Hobie Cat Kayak

Final Paper
April 29, 2009

Team Leader

Matthew Ricciardi

Technical Liaison

Brian Back

Purchasing Agent

Ryan Wackerly

Webpage Specialist

Zach Walker

Faculty Advisors

Dr. Mohamed Hefzy

Dr. Mehdi Pourazady

Client Advisors

Ms. Jill Caruso

Dr. Chris Beins

Sponsors

National Science Foundation

Ability Center of Toledo

Table of Contents

List of Figures & Tables

Nomenclature

Mirage Drive – Alternative kayak propulsion system that uses leg driven pedals to allow hands-free kayaking. Adjustable mechanism built with injection molded plastics, anodized aluminum, and stainless steel fittings.

Hobie Cat – A large kayak that uses the Mirage Drive for propulsion instead of the traditional kayak paddle.

Foils – A device consisting of a flat or curved surface that reacts to the water as it is passing through. Similar to fins of a fish or wings of a bird.

Computational Fluid Dynamics (CFD) – Computerized solution procedure that uses numerical methods and algorithms to solve and analyze problems that involve fluid flows to determine forces, drag, etc.

Free body diagram (FBD) – A pictorial representation of a physical system used to analyze the forces acting on free bodies within the system.

AutoCAD – A 2D computer aided design (CAD) program used to generate free body diagrams and fabrication drawings.

MathCAD – Computer program used to perform force and stress calculations based on free body diagram.

Spring scale – Hand held device used to apply a measured input force to the input arm of the Mirage Drive. Used to determine maximum applied force for use in stress calculations.

Von Mises stress – The von Mises stress is used to predict yielding of materials under any loading condition from results of simple uniaxial tensile tests.

Finite Element Analysis (FEA) – A meshed model of a material or design that is stressed and analyzed for specific results.

ProEngineer – A 3D parametric CAD program. Used to generate kinematic animation and check fitment of all components drawn in AutoCAD.

Solidworks / COSMOSWorks – A 3D parametric CAD and FEA program. Used to perform FEA displacement and von Mises stress analysis.

Executive Summary

The objective of the project is to design and fabricate an adaptive system to transfer energy to propel the Mirage Drive System of a Hobie Cat Kayak. The Mirage Drive System propels a kayak by using two oscillating foils instead of the traditional kayak paddle by transferring energy from the user's legs to the foils. The project adapts the Mirage Drive System to transfer energy from the user's arms to the foils. This mechanism allows any user, with at least one functional arm to efficiently propel the kayak with a simple push and/or pull motion and reduces the minimum requirements for independent kayaking to just one functional arm.

Introduction

Independent kayaking and fishing is a healthy, enjoyable and recreational activity that millions of people around the world enjoy. Unfortunately, hundreds of thousands of people with limited use of their legs and arms are unable to enjoy independent kayaking and fishing. The Mirage Drive System is a third generation propulsion system that allows people with limited use of their arms to propel a Hobie Cat Kayak. The Hobie Cat website listed in the References section of this paper has an excellent description of the Mirage Drive System with pictures and videos to further explain the mechanism's operation[1].

The Mirage Drive System essentially transforms the back and forth motion of a user's legs to a forward thrust via two oscillating foils mounted through the bottom of the kayak into the water. These foils have been modeled from analyzing the flippers of penguins and tuna and then refined with computational fluid dynamic (CFD) analysis. The Mirage Drive generates three times more thrust per stroke than a traditional kayak paddle with surprising ease. The Mirage Drive has been proven as a revolutionary device, but still leaves behind anybody without use of their legs. The purpose of this project is to develop a mechanism that attaches to the Mirage

Drive to allow anybody with at least one functional arm to propel the kayak. There are currently no similar products available that provide similar functionality to the proposed design. A one-armed kayak paddle exists for traditional kayaks, but requires complex motion, transmits very little force, and decreases the stability of the kayak in operation. The value that this project can provide to disabled users far exceeds the small cost of design, materials, and fabrication.

Team Roles

All group members were expected to make equal contributions to the completion of the project, and all group members understand the time and effort requirements needed to meet these goals. Each group member chose a specific role based on his individual talents. Matt Ricciardi is the team leader and is responsible for coordinating communication and meetings between all parties. He will also continually evaluate group progress and establish intermediate goals to meet deadlines. Brian Back is the technical liaison. Brian has a strong fabrication background and access to machining resources. Ryan Wackerly is the purchasing agent. Ryan has worked in heavy industry maintenance and has experience purchasing from major suppliers. Zach Walker is the webpage specialist. Zach has extensive experience with designing, hosting, and maintaining small to medium scale web applications for small businesses and individuals.

Project Description

The goal of the project is to develop a mechanism to transfer power from the user's arms to the Mirage Drive System of a Hobie Cat Kayak. All of the proposed mechanisms are simple, reliable, and cost effective. The project must not eliminate any functionality of the existing kayak nor the Mirage Drive System. This means that the user will be able to sit comfortably in the kayak with or without the mechanism installed. The mechanism must be simple and lightweight to minimize cost, maintenance, and transportation effort. The mechanism must be easy for the user to install and remove without the use of any special tools. Finally, all of the materials must be resistant to corrosion as well as continuous wear, as our operating environment includes both freshwater as well as saltwater.

