that the rigid constraints of this problem required a specialized engineering solution that is vetted and reliable.

Like him, millions of people enjoy boating on lakes and coves and often do not want to face the hassle of driving multiple vehicles out from the dock. They find it troublesome to manage watercrafts that are not in use safely and efficiently. Currently, the products available on the market which enable such towing are expensive, scarce and can neither be used to tow multiple jet skis nor can they perform an emergency tow. They are also mostly targeted for in-board engines whereas pontoon boats have outboard engines and thus do not have space at the back of the boat where these products connect. The most common solution is for people to perform custom modifications on their watercrafts to jerry-rig attachments which can tow. These attachments are unreliable, non-standardized, and often do not adhere to maritime safety guidelines. To address this problem that affects a vast market, Team Waverunners has been tasked to design, prototype, test and verify a mechanism which allows simultaneous and emergency towing. As an initial step, the scope of the product has been narrowed down to focus on jet skis and pontoons with similar attachments and geometry to Ken's, and its operation is limited to lakes and not open oceans.

This report outlines the entire design process from inception to fabrication. It begins with identifying unique customer requirements which must be addressed by the design. Experimental analysis is conducted to collect real-world data which informs the engineering specifications of the design, along with the customer requirements. Keeping in mind the constraints defined, multiple design options are presented, and the benefits and drawbacks of each one is discussed and quantified. After analyzing towing forces and wave dynamics, a final design is formed which addresses customer requirements, engineering specifications and can withstand all forces. This design is further validated through lifetime expectation calculations, which yield an infinite life for the product. Initial prototyping is conducted to verify these results further and define a manufacturing process. This enables the identification of current limitations of the design and the proposal of future improvements and iterations.

The report provides a comprehensive analysis of the problem, its constraints, and the possible solutions for it. The final design chosen is backed by experimental analysis, force analysis and kinematic analysis. The high Factor of Safety of parts and infinite life cycle of the product proves the product to be a safe and reliable product. The fabrication process allows the product to be mass-manufacturable and thus profitable and scalable. The unique design enables it to be patentable. And since there is always room for improvement, the next steps are to follow the future improvements proposed.

Customer Requirements

A complex and multi-faceted problem such as this one has multiple customer requirements that must be met to yield a satisfactory solution. Through analysis of the problem statement and sponsor needs, the following customer requirements must be met to deem the design as sufficient.

Dual Towing Capacity

As per the problem statement, the mechanism must enable a pontoon to simultaneously tow two jet skis, each of which weigh between 700 to 1000 pounds. The attachment must distribute the load across the pontoon's structure in a way which does not negatively impact its buoyancy, stability, propulsion or overall efficiency.

Emergency Tow

In case of an emergency such as failure of the boat's engine or other mechanical failure, the two jet skis should be able to tow the pontoon back to safety. To enable this, the mechanism must be able to transfer propulsion from the jet skis into a towing force for the pontoon.

Ease of Use

The sponsor emphasized the need for a system which can be assembled, attached and detached by one person. To further make it user-friendly, all removable parts should be attached to the main mechanism to prevent loss of parts in the water. Since recreational boaters who might lack mechanical skills will be using the product, the design must be simple and user-friendly.

Durability

The product will be used in harsh conditions, including fresh and salty water, extreme sunlight, cold waters and more. Thus, the materials selected must be resistant to corrosion, UV exposure, colder temperatures, and physical wear and tear. It must be robust to withstand long-term use while requiring minimum maintenance.

Regulatory Compliance

When operating personal watercrafts in lakes and other waterbodies, there are multiple maritime safety guidelines that must be adhered to. These are enforced by entities such as the U.S. Coast Guard nationally and the Georgia Department of Natural Resources at a state level. Thus, the mechanism itself and the way it is used must follow rules such as weight distribution, towing safety standards and operating procedures.

Aesthetic Integration

While the performance of the mechanism is critical, it must not detract from the visual appeal of the pontoon and jet skis. Since many boaters take pride in the appearance of the vehicles, it is imperative that the design be sleek, compact and easily retractable when not in use. It should seamlessly integrate with the existing aesthetic.

Experimental Analysis

To ensure the towing system's reliability and safety, the team conducted a series of experimental tests to verify the forces acting on the system during towing operations. By using a force transducer, the team captured data under various dynamic conditions, focusing specifically on the towing attachment. This data informed the design by validating the assumptions made during the initial calculations and confirming the structural integrity of the system. Each test was repeated multiple times to ensure consistent and reliable results. Key experimental scenarios and findings are listed below:

1. Force During Abrupt Start (Highest Tension Force)

This test aimed to simulate the maximum tension force experienced when the jet skis are rapidly accelerated from rest. The team recreated an abrupt start by pulling the jet ski with a sudden jerk and recorded the peak force using the force transducer. The highest tension values identified from this test were critical in defining the towing attachment's peak load capacity, ensuring the system could withstand extreme conditions without failure.

Takeaway for Design:

The maximum recorded tension force helps determine the material strength and thickness needed for the towing attachment. This ensures the component can handle sudden, high loads during operation without deformation or failure.

2. Force at Steady Speed (Normal Operating Load)

This test focused on the forces experienced while towing at a constant speed, replicating typical operating conditions. The force transducer measured the steady-state towing force, providing insight into the average load the towing system would endure during extended use.

Takeaway for Design:

The steady-state force informed decisions on optimizing the towing attachment's geometry to avoid over-engineering. Designing with these values ensures the system remains robust while minimizing unnecessary weight and material costs.

3. Force During Deceleration (Compression Forces)

This scenario evaluated the compression forces generated during sudden deceleration, such as when the pontoon or jet ski comes to a stop. A force transducer was placed to capture these forces, which are vital for understanding potential stress points and ensuring a smooth deceleration experience.

<u>Takeaway for Design:</u>

Understanding compression forces enabled the team to incorporate reinforcement in areas prone to high stress during braking. This ensures the system avoids damage or abrupt impacts, improving durability and user experience.

Practical Testing Procedures and Observations

To conduct the experiments, the team performed a series of site tests in Alabama as shown in Figure 1, using the sponsor's jet skis and pontoons. A force gauge was attached between the towing jet ski and the towed object, with the speed gradually increased from 5 to 10 mph. When towing another jet ski, the maximum recorded towing force was 40 lbf. In the case of the jet ski towing the pontoon, the maximum observed force rose to 180 lbf, which closely aligned with theoretical calculations for the combined water and air drag forces. These observations validated the experimental setup and provided confidence in the accuracy of the collected data. Measurements of the holes and brackets on the sponsor's equipment were also recorded during the site visit, ensuring that the final design would be compatible with the existing towing setup.

Figure 1. Site Visit in Alabama

Design Implications

The results of these tests significantly influenced the design of the towing system. The measured forces helped determine the material selection for the towing attachment, ensuring that the components could withstand peak loads without unnecessary weight or over-engineering. High-strength alloys and lightweight composites were considered to achieve this balance. The insights from steady-state forces informed the optimization of the towing attachment's geometry, ensuring durability under normal conditions while avoiding excessive material usage. Additionally, the compression forces observed during deceleration guided the reinforcement of areas prone to high stress, enhancing the system's safety and longevity. Measurements from the sponsor's equipment ensured that the design incorporated precise bracket and hole placements, facilitating seamless integration with existing components. Finally, the system was designed with appropriate safety factors based on the maximum observed forces, ensuring reliable performance even under unexpected or extreme conditions.

Engineering Specifications

When designing a towing rig for two jet skis and a pontoon, it is crucial to meet the client's functional requirements while adhering to industry standards for safety, durability, and ease of use all while accounting for a margin via Factor of Safety. The team's site visit provided critical insights into real-world towing scenarios, including the forces involved, operational conditions, and equipment dimensions, which directly influenced the development of the following engineering specifications (summarized in Table 1).

Maximum Wave Amplitude	Up to 5 ft
Tilting	± 30°
Rolling	± 30° °
Wave Endurance	3 million load cycles
Rust resistance	10 million corrosion cycles
Jet Ski Tow Force (with FOS = 2)	80 lbf
Force required to tow (with FOS = 2)	320 lbf
Towing Speed	3 - 5 mph
Clearance Between Connection Points While Towing	4 ft
UV Resistance	Up to 5 years of use
Modular Assembly Time	< 20 minutes

Table 1. Target Values Per Specification

Derivation of Engineering Specifications:

1. **Wake Adaptability:**
   Derived from the average wave height in lakes (3 ft) with an added safety margin of 2 ft, ensuring the rig can handle significant wave motion while maintaining stability.
2. **Tilting and Rolling:**
   Set at ±30° based on typical watercraft behavior in lakes, ensuring the mechanism can accommodate lateral and rotational offsets without strain.
3. **Wave Endurance:**

Calculated from observed wave frequencies (300 waves/hour), expected daily operation (10 hours/day), and a design lifespan of 500 days, resulting in 3 million load cycles.

4. **Rust Resistance:**
   Based on standard steel corrosion resistance and ASTM B117 salt spray test results, with a target of 10 million cycles to ensure longevity in freshwater environments.
5. **Jet Ski and Pontoon Tow Force:**
   Measured during experimental towing tests, with a Factor of Safety of 2 applied to the maximum observed forces to account for emergency loads.
6. **Towing Speed:**
   Determined from maritime guidelines for towing inactive personal watercraft (PWC), set between 3–5 mph to ensure safety and compliance.
7. **Clearance Between Connection Points:**
   Derived from observations during the site visit, ensuring a 4-ft gap to prevent collisions while maintaining operational ease for a single user.
8. **UV Resistance:**
   Set to 5 years, aligning with material properties and expected exposure to sunlight during typical use in open-water environments.
9. **Modular Assembly:**
   Influenced by the client's preference for ease of use, ensuring a setup time of under 10 minutes with lightweight, pre-assembled components.

To meet the wave endurance requirement, the design will use high-grade materials with proven fatigue resistance and incorporate flexible joints to reduce mechanical wear. For rust resistance, all exposed components will undergo corrosion-resistant treatments, and optional saltwater-rated materials will be available for customers using the rig in harsher marine environments. UV-resistant coatings will be applied to prevent material degradation from sunlight exposure. Additionally, vibration damping and modular assembly features will improve durability and usability under diverse marine conditions.

By considering these expanded specifications, the towing rig will exceed the typical standards for marine applications, ensuring safety, reliability, and ease of use for the client.

Design Options

The development of the towing mechanism involved an iterative process, with two initial designs evaluated before arriving at the final comprehensive solution. Each design iteration was assessed based on its ability to meet the engineering specifications outlined earlier, including adaptability to dynamic water conditions, durability, ease of use, and stability. The early designs highlighted critical limitations, which informed the improvements implemented in the final design.

Design 1

Design 1 was created for pontoons with front hooks located at the corners of the pontoon tube, as shown in Figures 2 and 3. These pontoons typically have decks close to the tubes, leaving enough space for mounting attachments. This design used independent hooks on either side to secure the towing mechanism, ensuring effective stress distribution and allowing the structure to collapse for compact storage.

Figures 2 and 3. Front Hook Located in the Corner of the Pontoon

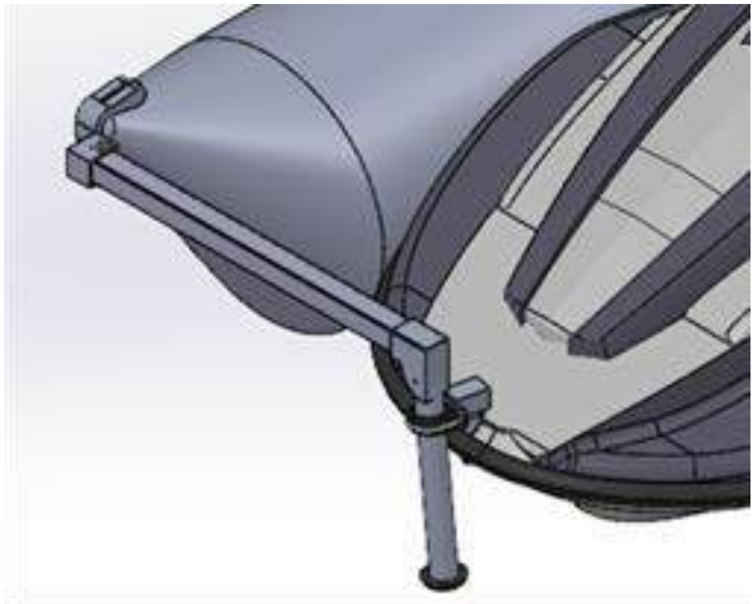


Figure 4. Isometric View of Design 1

However, while Design 1, shown in Figure 4, addressed the specifications for rust resistance and wave endurance by utilizing corrosion-resistant materials and sturdy construction, it fell short in other areas. The independent hooks lacked flexibility, preventing the system from accommodating the ±30° tilting and rolling requirements specified for stability in dynamic lake conditions. This rigidity risked creating excessive stresses at the attachment points, particularly when waves or uneven water surfaces caused rolling or tilting. Additionally, the design failed to fully address the towing force specification, as the independent setup struggled to distribute the forces evenly, especially under emergency towing conditions.

Design 2

Design 2 built upon the insights from Design 1 and introduced a configuration suited for pontoons with a flat surface at the front of the tube. As shown in Figures 5 and 6, this design expanded the system's versatility, enabling it to fit a wider range of pontoon configurations, including those described in Design 1. By focusing on a more modular approach, Design 2 improved on some of the functional limitations of its predecessor, offering better stability and adaptability for varying attachment points.

Figure 5 and 6. Pontoons with Front Hook with Surface Area

Despite its improvements, Design 2, shown in Figure 7, still failed to fully meet the engineering specifications. While it accounted for the towing speed specification (3–5 mph) and improved upon the clearance requirement of 4 feet between connection points, it did not resolve the issue of dynamic adaptability. The independent nature of the connections persisted, limiting the design's ability to handle rolling and tilting forces effectively. Moreover, while the flat surface connection reduced the risk of jet ski lifting, it did not fully prevent it, leaving stability concerns unaddressed under dynamic towing conditions.

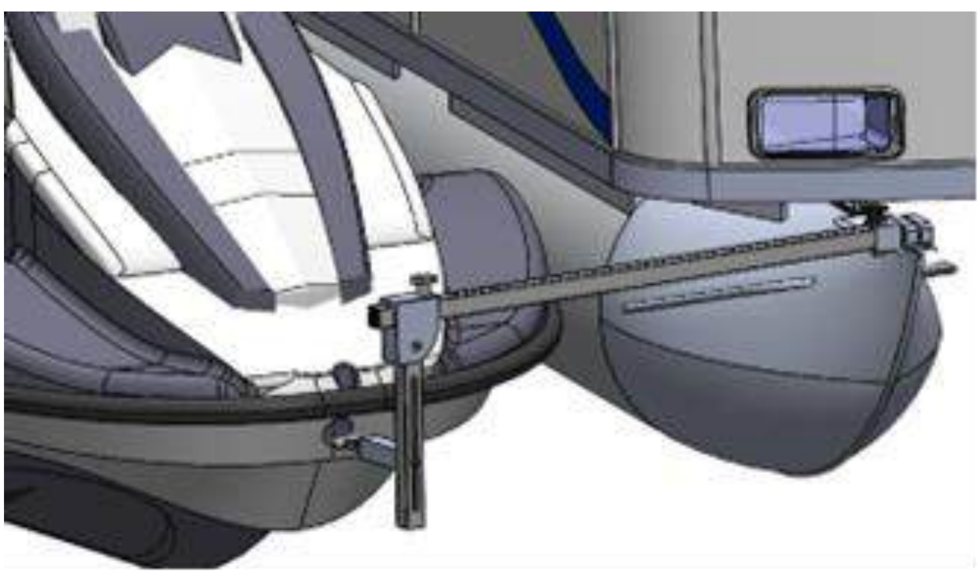


Figure 7. Final Assembly of Design Option 2

The development of these designs was heavily guided by the engineering specifications established earlier in the project. The wake adaptability requirement of up to 5 feet was unable to be addressed by either of the designs which led to the incorporation of a unified single-beam structure capable of distributing forces evenly during wave impacts, ensuring stability even in dynamic conditions. The tilting and rolling specification of ±30° was a critical factor, leading to the integration of flexible joints in the final design, allowing for smooth movement and adaptability to uneven water surfaces. To meet the wave endurance target of 3 million load cycles, high-strength steel was used, with additional reinforcements in high-stress areas to ensure long-term durability. Similarly, the rust resistance specification of 10 million corrosion cycles was achieved by selecting corrosion-resistant materials and applying protective coatings. The towing force requirements—80 lbf for jet skis and 320 lbf for pontoons (with a factor of safety of 2)—were directly addressed through the unified beam structure, which allowed for efficient force distribution, minimizing localized stress. Finally, the specifications for towing speed (3–5 mph) and clearance (4 feet between connection points) were met through careful design considerations that ensured smooth, safe operation while maintaining user convenience and simplicity. These specifications provided the foundation for developing a robust and reliable towing mechanism.

Final Design and Validation

Towing Forces

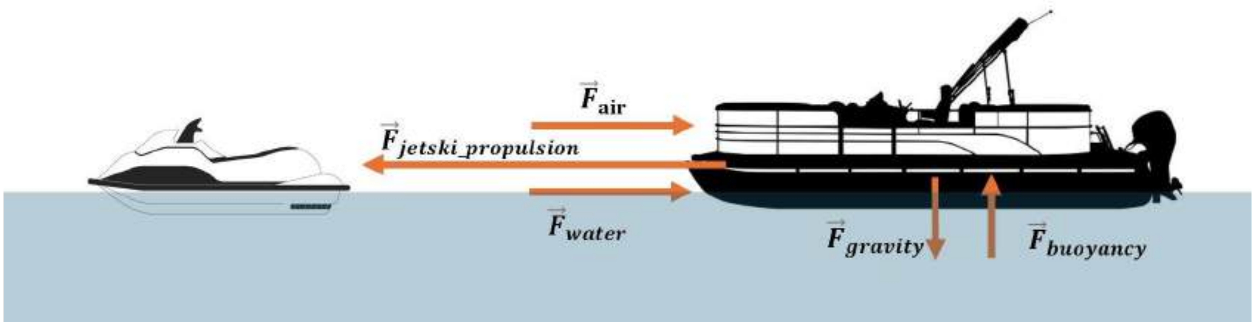


Figure 8. FBD of Jet Ski Towing a Pontoon

The maximum towing force is experienced in the alternate application, towing a pontoon with a jet ski. To calculate the maximum force, the pontoon is modeled as a cuboid with dimensions 288, 96, and 96 inches in length, width, and height respectively. This simplifies the calculations without underestimating the buoyancy and drag forces as shown in Figure 8. Similarly, the worst-case scenario for the weight can be estimated by adding the weight of the pontoon with the weight of 10 people, the max capacity:

$$m_{pontoon} = 2200lb + 10 \cdot 180lb = 4000lb$$

The volume of the pontoon can be found by modelling its base as a right rectangular pyramid using the formula below:

$$Volume_{submerged} = \frac{length \cdot width \cdot height}{3} = \frac{288 \cdot 96 \cdot height}{3}$$

Based on these assumptions, the height of the pontoon submerged in water can be calculated as follows:

$$F^{\rightarrow}{}_{buoyancy} = F^{\rightarrow}{}_{gravity}$$

$$\rho_{water} \cdot g \cdot Volume_{submerged} = m_{pontoon} \cdot g$$

$$height = \frac{m_{pontoon} \cdot 3}{288 \cdot 96 \cdot \rho_{water}} = 12.0229\, inch \approx 12\, inch$$

This submerged height of 12 inches can be estimated to approximate the drag forces by air and water. Continuing with a cuboid model of the pontoon to provide the worst case, the drag forces can be calculated as follows:

$$F^{\rightarrow}_{drag} = \frac{1}{2} \cdot \rho \cdot v^2 \cdot C_D \cdot A$$

Estimating the force on the pontoon due to water and air drag:

$$F^{\rightarrow}_{water} = \frac{1}{2} \times \rho_{water} \times 54.6807^2 \times 1.05 \times 12 \times 96 = 169.21\ lb$$

$$F^{\rightarrow}_{air} = \frac{1}{2} \times \rho_{air} \times 54.6807^2 \times 1.05 \times (96 - 12) \times 96 = \ 1.42813\ lb$$

Calculating the total force needed to be provided by the jet ski for propulsion:

$$F^{\rightarrow}_{jetski\_propulsion} = F^{\rightarrow}_{water} + F^{\rightarrow}_{air} = 169.21 + 1.42813 = 170.638\ lb$$

These results are approximately close to the experimentally obtained forces also recorded for the worst-case scenarios. For further design calculations, the figures will be estimated to be $F^{\rightarrow}_{jetski} = 40\ lb$ and $F^{\rightarrow}_{jetski\_propulsion} = \ 180\ lb$. These provide a safe margin while designing.

Wave Dynamics

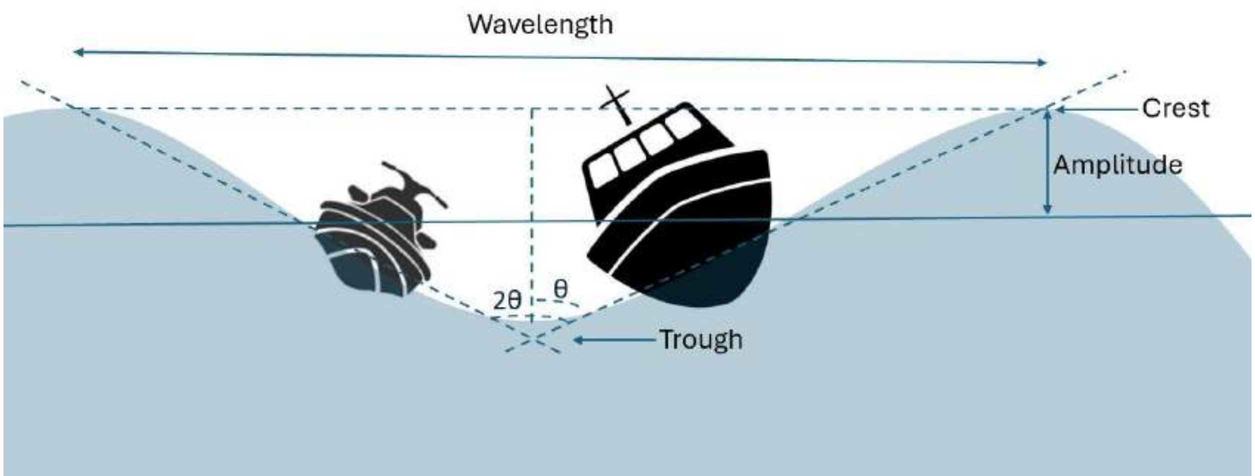


Figure 9. Worst-Case Tilt Angle caused by waves.

Alongside towing, the attachment will endure forces from waves, the worst of which, for lakes, is assumed to be 5-foot waves in amplitude with a 110 feet wavelength. The resulting angle between the boats, as seen in Figure 9, is calculated using the following formula:

$$\theta = \tan^{-1}\left(\frac{\frac{Wavelength}{2}}{2 \times Amplitude}\right)$$

$$Worst \text{ Case Angle} = 180 - 2\theta \approx 20°$$

Accounting for a FOS of 1.5 for the worst-case angle is taken to be 30° for further calculations.

Design Overview

Through the iterative design process, the front and back mechanism final design have evolved significantly from the first iteration. The front mechanism final design comprises three distinct sub-assemblies or subsystems to better understand its components: the square tube, the tube support, and the front tube assembly. This is shown in Figure 10.