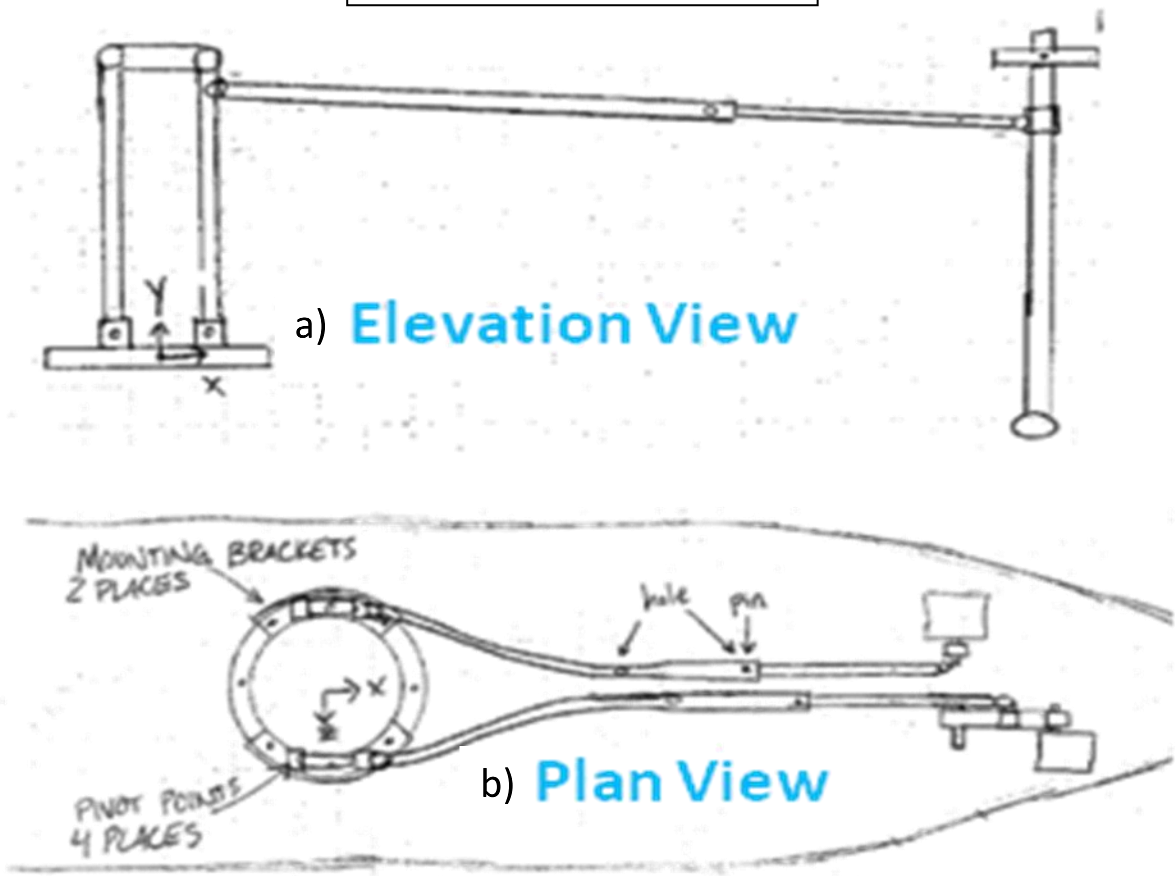
Proposed Designs

Three different designs were considered. The following is a detailed description of each of the three basic designs.

Design #1 – Two push pull rocker arms

This design consists of two rocker arms hinged at an access hatch just in front of the user's seat as shown below in Figure 1. These arms rock back and forth in opposite directions and are driven by a pushing and pulling motion from the user's arms. The rockers are connected to the arms of the Mirage Drive by two independent bars.

Figure 1: Rocker Mechanism

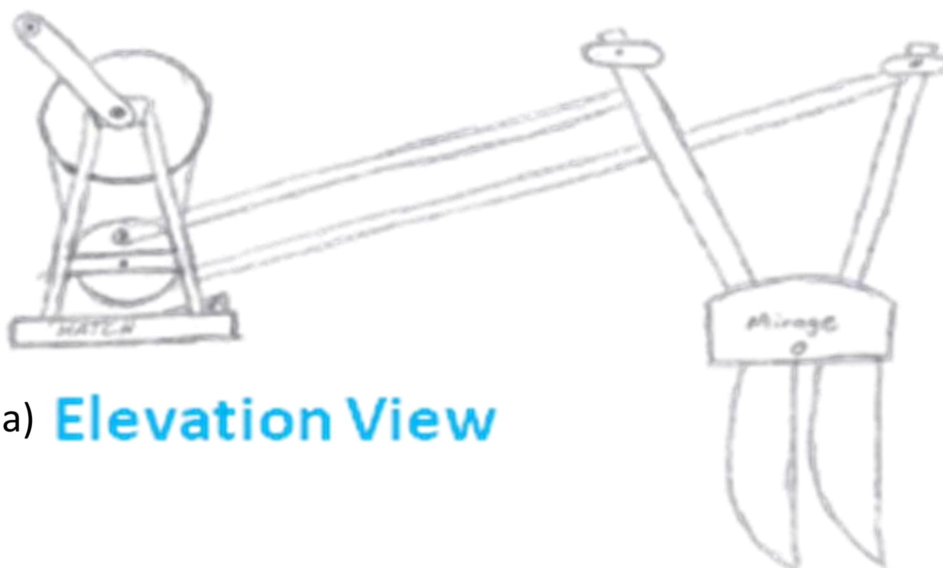


This system’s greatest advantage is the transmission of maximum force with the best path of motion. This design will cause the user the least amount of fatigue and will generate thrust at maximum efficiency (and highest kayak velocity).