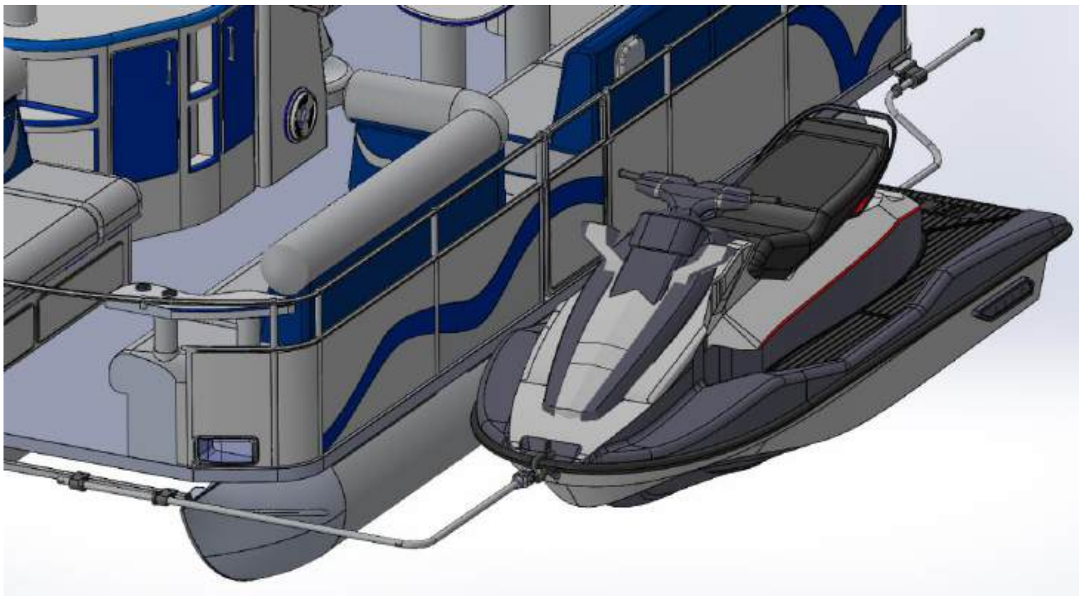


Figure 10. Front Mechanism with Subassemblies.

The back mechanism is composed of two subassemblies: the pontoon back connection and the S-Rod. This is shown in Figure 11.

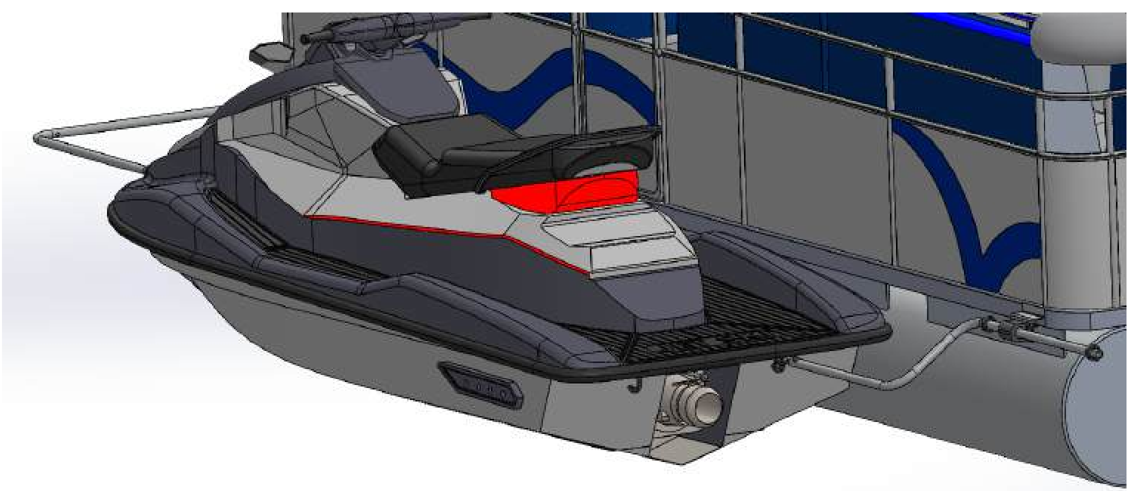


Figure 11. Back Mechanism with Subassemblies.

Front Attachment

The entire assembly for the front mechanism without the watercrafts is shown in Figure 12.

Figure 12. Front Mechanism

The square tube assembly shown in Figure 13 consists of a 6.3-foot steel square tube that connects to the front hooks of the pontoon using two bolts and two custom square nuts welded into the tube. This design was chosen over others to accommodate the axial forces applied to the front hooks of the pontoon. The front hooks can support massive loads, as it should be able to support the whole weight of the pontoon in the air, that is why the design uses them as the attachment point for the entire front mechanism.

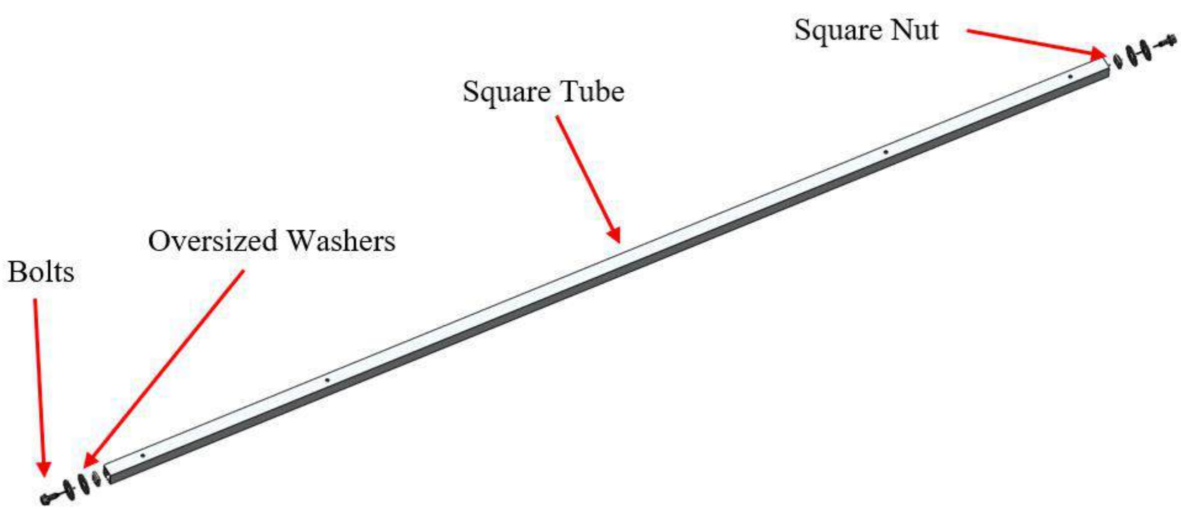


Figure 13. Square Tube Assembly

In earlier iterations, the design planned for a mechanism that would attach to only one hook at a time, leaving the two front hooks unconnected. However, this approach would have increased the moment applied to each front hook, reducing stability. To better balance the load, it was decided to connect the two hooks with a metal bar. The square tube used in the final design is made of 304 stainless steel, it has an outer thickness of 2 inches and an inner thickness of 1.5 inches. The connection is secured with a 3/8-16 bolt, chosen for its ability to provide a clamping force of 7041 lb calculated later. This configuration ensures a secure and stable attachment.

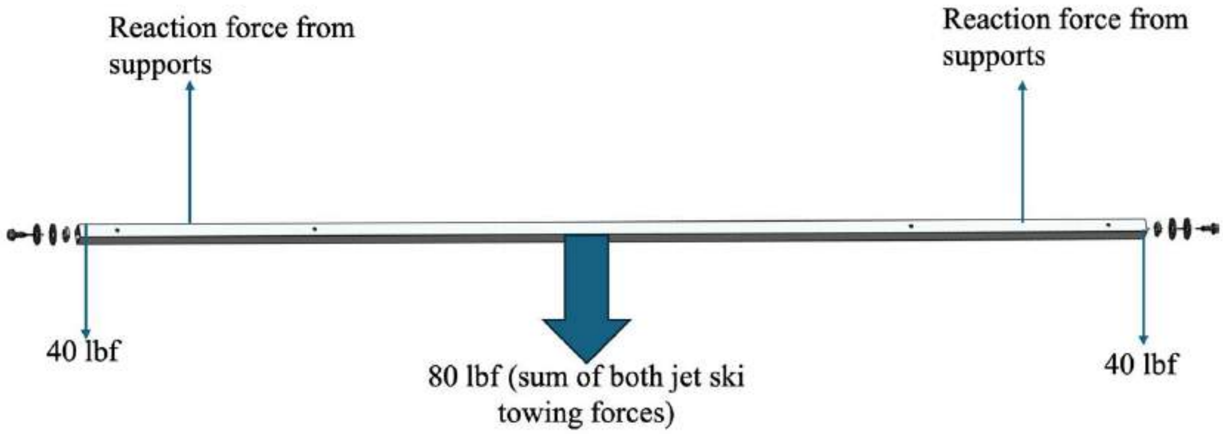


Figure 14. FBD for Square Tube Deflection

The deflection of the square tube, a critical component of the towing mechanism, was analyzed to ensure compliance with structural and operational requirements. Figure 14 shows the free-body diagram for the load experienced by the bar. The 80 lb force was used to calculate the deflection to simulate the worst case deflection of the square tube. For maritime applications, the maximum permissible deflection is L/360 , where L = 76.5 inches, resulting in an allowable deflection of 0.211 inches. The square tube, with dimensions of 2 inches in width and height, and a wall thickness of 0.5 inches, was subjected to a load of 80 lb force. Using Siemens NX, the deflection was calculated as 0.19 inches, which falls within the permissible limit, leaving a safety margin of approximately 10%. This analysis confirms that the square tube can safely withstand the applied loads without excessive deformation, ensuring structural stability and reliable towing performance under expected operating conditions.

The square nut is a very important part of the mechanism. The dimensions of this part were derived based on the maximum thickness that will be supported based on bolt grip length. As the grip length is limited by the diameter and hence it does not provide any advantage to have the thickness of the square nut greater than 0.375 inches. These dimensions are more than sufficient to securely hold the metal bar in place, accounting for the movement and vibrations the pontoons will experience while cruising on the lake. Figure 15 shows the square nut.

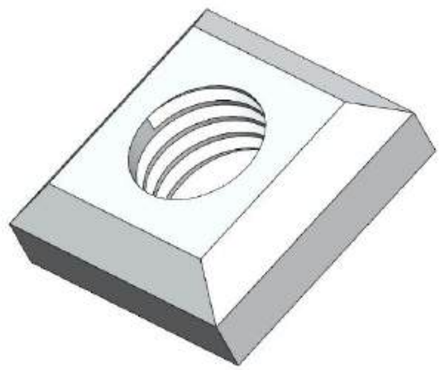


Figure 15. Square Nut

The geometry of these parts allows for penetration welding, ensuring a strong and durable connection. As shown in Figure 16, the surface of the square nut must be flushed with the surface of the square tube. This alignment ensures optimal clamping force when the mechanism is mounted. To achieve a rigid welding, the square nut must be made from the same material as the square tube. In this case, both are made of 304 stainless steel.

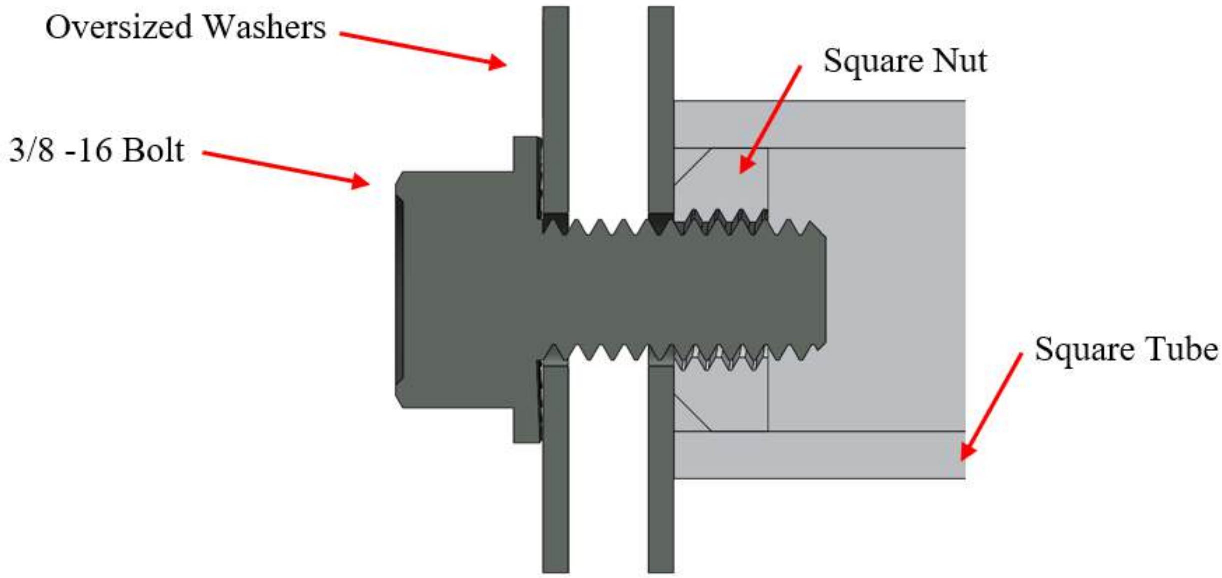


Figure 16. Section View of Square Tube Assembly

Washers are also visible in this figure above. These are 2.5 inch diameter and 0.068 in thickness. washers are oversized to enhance the clamping force. Additionally, the washers contribute to the modularity of the design. Adding more washers increases the overall length between the two hooks, offering additional flexibility. As shown in the image, washers can be used on both sides of the front hook to ensure an even distribution of forces.

The bolt used is a grade 5 steel bolt with a diameter of 0.375 inches. The rated proof strength for the bolt is 85,000 psi. The proof load on the bolt is calculated as below:

$$F_p = A_t \times S_p = \frac{\pi \times (0.375)^2}{4} 85000 = 9388\ lb$$

The preload for nonpermanent connections, reused fasters is given by $F_i = 0.75 \times F_p$ and is found to be 7041 lb for the application. The stress and factor of safety is found below using the yield strength of 92000 psi for grade 5 bolts:

$$\sigma = \frac{F_i}{A_t} = \frac{7041}{0.1104} = 63750\ psi$$

$$FOS = \frac{S_y}{\sigma} = \frac{92000}{63750} = 1.4431$$

The next sub-assembly is the tube support assembly, which is a smaller sub-assembly composed of fewer components. This sub assembly is shown in Figure 17. The most critical part of this assembly is the tube support. This component is initially made from Onyx with carbon fiber infill, using a Markforged printer, as it is easy to prototype. However, for the final iteration, this part will be machined from aluminum. This change will ensure it can withstand the required forces needed to support the entire mechanism.

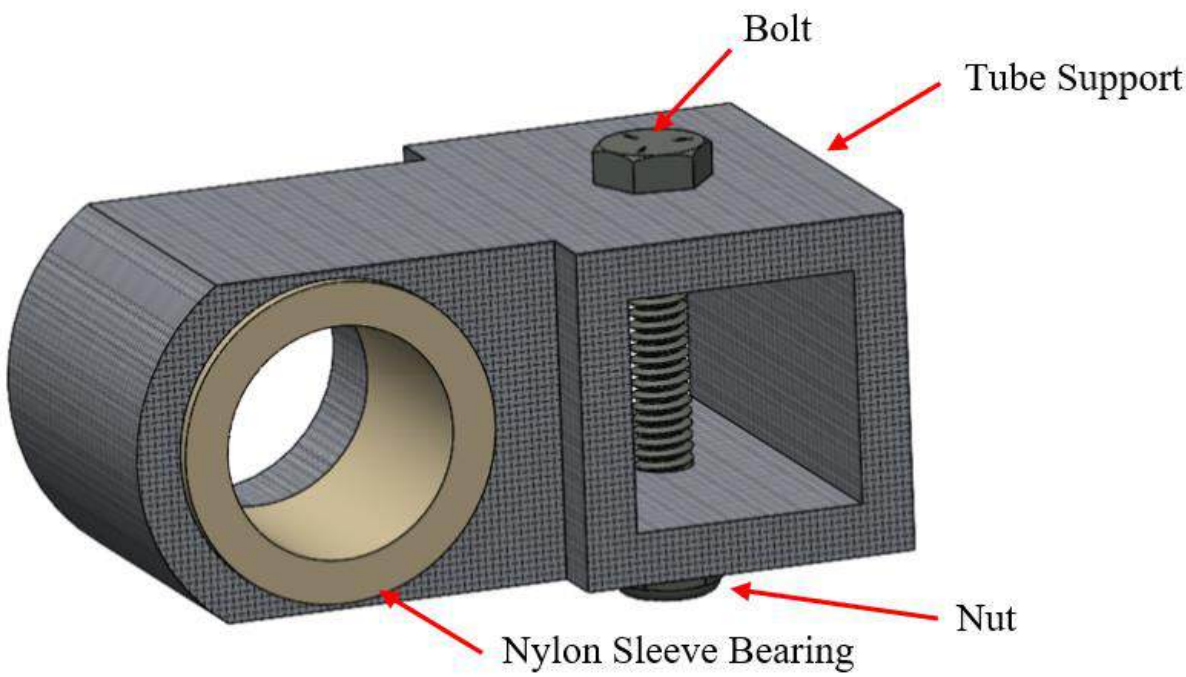


Figure 17. Tube Support Assembly

The tube support is connected to the square tube assembly with a bolt and nut, ensuring that these components remain securely in their intended position. Additionally, the sub-assembly includes a nylon sleeve that is inserted into the larger hole of the tube support. This sleeve allows the front arm to rotate freely without experiencing excessive friction.

For simplification purposes, the calculations for the front tube support are done as if there is only one support in the front, this is an overestimate of the stress one bracket would experience, however if it passes this calculation it would definitely pass for when it working within a pair. The half width of contact area for one front support tube is found as follows:

$$b = \sqrt{\frac{2F}{\pi L}\left(\frac{(1-\nu_1^2)/E_1 + (1-\nu_2^2)/E_2}{1/d_1 - 1/d_2}\right)} = 13.28$$

The calculation takes into consideration the dimensions and material properties of both the support tube and the front curved tube. The force is also multiplied by the maximum moment arm which would be when the front tube is fully extended out away from the pontoon. This is then used to calculate the contact stress as given below:

$$p_{\max} = \frac{2 \times F}{\pi \times b \times L} = \frac{2(3690)}{\pi(13.28)(1.5)} = 118\ psi$$

This calculation shows a stress substantially lower than the yield strength of the support tube's material, 6061 aluminum. For this reason, it is unnecessary to calculate a factor of safety.

Front Tube

The final sub-assembly in the front mechanism is the front tube assembly. It consists of a circular cross-section tube that connects, with the help of other components, the pontoon to the jet ski. This sub-assembly is connected to the others using the tube supports. As shown in Figure 18, a complete view of the sub-assembly includes several components, like a 6-foot circular tube, two pins, two quick release shaft collars, a U joint, and a shackle.

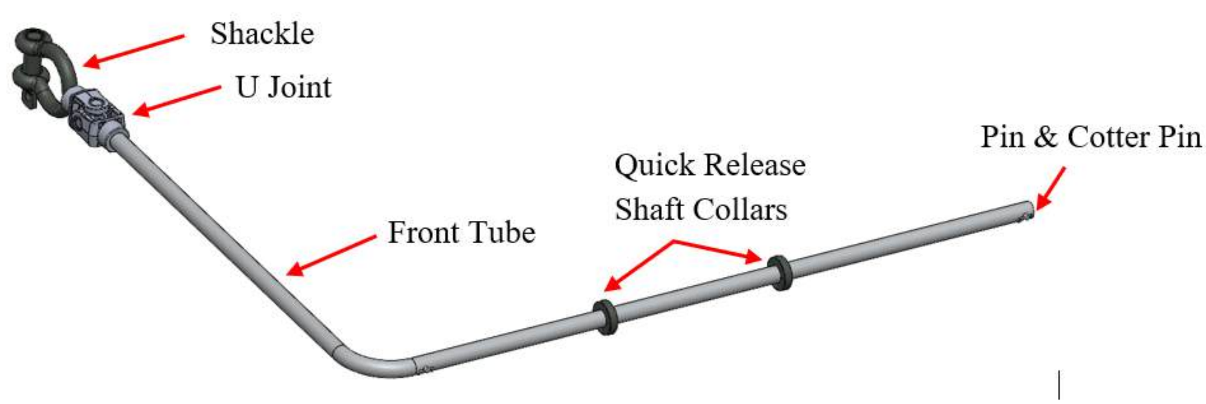


Figure 18. Front Tube Assembly

The chosen tube is made out of 304 stainless steel with 2 inch outer diameter and 1.5 inch inner diameter. This material and thickness are chosen from the following calculations:

The curved rods in front are identical, and so calculations of one side are done. The rod experiences a concentration of stress within its curved corners due to internal forces reacting to external forces on the entire attachment. Using superposition, the isolated stress while towing can be added to the isolated stress from wave dynamics. During towing, the rod experiences curved beam bending stress and shear stress, as seen in Figure 19.

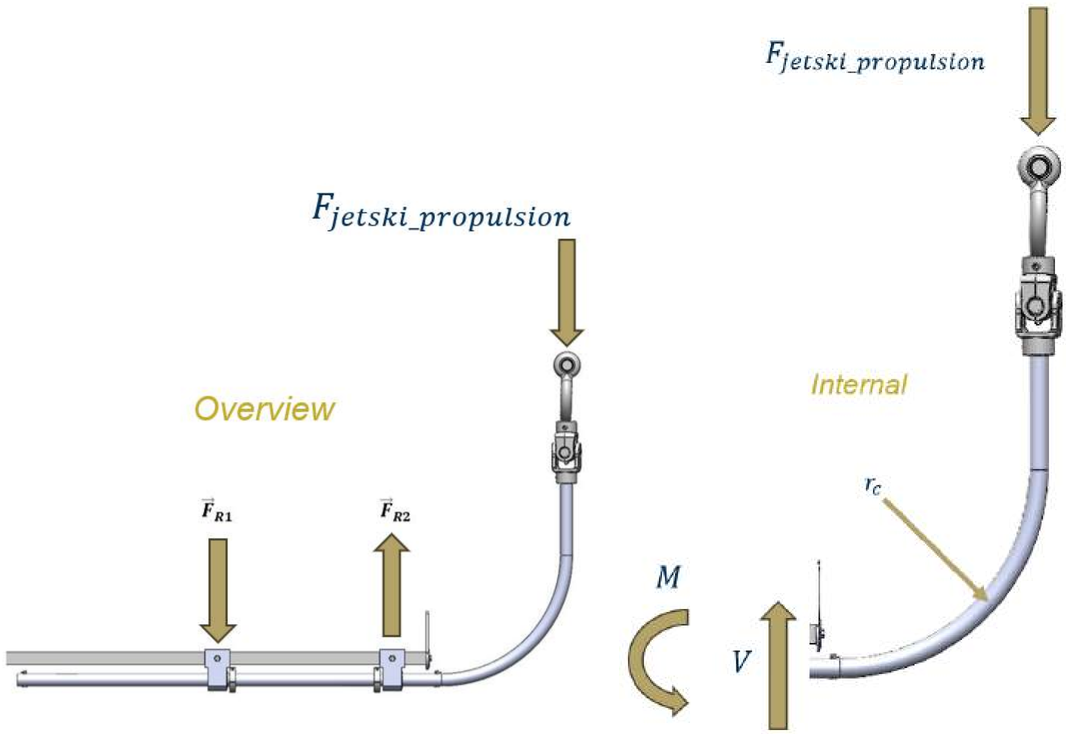


Figure 19. Front Rod FBD for Towing

For wave dynamics, the main difference in the internal forces is the change in direction of the applied force. This introduces axial stress in place of shear stress, depicted in Figure 20.