<u>Design #2 – Hand crank</u>

The second design also employs the same four bar linkage principle, but replaces the two rocker arms with a crank –connecting rod linkage to the Mirage Drive arms as shown below in Figure 2. The crank would make a complete revolution for each 60 degree stroke of the Mirage Drive arms. The crank would be similar to the hand cranks already in use on bicycles for the disabled.

Figure 2: Hand Crank Mechanism



a) Elevation View

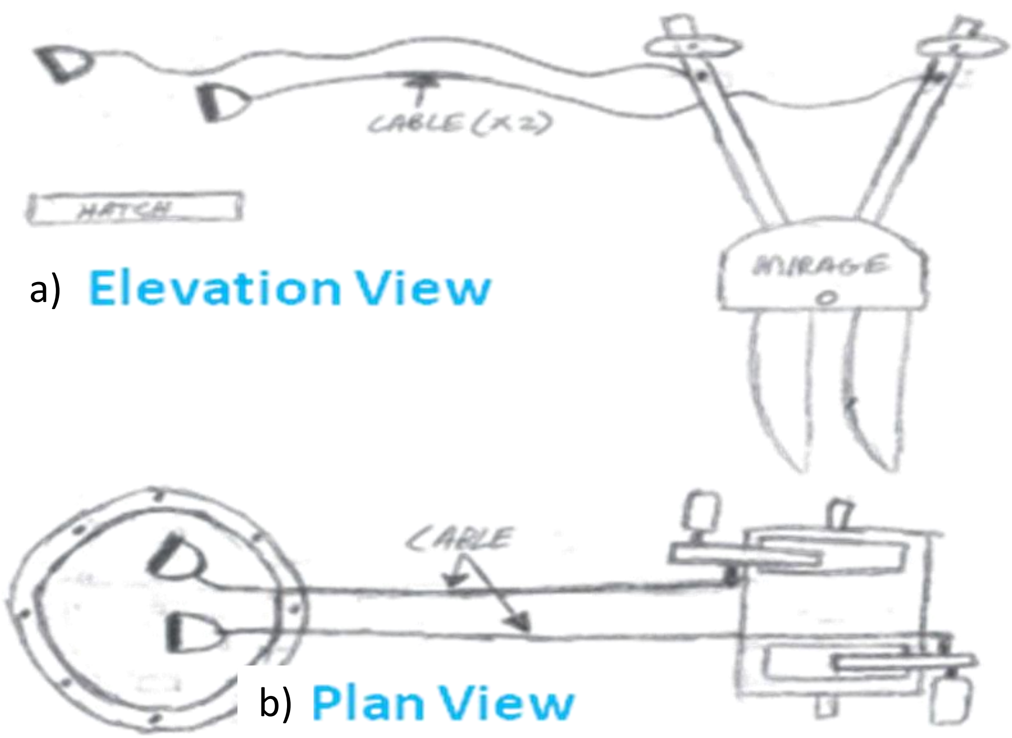
b) Plan View

This design would allow the use of commodity parts that are easily changeable by the user. A rotary motion might be easier for certain disabilities as well. This system would be the heaviest and most complex, with most of the weight above the center of gravity of the kayak, possibly increasing the tendency to roll the kayak. This design also introduces the requirement for at least one sealed bearing which adds cost, weight, and complexity.

<u>Design #3 – Direct cable</u>

The third design is the most simple. It consists of two cables, one tied to each of the Mirage Drive's arms as shown below in Figure 3. The other end of each cable attaches to a handle. The user pulls one cable at a time, with the pulling motion of one cable returning the opposite arm in preparation for the next pull. The cables can have changeable handles and variable lengths.

Figure 3: Direct Cable Mechanism



An obvious disadvantage of this design is the exclusive use of a pulling motion for propulsion. This cuts the amount of power supplied per stroke roughly in half. However, a very significant advantage is the extreme simplicity of a cable, handle, and clamp on the Mirage Drive. The use of cables also requires the user to provide an optimum motion for each pull without a fixed path like the previous two designs. This lack of a fixed path coupled with the pulling motion would cause much more fatigue as compared to the previous two designs. The simplicity of this system makes it the easiest for the user to install and remove. This design, however, would exclude one-armed users from being able to propel the kayak, as there would be no means to return the opposite arm of the Mirage Drive. This is obviously a significant disadvantage.

Design House of Quality

A house of quality was used to compare the three designs and select the best basic mechanism. The design house of quality is shown below as Table 1.