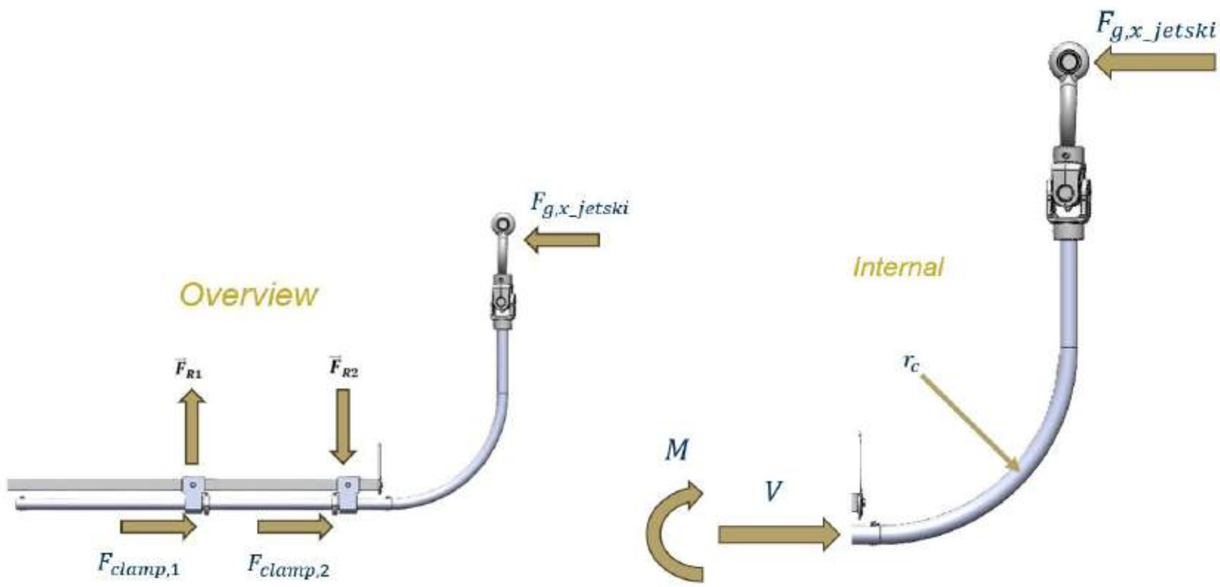


Figure 20. Front Rod FBD for Wave Dynamics

From towing the stress is as follows:

$$\sigma = \frac{2V}{A} + \frac{My}{I}\frac{r_C}{r}$$

This is the sum of shear stress for a hollow round beam and a modified version of the Flexure Formula that is applicable to curved beams, the latter is based on Figure 21, a reference available in Shigley's Mechanical Engineering Design Tenth Edition.

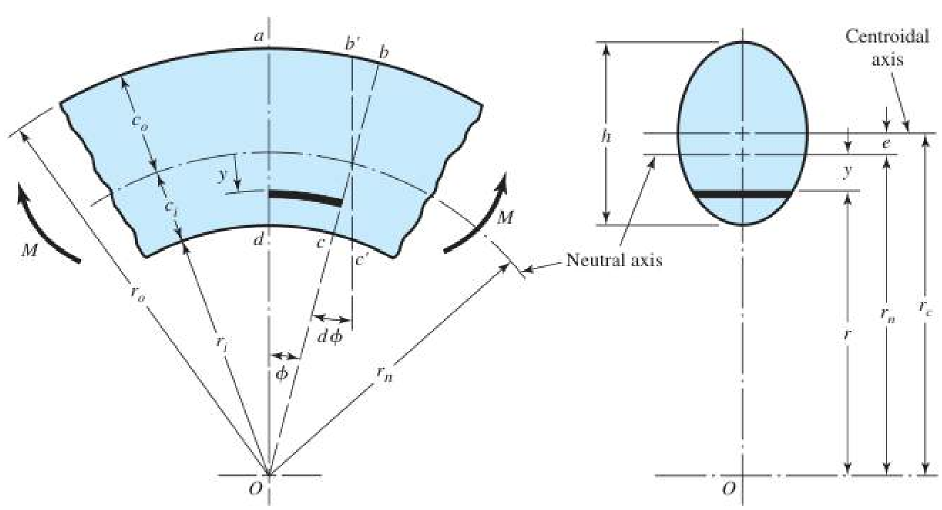


Figure 21. Curved Beam in Bending

This version of the Flexure Formula is also an approximation for large curves where e, the difference between the centroidal and neutral axes, is small. Where σ is the stress, M is the perpendicular distance of the forces from the jet ski to the end of the curve, r is the distance from the center of curvature to the point of interest within the cross section, y in this approximation is $r_c - r$,and I is the area moment of inertia of the hollow front beam with outer and inner diameters of 2 and 1.25 inches respectively. Shear stress is obtained as follows:

$$A = \pi(R_0^2 - R_i^2) = 1.9144$$

$$\tau = \frac{2(180)}{(1.9144)} = 188$$

In this context, the maximum stress is at the inner surface, when r equals r_i, here y would equal $r_c - r_i$ which is just R_o, the outer radius of the front rod. The moment arm while towing is 29.5 inches and the radius of curvature, r_c , is 3 inches. As such the bending stress is as follows:

$$M = Fr = (180)(29.5) = 5{,}310$$

$$I = \frac{\pi}{4}(R_0^4 - R_i^4) = 0.6656$$

$$\sigma_{bending} = \frac{(5{,}310)(1)}{(0.6656)}\frac{(4)}{(3)} = 11{,}967$$

The total stress from towing:

$$\sigma_{towing} = 188 + 11{,}967 = 12{,}155\ psi$$

This completes the stress experienced exclusively from towing. Axial stress replaces shear stress in the case of wave dynamics and is calculated as follows:

$$\sigma_{axial} = \frac{F}{A} = \frac{(123)}{(1.9144)} = 64$$

The bending stress from wave dynamics is obtained similarly to how the towing case's bending stress was calculated but with a moment arm of 28 and, since the front and back mechanisms work together to counteract wave dynamics, a force of 123 pounds; this results in the total stress due to wave dynamics as:

$$\sigma_{\omega ave} = 64 + 7{,}762 \ = 7{,}827\ psi$$

Finally, for a worst-case scenario the two isolated stresses are summed using superposition for a total stress on the front rod and to achieve a FOS above 2, steel, with a tensile yield strength of 50,800 psi, was chosen as the manufacturing material for the rod.

$$\sigma = 12{,}155 + \ 7{,}827 = 19{,}982\ psi$$

$$FOS = \frac{50{,}800}{19{,}982} = 2.54$$

A 2.54 factory of safety is substantially higher than 1, where 1 or less would indicate a design prone to permanent deformation. This means the rod may bend under stress, but it will return to its original orientation once the stress is alleviated. In fatigue, the endurance limit for steel used is given by $S_e = 0.5S_y = \ 25{,}400$ hence giving a factor of safety of 1.27 in fatigue as given below:

$$FOS = \frac{25{,}400}{19{,}982} = 1.27$$

Using Siemens NX analysis, the maximum deflection of the curved member under towing forces was calculated to be 0.164 inches, well within the permissible limit of 0.178 inches (L/360 for marine applications), offering a 7.8% safety margin. Figure 22 shows the finite element analysis showing a maximum deflection of 4.177 mm (0.164 inches), within the permissible limit for the curved member under towing forces. This ensures the member can withstand static and dynamic loads without excessive deformation, maintaining structural integrity during prolonged use. The high-strength steel provides sufficient tensile strength to avoid permanent deformation, even under worst-case loading scenarios. Additionally, the design’s corrosion resistance ensures durability in saltwater environments. Overall, the analysis confirms the curved member’s reliability for marine conditions, meeting all performance and safety requirements.

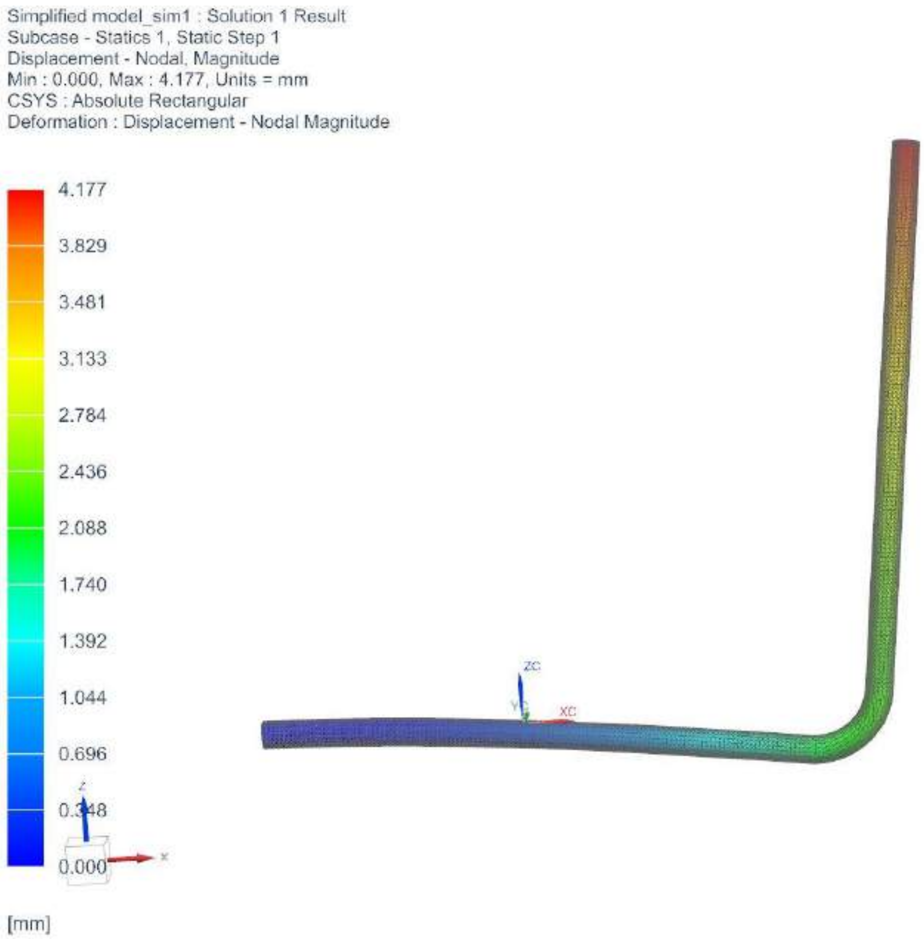


Figure 22. Siemens NX FEA model for Deflection

Quick-Release Clamps

The front sub-assembly features two adjustable quick-release clamps. These clamps lock the arm's movement axially, allowing it to rotate only vertically. This design accounts for waves and the relative movement of the jet ski and the pontoon. These forces will be transmitted due to the horizontal shift of the weight of the jet ski caused to the tilt angle. This force can be calculated as below:

$$F_x = F_{grav\_jetski} \times \sin(30) = 275lb$$

A visual representation is provided in Figure 23.

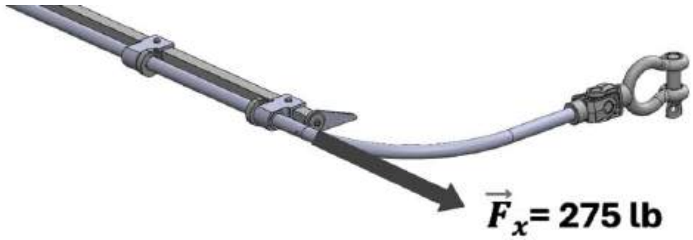


Figure 23. Force on Pins Transmitted During Slippage

The clamps make use of ¼"-28 alloy socket head cap screws, which are rated for 1510 lbs of axial holding strength according to the industry standard. The forces the clamps experience are well within its holding capabilities as validated below:

$$FOS = \frac{1510}{275} = 5.49$$

Safety Pins

In case of clamp failure or user error in securely tightening the clamps, the tube includes two pins as a safety precaution to prevent the entire sub-assembly from disconnecting from the main mechanism. To prove that the pins will be able to handle the worst-case forces, the stresses on them need to be analyzed. The pins will experience direct shear and contact stresses. The direct shear on the pin is calculated using this load below:

$$\boldsymbol{\tau} = \frac{V}{4 \times A} = 896 \text{ lb}$$

This is a considerably small stress that can be easily handled by the pin and what is more significant is the contact stress must the pins endure. The contact area of the stress is calculated below:

$$b = \sqrt{\frac{2F}{\pi L}\left(\frac{(1-\nu_1^2)/E_1 + (1-\nu_2^2)/E_2}{1/d_1 - 1/d_2}\right)} = 0.0377$$

This is then used to calculate the contact stress as given:

$$p_{\max} = \frac{2 \times F}{\pi \times b \times L} = 18599 lb$$

This is a much higher value and will dictate the factor of safety calculated below:

$$FOS = \frac{35000}{18599} = 2.9$$

With the high factor of safety of 2.9, the pin is assured not to fail in operation. As the loads being transferred to the pin is not the standard practice and is instead meant as a fail-safe measure, fatigue calculations are not required.

U-Joint

At the very end of the tube, the sub-assembly includes a U-joint. This U-joint allows slight movement in all directions, helping to mitigate the forces applied to the mechanism. It is welded to the tube to ensure a rigid connection. On the opposite end, the U-joint is welded to a shackle, as shown in Figure 24.

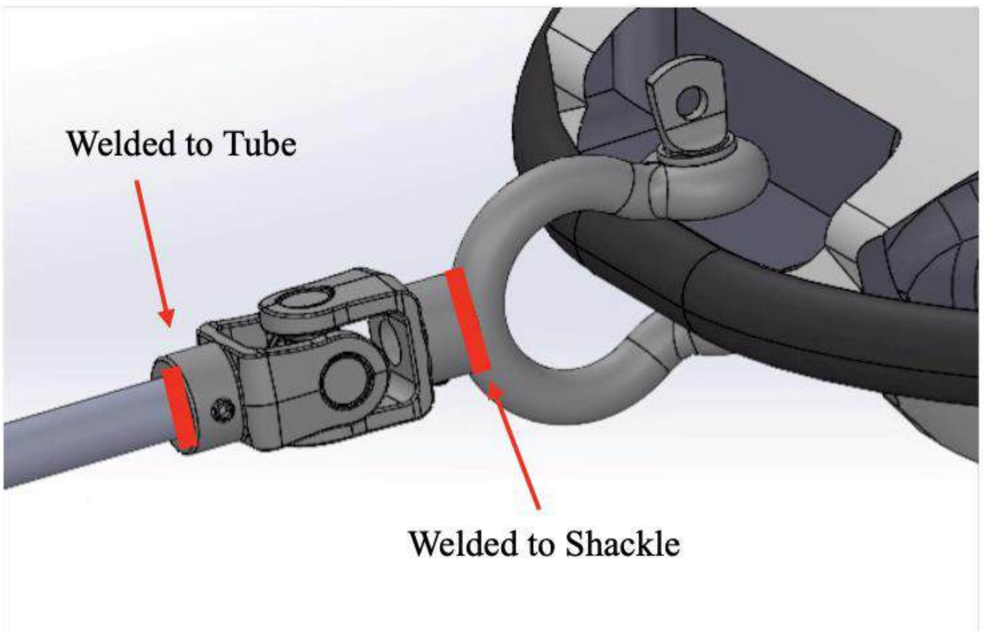


Figure 24. Shackle welded to Universal Joint

The shackle is used in this model because it fits the sponsor's jet ski. If this product were to be mass-produced, an alternative method of attachment would involve a smaller shackle designed to fit the front hook of various jet skis like the one shown in Figure 25.

Figure 25. Universal front hook of jet ski

Back Attachment

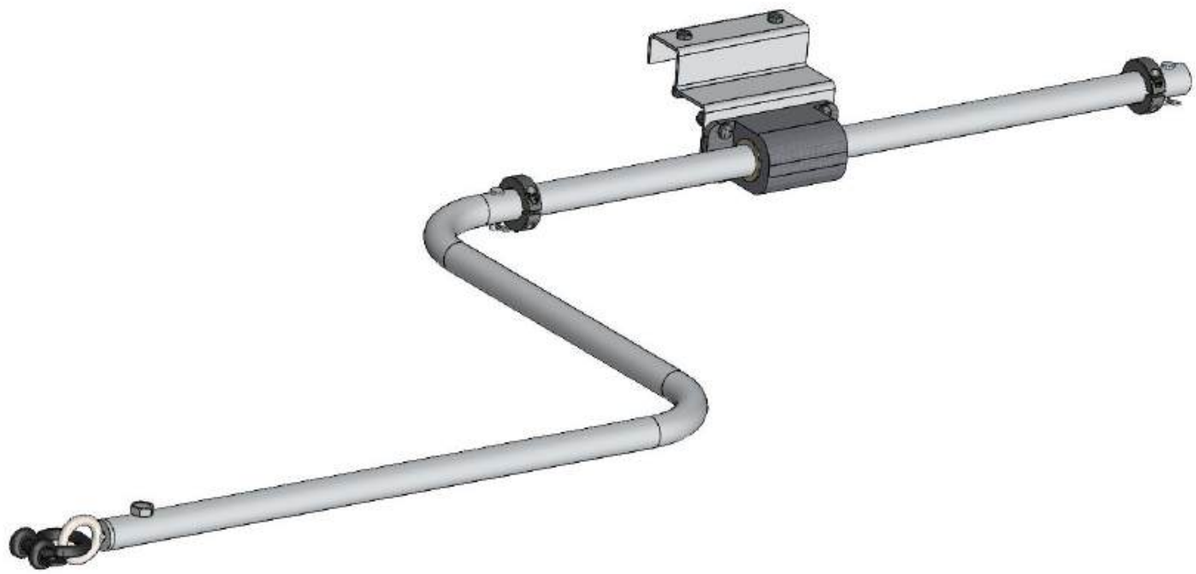


Figure 26. Back Attachment Assembly

The front attachment handles the towing of the jet ski, but it works in tandem with the back attachment, seen in Figure 26, to limit motion due to wave dynamics. This ensures the jet ski does not sway away from or into the pontoon. The mechanism is not designed to constrain forward and backward movement; that responsibility is taken solely by the front mechanism. The back attachment consists of two parts: the pontoon back connection and the S-Rod.

Pontoon Back Connection

The pontoon back connection is the structure that connects to the pontoon. As shown in Figure 27, the mounting rack fits perfectly. This part is made from 0.1-inch-thick aluminum 5052, which was selected over 6061 aluminum because it is more suitable for bending. The mounting rack is secured in place with two screws that pass through the pontoon flooring, creating a rigid connection. The mounting rack is also capable of supporting the back tube support, this part is constraint with four screws that pass through a thin sheet metal layer on the pontoon to add rigidity to the model.

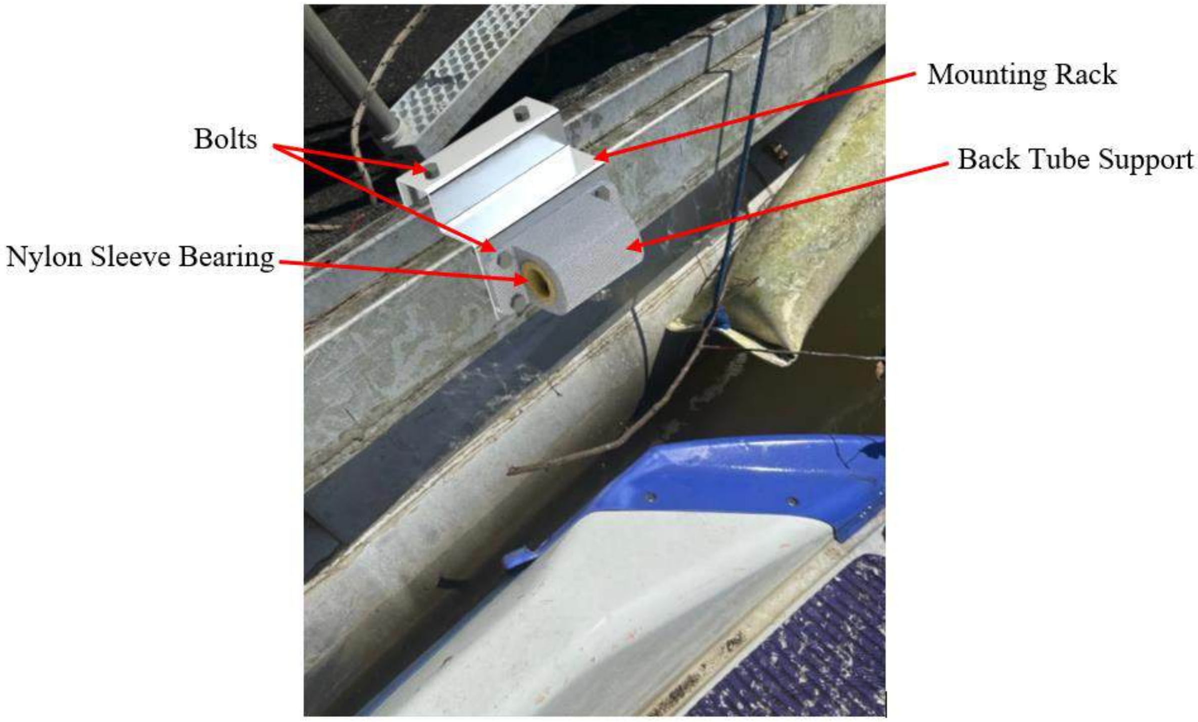


Figure 27. Back Connection Rendered on Pontoon

The back tube support is made from Onyx with carbon fiber, produced on a Markforged printer. However, the final iteration will use machined aluminum. The half width of contact area for the back support tube is found as follows:

$$b = \sqrt{\frac{2F}{\pi L}\left(\frac{(1-\nu_1^2)/E_1 + (1-\nu_2^2)/E_2}{1/d_1 - 1/d_2}\right)} = 19.94$$

The calculation takes into consideration the dimensions and material properties of both the support tube and the S-Rod. This is then used to calculate the contact stress as given below:

$$p_{\max} = \frac{2 \times F}{\pi \times b \times L} = \frac{2(7380)}{\pi(19.94)(3)} = 79\ psi$$

This calculation shows a stress substantially lower than the yield strength of the support tube's material, 6061 aluminum. For this reason, it is unnecessary to calculate a factor of safety.

Two nylon sleeve bearings are included to allow the tube to move forward and backward without experiencing too much friction.

S-Rod Subassembly

The S-Rod is the second sub-assembly and consists of the bent tube along with smaller parts that securely attach one side to the pontoon back connection and the other to the back of the jet ski. Figure 28 shows the sub assembly.

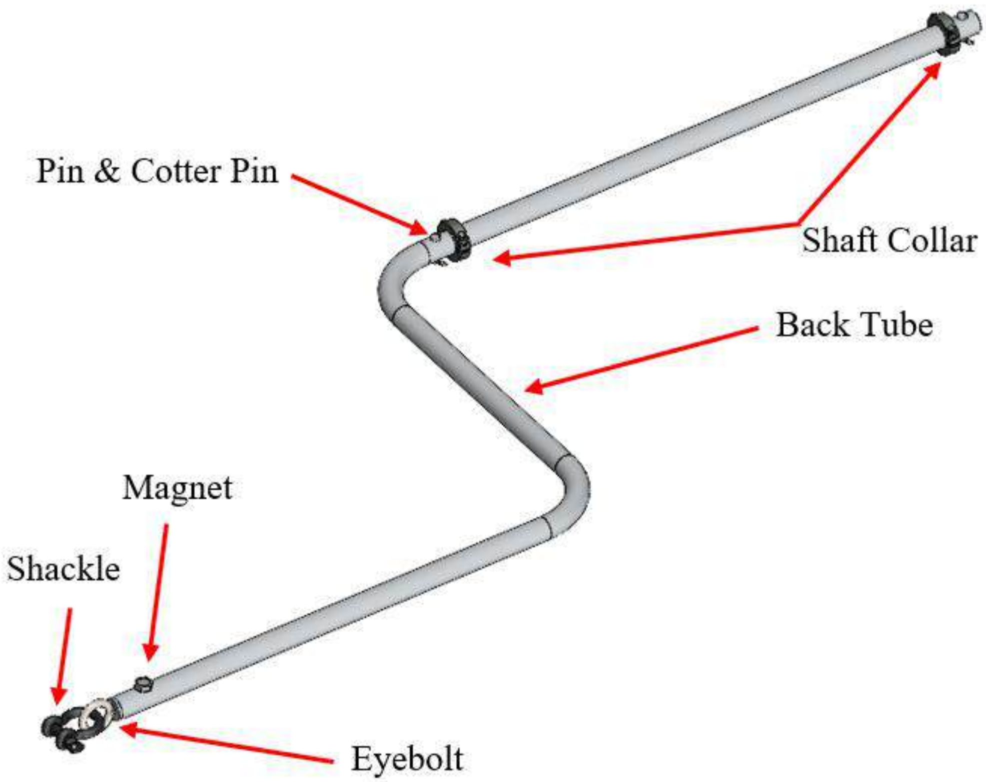


Figure 28. Back Tube Subassembly

Similar to the front tube, the S-Rod is 304 stainless steel. The S-shaped rod experiences stress comparable to what the front mechanism’s curved rod bears, as seen in Figure 29, and it is made of the same material, so the calculations are similar.