Table 1:
House of Quality for Basic Design

	Importance	Rocker (Design 1)	Crank (Design 2)	Cable (Design 3)
Safety	5	4	4	5
Light Weight	4	4	3	5
Cost	3	4	3	5
Original Function	2	5	3	5
Ease of Use	4	5	3	3
Adaptable	5	5	4	1
Durable	5	4	3	4
Score		123	94	107

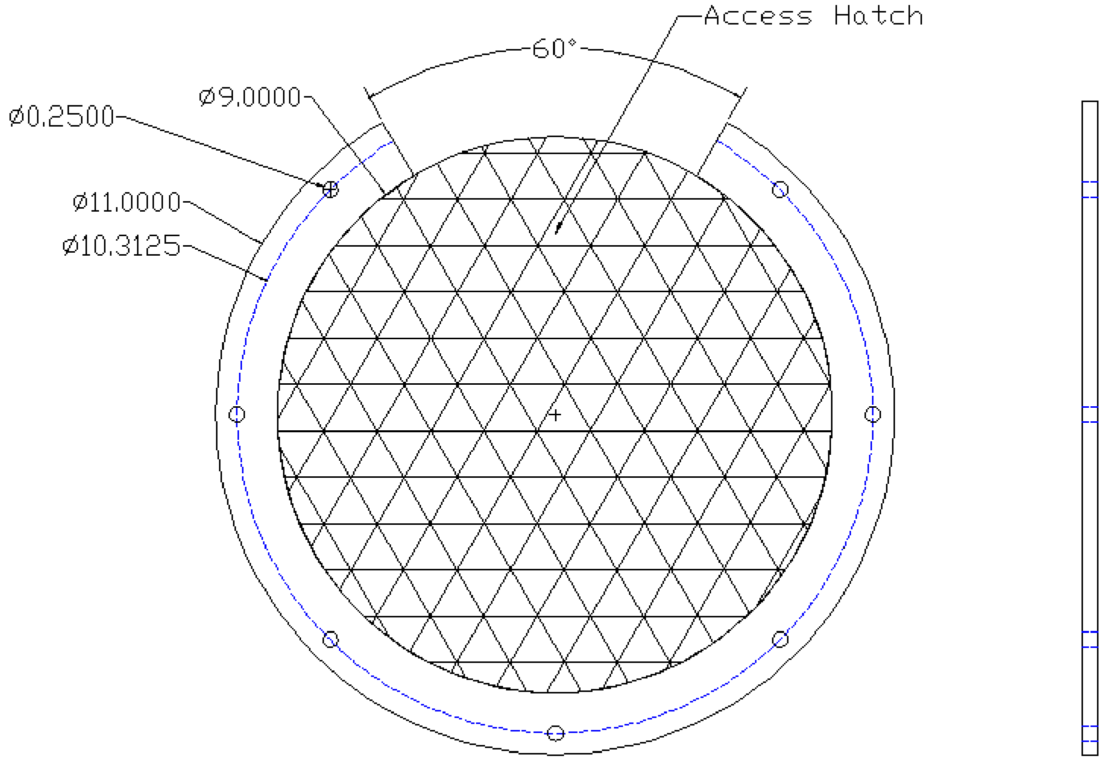
The house of quality above uses an importance factor of 5 for the safety category as recommended by faculty advisors. The original function category has a low relative importance factor since the client asked to focus on maximizing functionality of the mechanism and would accept eliminating some existing function if necessary. Assigning a rating of 1 for Adaptability for the Cable Design is due to the fact that the design fundamentally excludes many potential users. Separate houses of quality will be compiled as needed throughout detailed design of our project. The most likely candidate for a house of quality is the material selection process, as the environment can very corrosive in certain situations, specifically, using the kayak in saltwater.

Design and Analysis

Based on knowledge of mechanical systems and client requirements, a simple four bar linkage was found to be the best solution for the presented problem. Weight, adaptability, and functionality were basic premises for the design guidelines of this project. It is important that the design does not have a negative impact on the steering, stability, and safety of the kayak. The design must also meet space constraints inside of the kayak without interfering with different sizes and shapes of users.

The mechanism consists of three beams connected by pin joints that transmit the input force from the user to the Mirage Drive. Hobie Cat Kayaks include a storage hatch directly in front of the operator's seat that provides a solid mounting surface. The first draft design proposed mounting directly to the hatch cover because of simplicity, but this would eliminate the use of the access hatch therefore reducing the original functionality of the kayak. It was then decided to use a much stronger aluminum ring as a base plate to mount around the access hatch. This is accomplished by using the existing mounting holes for the hatch to secure the base plate. The base plate has an outer diameter of 11 inches and an inner diameter of 9 inches. A drawing of the new base plate is shown below in Figure 4. The plate is circular in shape, approximately 300 degrees of a circle. Design limitations arose due to the fact that there is only a 1 inch wide surface to mount to and approximately 6 inches of clearance between the ring and the inner thighs of the user.

Figure 4: Base Plate



After the parameters for the four bar linkage were determined, the connecting rod and rocker arm for the prototype could be created. Prototyping allowed for inspection of clearances as well as functional limitations with respect to the user and kayak. Scrap steel was readily available and used for building the prototype. The prototype connecting rod was built with several holes at each joint to allow testing of multiple positions for each member. This range of adjustment allowed testing to find the optimum handle height and range of motion. A simple bracket was also designed to mount the connecting rod of the prototype to the Mirage Drive input arms. The mounting bracket was constructed with three different pivot points for testing multiple configurations as well.

Prototype Design and Testing

Once the prototype was completed and assembled, it was installed on the kayak. Prototype testing was then performed on the Maumee River in downtown Toledo. The different prototype configurations were used to operate the kayak by members of the group.