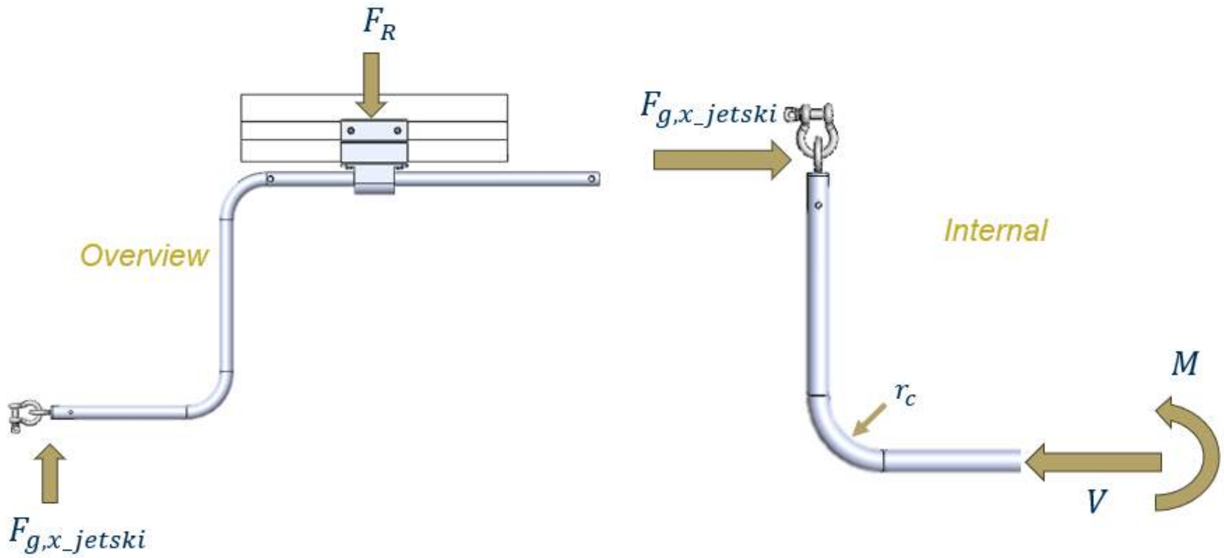


Figure 29. S-Rod FBD

The exceptions are the moment and dimensions of the rod. The moment arm and inner and outer diameters of the rod are 31, 1.5, and 1 inch; the moment and overall stress calculations follow:

$$M = Fr = (123)(31) = 3813$$

$$\sigma = \sigma_{\omega ave} = 125.3 + 19{,}121.5 \ = 19{,}247 \, psi$$

$$FOS = \frac{50{,}800}{19{,}247} = 2.64$$

The back mechanism is thus guaranteed to not deform during peak stress as it is less than half the material's yield strength. With this, all components of the design never dip below a factor of safety of 2.5, this represents a design with high durability and longevity with the absence of permanent deformation. In fatigue, as discussed earlier, the endurance limit for steel used is given by $S_e = 0.5S_y = \ 25{,}400$ hence giving a factor of safety of 1.32 in fatigue as given below:

$$FOS = \frac{25{,}400}{19{,}247} = 1.32$$

Using Siemens NX analysis, the maximum deflection of the back mechanism was calculated under towing forces. For a span length of 72 inches, the permissible deflection for marine applications is determined as L/360 = 72/360 = 0.2 inches. The analysis revealed a maximum deflection of 0.184 inches, which is well within the allowable limit, providing a 8% safety margin. This confirms that the back mechanism maintains structural integrity under the applied loads, preventing excessive deformation. The analysis validates the design's ability to endure towing forces and wave-induced stresses while ensuring durability in marine environments.

The tube is held in place by two shaft collars, enabling the bar to move while restricting it to the desired distance. In case of failure, two pins placed on either side act as a safety backup to prevent the arm from detaching from the back connection. The pins have been proven to match safety requirements in earlier sections.

At the very end of the circular tube is a ring of the same diameter as the tube, designed to accommodate an eye hook. Both the eye hook and the ring are threaded to ensure a perfect fit and to hold the required forces. The forces here are minimal because most of the pulling force is handled by the front mechanism. The use of a shackle connects the eye hook to the back hook of the jet ski, as shown in Figure 30.

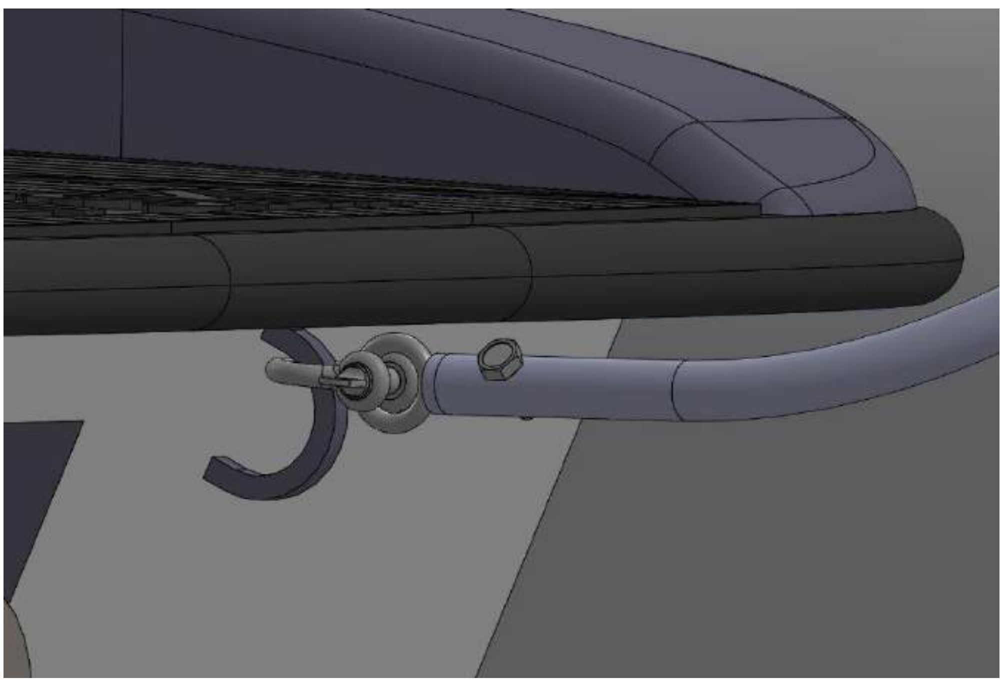


Figure 30. Shackle Connecting Jet Ski and Back Mechanism

When the mechanism is in the rest position (not in use), a magnet is used to secure the arm to a magnetic connection point on the pontoon. For the final iteration, this will likely be replaced with a quick-release strap. The strap will allow the design to be more modular, accommodating different types of pontoons.

User-Friendly and Safety Features

To address the customer requirements of ease of use and prevention of loss of pieces, multiple measures were taken in the design. There are parts such as the pins, bolts and shackle bolts which must be detached from the mechanism during use, assembly and disassembly. The shackle bolts which connect to the jet skis have to be unscrewed once in the water to detach the jet ski for use. This might happen in deep water and thus the user risks dropping and losing the piece. To prevent these, all detachable parts are connected to the main frame of the mechanism using stainless steel wires secured via crimped end caps, as seen in Figures 31, 32 and 33.

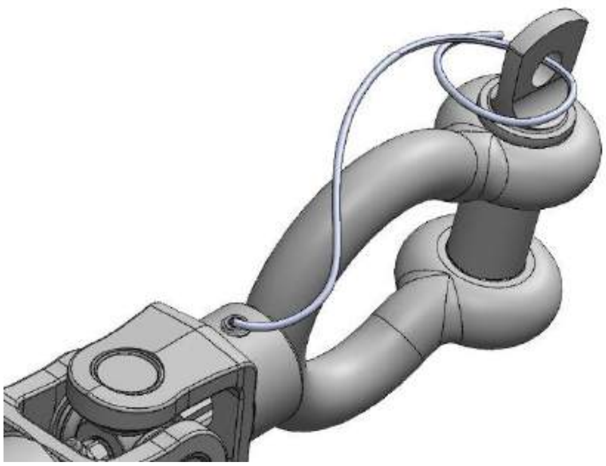
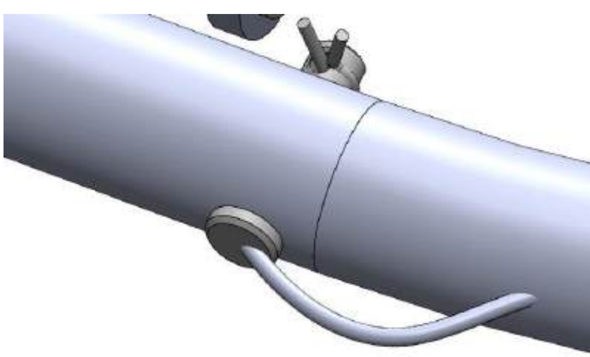


Figure 31. Wire attaching shackle bolt to main assembly.



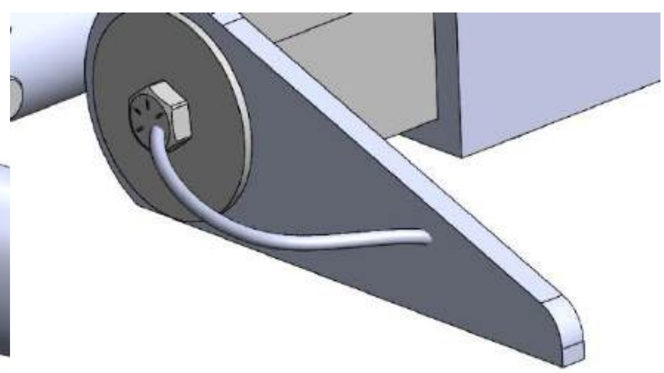


Figure 32. Wire securing pin.

Figure 33. Wire securing bolt.

Additional safety features are incorporated such as pins at the end of tubes to prevent them from detaching from the mechanism completely. The pins in the towing mechanism play a crucial role in enhancing safety by ensuring secure attachment of critical components while preventing unintended detachment during operation. Each pin is equipped with tethering wires made of corrosion-resistant stainless steel, which keep them permanently connected to the main frame even if accidentally dropped or loosened. This design eliminates the risk of losing essential parts in deep water, where retrieval could be challenging. Additionally, the pins are secured with locking mechanisms, such as safety clips, to ensure they remain firmly in place under dynamic loads and vibrations caused by towing and wave impacts. This dual-layer of protection—physical tethering and secure locking—provides a reliable and user-friendly solution that reduces the likelihood of mechanical failure and ensures uninterrupted operation in recreational and emergency scenarios.