From this, the optimum length of each link was decided to be used in the final design. Testing was also done to measure the required input force F_{in} to the Mirage Drive. Tests were executed with the kayak stationary and moving in the water by using a spring scale attached to the Mirage Drive to apply the input force. The absolute maximum force generated during all tests was measured as well as the force required to maintain a moderate pace. The results of the test are shown in Table 2 below.

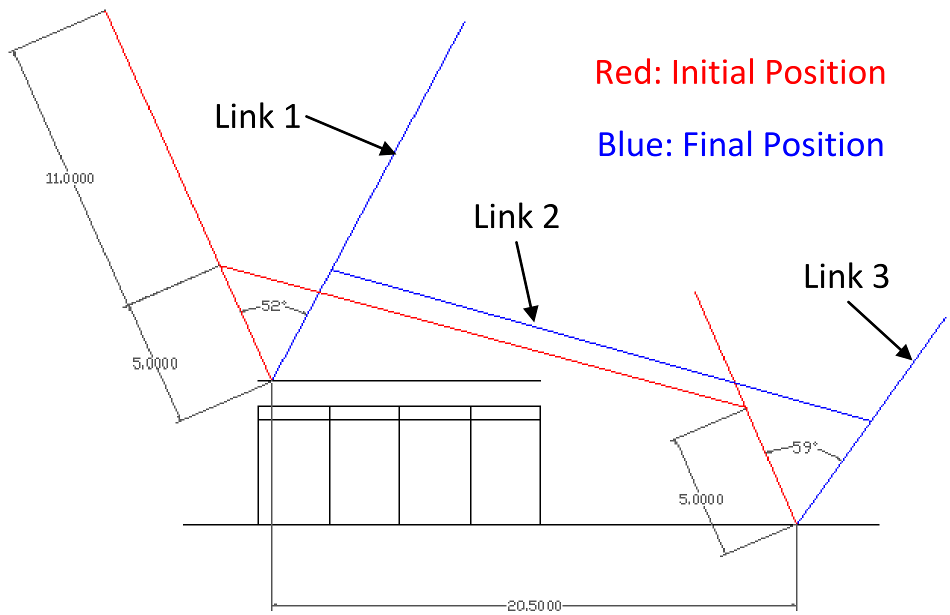
Table 2 – Prototype Test Results

Prototype testing on Maumee River, 26-Mar-2009					
	Stationary			Moving	
	Typical	Maximum		Typical	Maximum
Test 1	21	75		20	67
Test 2	18	65		23	70
Test 3	22	82		17	76

After discovering that the maximum input force was 82 pounds, it was decided that the original assumption of 100 pounds for typical operation would be increased to 150 pounds, for a built-in minimum safety factor of approximately two. Once this data was obtained, the group modified all calculations thus far to include the new load input. This new load did not force a change in the main components of the design, but it was decided to revise the smaller components that had lower factors of safety. This resulted in increasing the thickness of the three mounting brackets used. By basing the calculations on actual loading as opposed to the previous theoretical loading, a more realistic analysis was able to be conducted.

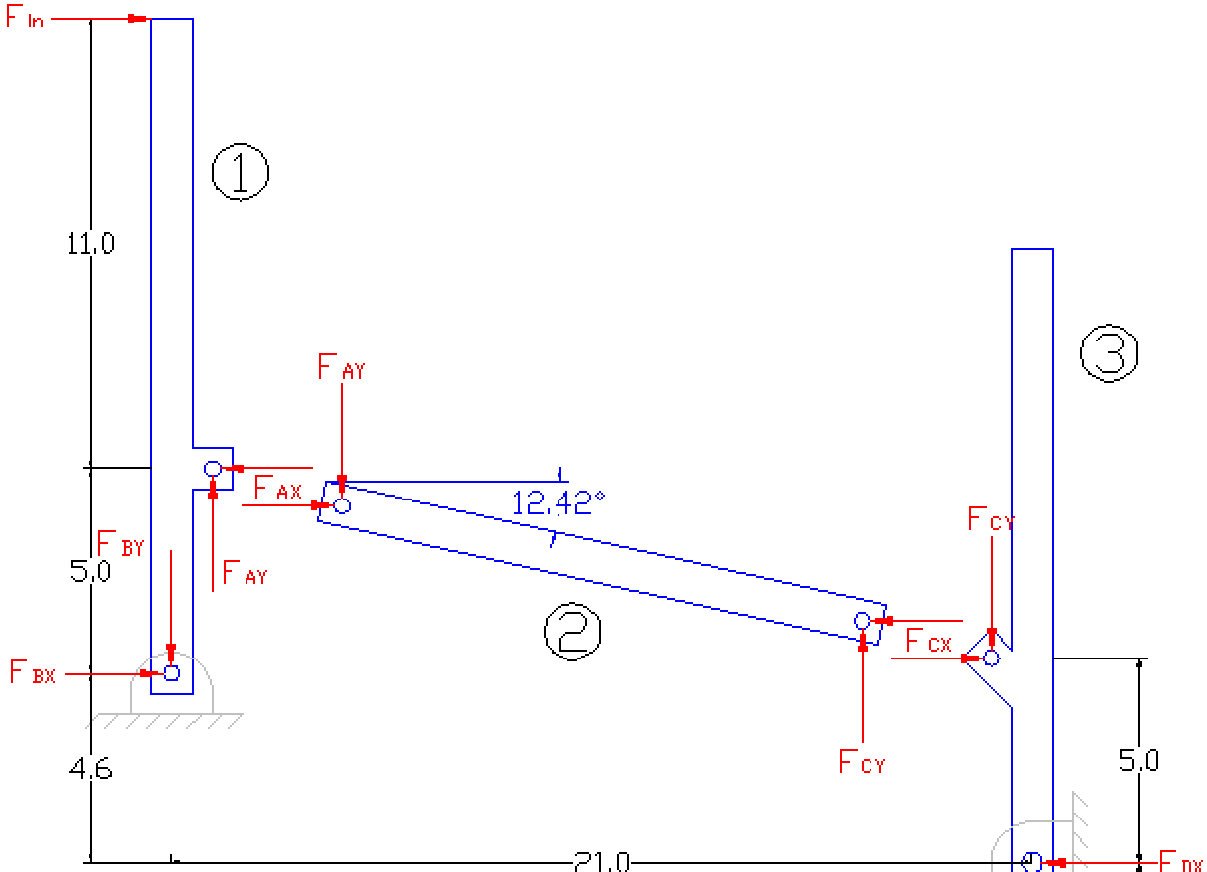
Parameters of the four bar linkage were determined by an analysis using AutoCAD. The dimensions of the Hobie Cat Kayak were measured and then used to construct a representative sketch. Several scenarios were drawn in AutoCAD using different lengths for each link, multiple starting positions, and different ranges of motion. Figure 5 shows one such scenario below. Link 1 of the mechanism is considered the rocker arm, while link 2 is referred to as the connecting rod. The existing Mirage Drive system is represented as link 3, and the body of the Kayak is the ground link of the four bar linkage.

Figure 5: Parameter Analysis



A free body diagram was constructed to determine the reaction forces in each member. This analysis was used in determining the stresses for all components of the design. The input force F_{in} was determined by testing the Mirage Drive on the water. Figure 6 shows the free body analysis.

Figure 6: Free Body Diagram



$F_{in} := 150\,lbf$

$a := 5in \quad b := 11in \quad c := 4.625in \quad d := 21in \quad e := 6in \quad f := 5in$

$F_{ax} := \frac{F_{in} \cdot (a + b)}{a}$ $\quad F_{ax} = 480\,lbf$

$F_{bx} := F_{ax} - F_{in}$ $\quad F_{bx} = 330\,lbf$

$F_{cx} := F_{ax}$ $\quad F_{cx} = 480\,lbf$

$F_{ay} := \frac{F_{ax} \cdot c}{d}$ $\quad F_{ay} = 105.714\,lbf$

$F_{cy} := F_{ay}$ $\quad F_{cy} = 105.714\,lbf$

$F_{by} := F_{ay}$ $\quad F_{by} = 105.714\,lbf$

$F_{h2o} := \frac{F_{cx} \cdot a}{e}$ $\quad F_{h2o} = 400\,lbf$

$F_{dx} := F_{cx} + F_{h2o}$ $\quad F_{dx} = 880\,lbf$

$F_{dy} := F_{cy}$ $\quad F_{dy} = 105.714\,lbf$

Material House of Quality

A second house of quality was compiled to aid the material selection process. The selection process included materials commonly used in similar mechanisms. The group further limited the materials to include only those that are commonly available, have sufficient material properties, are easy to machine, and are relatively inexpensive. The material house of quality is shown below as Table 3.

Table 3:
House of Quality for Material Selection

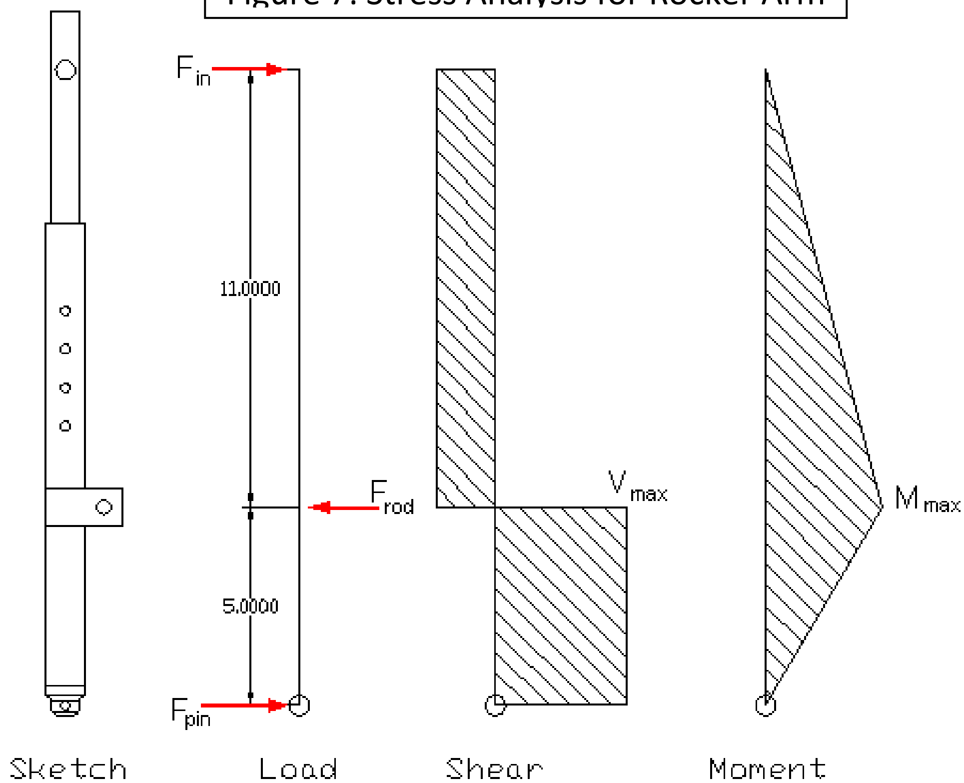
	Importance	Aluminum	Stainless Steel	Steel	High Strength Plastic
Safety	5	5	5	5	5
Light Weight	4	4	3	3	5
Cost	3	4	3	5	3
Weldability	3	4	5	5	0
Manufacturer's Use	5	5	0	0	5
Corrosion Resistant	5	4	4	2	5
Strength	5	4	4	5	3
Score		130	101	102	119