Prototype and Manufacturing

The prototyping phase began after finalizing the design. The most critical task was bending the circular tubes. The machine available for this task had a fixed 3-inch radius, which constrained the overall design. Figure 34 shows the machine used on the school premises.

Figure 34. Tube Bending Machine used.

Figure 35 displays the bent tube. The final dimensions can be found in the fabrication package. For prototyping, we chose aluminum because it is easier to work with than steel. Bending the tube proved to be relatively straightforward.

Figure 35. Bended Front Tube

Once the tube was deemed functional, the tube support was 3D-printed using the Markforged printer. Figure 36 illustrates the 3D-printed part placed in the square tube. This part includes a nylon sleeve to reduce friction and is secured to the square tube with a nut and bolt.

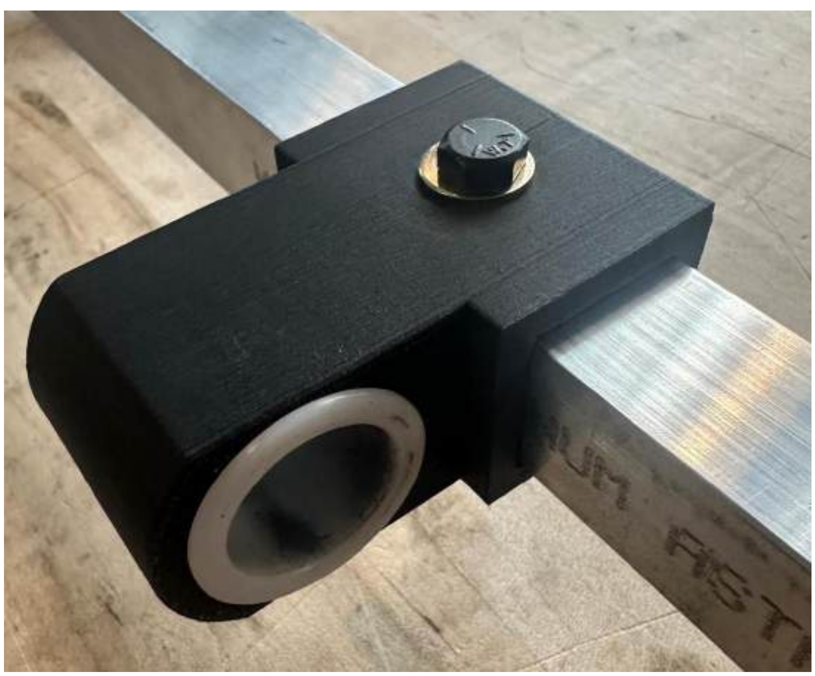


Figure 36. Tube Support Placed on the Square Tube

Figure 37 demonstrates how the front mechanism will be mounted onto the pontoon. The gray 3D-printed part represents the pontoon's front hook.

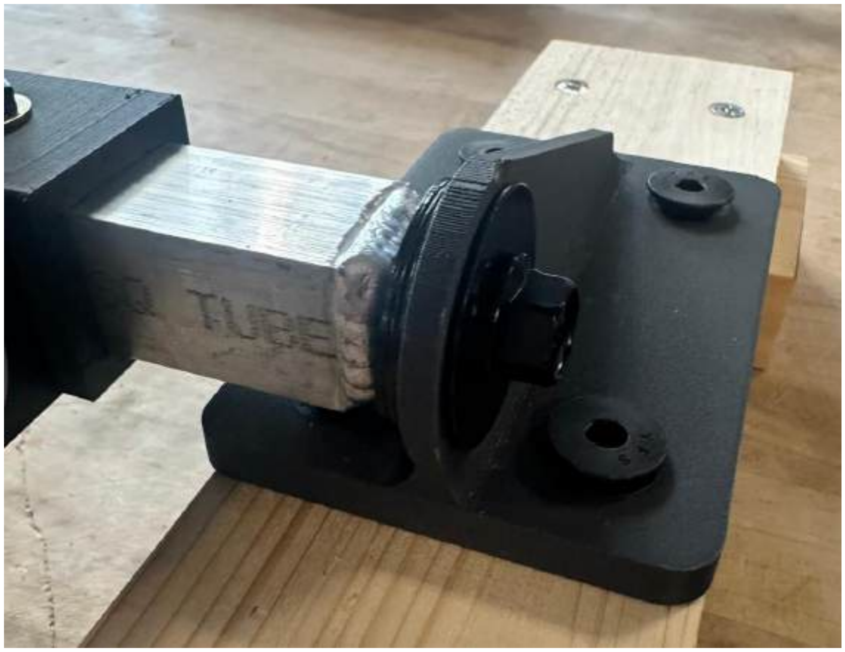


Figure 37. Front Mechanism Connection to Pontoon

As shown, the square nut is welded to the square tube. In the final prototype, the penetration welding finish must be improved. The two surfaces need to be coincident to allow the washers to generate maximum clamping force. Figure 38 provides an example of how this should appear in the final iteration. This part was made by water jetting the square pattern with the inside hole and then threaded to fit the bolt size.

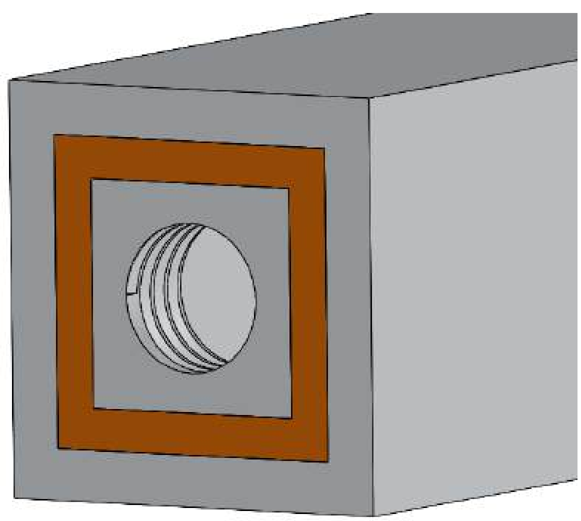


Figure 38. Future Welding Fitment

A shackle was welded to one side of the U-joint, as shown in Figure 39, while the other side was bolted to the round tube.

Figure 39. D-Shackle welded to Universal Joint

Figure 40 gives a broader view of the front mechanism, highlighting a thin metal cable that prevents the pins from falling into the water. The quick-release clamps are also visible in this image.

Figure 40. Front Mechanism Assembly

The arm can move freely up and down, while the quick-release sleeves restrict axial motion. In case of an emergency, the pins prevent the front tube from disengaging from the mechanism.

The back mechanism was prototyped after finalizing the front mechanism. The back tube was bent in a similar manner to the front mechanism's tube. Figure 41 illustrates how the mounting rack connects to the pontoon. This component was manufactured by cutting sheet metal with a waterjet and then bending it into the desired shape. The mounting rack is secured to the pontoon using two long bolts that pass through the flooring.

Figure 41. Pontoon Back Connection

In the figure, the tube support is also visible. Similar to the front tube supports, this part was 3D-printed on the Markforged printer with carbon fiber infill. However, for future iterations, it would be preferable to machine this part out of metal to ensure greater durability and resistance to failure. The design also incorporates a nylon sleeve to reduce friction. The tube support is held in place with nuts and bolts, which also secure the mounting rack to the pontoon.

Figure 42 provides a wider view of the prototype, showing the use of shaft collars and safety pins to prevent the mechanism from coming apart. The pin holes were drilled to ensure a precise fit for the pins.

Figure 42. Back Mechanism Assembly

Figure 43 focuses on the end of the tube, which connects to the jet ski. The circular ring at the end was cut using a waterjet and welded onto the tube. An eye hook was threaded and inserted into the tube. A shackle is attached to connect the mechanism to the jet ski. The image also shows a stainless-steel cable, which, similar to the front mechanism, prevents the shackle bolt from falling into the water.

Figure 43. D-Shackle connection to Jet Ski

Overall, the prototyping phase provided valuable insights into the feasibility of mass-producing this mechanism. While the materials and dimensions used in this phase are not final, they allowed us to better understand the functionality of the mechanism and identify areas for improvement to enhance its security and reliability in the final design.

Current Limitations and Future Improvements

While the design is robust, satisfies all requirements and is validated by rigorous analysis, there are still limitations to it and thus scope for improvement. The current limitations, future improvements and next steps are discussed below.

One of the major restrictions of the design is that its focused-on jet skis and pontoons that are similar in geometry and attachment points to Ken's. The pontoon must have front hooks to attach the square tube to and the jet skis must have slots in the front to fit the D-shackle. Since the range of watercrafts available is so vast, this was a necessary narrowing of scope to create a reliable design. However, an improvement to the current design would be to adapt it and make it modular to fit a larger range of jet skis and pontoons.

Similarly, the back tube is secured in the retracted position (when not in use) via a magnet since the chosen pontoon has a magnetic connection point. However, all pontoons do not have magnetic points of connection such as the sponsor's does. To increase the universality of the design, a method of securing such as a strap could be considered instead of a magnet.

The design for the front and back mechanisms were all done with tubes that had a 1 inch outer diameter for ease of fabrication with the resources on hand, however, over the course of performing stress and fatigue calculations, it was determined that achieving a factor of safety of 2 would improve the life of the product and make it capable to withstand an infinite number of wave cycles, which is why the front tubes were changed to have a 2 inch outer diameter and the back tubes were changed to have a 1.5 inch outer diameter.

Another restriction applied is limiting the use of the attachment to waterbodies where the maximum amplitude of the waves can reach five feet. While it has a Factor of Safety above 2.5 for the scope of the project, it must be improved to enable it to function in rougher conditions if it can be scaled to wider usage since most oceans have waves that can have higher amplitudes.

A possible improvement to increase the durability of the design is to include a damping system at the connection point between the front tube and the jet ski. The inclusion of a shock absorber was considered initially, but the geometrical constraints made it difficult to incorporate. Since the mechanism could withstand forces within the defined scope, it was ultimately ruled out. However, for further improvements a shock absorber might aid in extending the scope of usage.

In the case of an emergency, the mechanism enables the jet skis to tow the pontoon to safety. Currently, the coordination of the speed and direction of the jet skis is reliant on communication between the boat operator and jet ski drivers. To improve this, a control system could be integrated such that both the jet skis operate on the same speed and the combination of their directions steer the pontoon as desired. An additional safety feature would be the incorporation of a kill switch which would immediately switch off both engines.

The next step would be to take the product to the market for consumers to order directly. For prototyping, singular parts were ordered from McMaster, due to which the cost was very high. To increase the feasibility of mass manufacturing, these parts would instead be ordered in bulk and custom parts would be machined, thus decreasing the cost. This would make the product scalable and at a competitive price point.

Lastly, the uniqueness of the design might qualify it as a patentable product. Future Capstone teams could make the proposed improvements, and the final product could then be patented and put on the market.

Conclusion

The towing mechanism effectively addresses critical customer requirements and engineering specifications, ensuring dual towing capacity for two jet skis while maintaining a factor of safety above 2.5. The design guarantees durability and regulatory compliance for use in marine environments. It also provides an emergency towing capability for pontoons, fulfilling the need for operational safety. With a modular structure, the mechanism allows assembly by a single user in under 20 minutes, and the use of corrosion-resistant materials ensures durability under harsh conditions and prolonged exposure to the elements. Engineering specifications, such as wave adaptability for up to 5-foot waves, ±30° tilting and rolling, and towing forces up to 320 lbf, directly informed the structural and material choices. Flexible joints and reinforced components address the dynamic forces from towing and wave motion, while finite element analysis validated that deflections remain within acceptable limits for both the front and back mechanisms. Compact and modular assembly ensures seamless aesthetic integration, aligning with customer preferences for functionality without compromising the visual appeal of the watercraft. The most important design constraints, including compatibility with existing pontoons and jet skis, adherence to maritime safety standards, and operation in lakes, ensured the final product is reliable within the defined scope. This design demonstrates how careful consideration of engineering requirements, iterative testing, and modular construction can create a reliable and scalable solution. These lessons provide a valuable framework for future projects requiring similar safety, durability, and adaptability in marine applications.