Aluminum turned out to be the best overall material for the project. This house of quality result agreed with the Mirage Drive manufacturer's material selection. Advisor feedback about the last house of quality was incorporated to change the importance factor of safety to 5. Even though the importance level was raised to 5, proper sizing, machining, and coating should maintain all materials at a rating of 5 for safety. Light weight is an important quality since the project will add weight to a relatively light kayak. The center of mass of the project is more important than net weight, since a high center of mass will decrease the roll stability of the kayak. The cost, strength, and corrosion resistance of the materials listed also varies greatly depending on heat treating processes and alloys, so this stage of design is focused more on a class of materials rather than a specific grade. Rather than building a third house of quality to determine the final grade of aluminum we will use, purchasing agent research has indicated that the best aluminum alloy to use is Aluminum 6061 T6511. This alloy best satisfies all of the design requirements of the system.

Mechanism Design and Analysis

From mechanical design experience of the group members, it was decided that box tube would be an acceptable application for the rocker arm. It was modeled as a simply supported beam with two pin joints and one point load to perform a stress analysis. Figure 7 shows the relevant calculations for this analysis. Reaction forces from the free body diagram were used to construct shear and moment diagrams to aid in determining the shear stress and bending stress in the member. By using MathCAD, different lengths, widths, and geometries could be quickly analyzed. This quick analysis proved helpful when experimental loads turned out to be higher than the first predicted loads. From the analysis, it was determined that 1 inch box tube with a wall thickness of 0.125 inches could be used for the top section of this application with a fatigue safety factor of approximately 8.5, and a 1.5 inch box tube with a wall thickness of 0.125 inches could be used for the bottom with a fatigue safety factor of approximately 8.

Figure 7: Stress Analysis for Rocker Arm



Section A

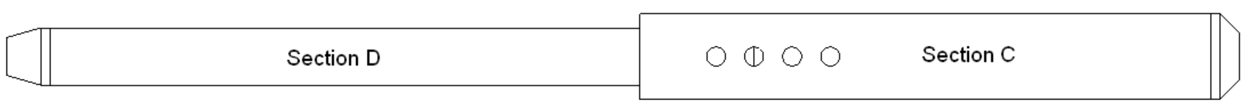
Cross sectional Area	$A_A := (1.5in)^2 - (1.25in)^2$	Moment of Inertia	$I_A := \frac{(1.5in)^4}{12} - \frac{(1.25in)^4}{12}$
Max Shear Force	$V_{maxA} := F_{pin} = 330 \cdot lbf$	Max Moment	$M_{maxA} := V_{maxA} \cdot 5in$
Shear Stress	$\tau_A := \frac{V_{maxA}}{A_A} = 480 \cdot psi$		
Bending Stress	$\sigma_{bA} := \frac{\lvert M_{maxA} \cdot .75in \rvert}{I_A} = 5665.57 psi$		
Combined Stress	$\sigma'_A := \sqrt{3\tau_A^{\ 2} + \sigma_{bA}^{\ 2}} = 5726.25 psi$	Factor of Safety	$n_A := \frac{Sy_{alum}}{\sigma'_A} = 7.86$

Section B

Cross sectional Area	$A_B := (1in)^2 - (.75in)^2$	Moment of Inertia	$I_B := \frac{(1in)^4}{12} - \frac{(.75in)^4}{12}$
Max Shear Force	$V_{maxB} := F_{in} = 150 \cdot lbf$	Max Moment	$M_{maxB} := V_{maxB} \cdot 4in$
Shear Stress	$\tau_B := \frac{V_{maxB}}{A_B} = 342.86 psi$		
Bending Stress	$\sigma_{bB} := \frac{\lvert M_{maxB} \cdot .5in \rvert}{I_B} = 5266.29 psi$		
Combined Stress	$\sigma'_B := \sqrt{3\tau_B^{\ 2} + \sigma_{bB}^{\ 2}} = 5299.66 psi$	Factor of Safety	$n_B := \frac{Sy_{alum}}{\sigma'_B} = 8.49$

An analysis for the connecting rod was also conducted to analyze the stresses within this beam. Round tube was chosen to be used for this member due to an anticipated lower stress. The calculations in Figure 8 below support this theory and show that a smaller diameter tube can safely be used, having a maximum stress of 1156 psi for the proposed design. Even with the conservative assumptions used in the calculations in Figure 8, the fatigue factor of safety is greater than 24, which is well above the specified minimum of 2 for this project. The material size used in these calculations is based on the actual size of tubing commercially available at the time of purchasing that met the telescoping requirement set by the client. These calculations are also conservative due to a lower factor of safety for the mounting plates.

Figure 8: Stress Analysis for Connecting Rod



Section C

Cross sectional Area $A_C := \frac{\pi}{4}\left[(1.3in)^2 - (1in)^2\right]$

Moment of Inertia $I_C := \frac{\pi}{64}\cdot\left[(1.3in)^4 - (1in)^4\right]$

Axial Stress $\sigma_C := \frac{F_{rod}}{A_C} = 885.73psi$

Factor of Safety $n_C := \frac{Sy_{alum}}{\sigma_C} = 50.81$

Critical Load for Buckling $P_{crC} := \frac{\left(\pi^2\cdot E_{alum}\cdot I_C\right)}{(20in)^2} = 23380lbf$

Factor of Safety $n_{crC} := \frac{P_{crC}}{F_{rod}} = 48.71$

Section D

Cross sectional Area $A_D := \frac{\pi}{4}\left[(1in)^2 - (.5in)^2\right]$

Moment of Inertia $I_D := \frac{\pi}{64}\cdot\left[(1in)^4 - (.5in)^4\right]$

Axial Stress $\sigma_D := \frac{F_{rod}}{A_D} = 814.87psi$

Factor of Safety $n_D := \frac{Sy_{alum}}{\sigma_D} = 55.22$

Critical Load for Buckling $P_{crD} := \frac{\left(\pi^2\cdot E_{alum}\cdot I_D\right)}{(20in)^2} = 11809lbf$

Factor of Safety $n_{crD} := \frac{P_{crD}}{F_{rod}} = 24.6$

Bearing Failure at Quick Release Pin

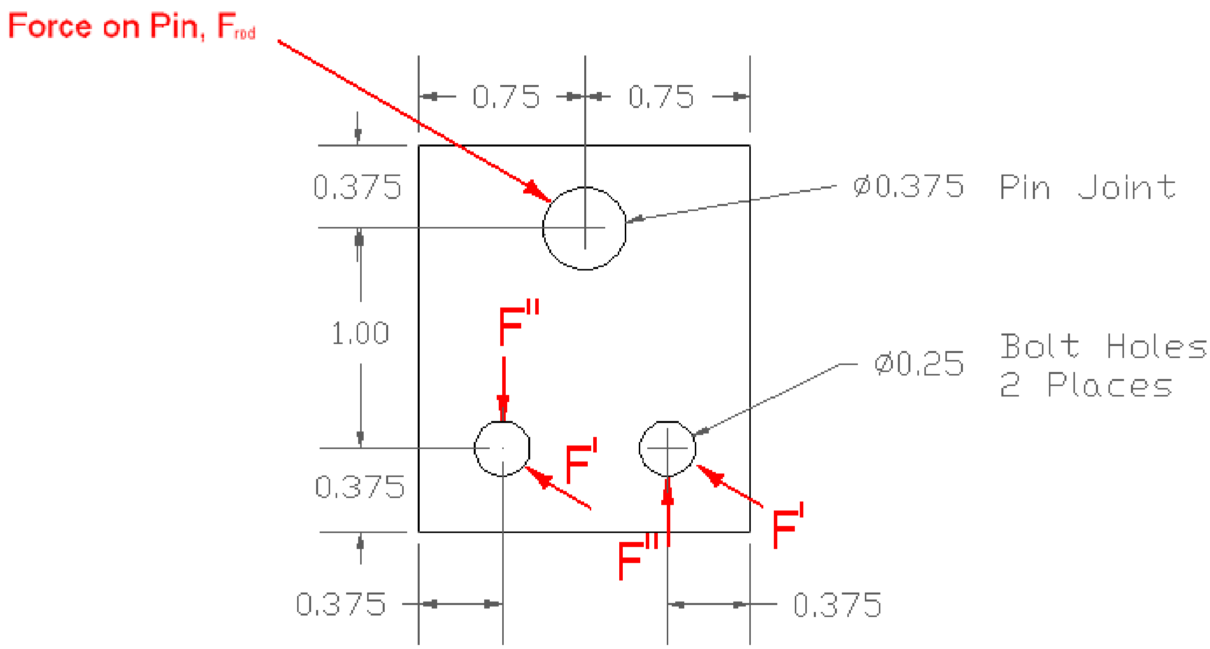
Projected Area $A_{CD} := D_{pin3}\cdot(1.3in - .5in)$

Bearing Stress $\sigma_{CD} := \frac{F_{rod}}{A_{CD}} = 1600psi$

Factor of Safety $n_{CD} := \frac{Sy_{alum}}{\sigma_{CD}} = 28.12$

The design and analysis for the three different connecting brackets followed the same process as for the main components (rocker arms and connecting rods). AutoCAD was used to design basic parts and hand calculations were used for stress analysis. An example of the first round of design and analysis for the brackets is shown below in Figures 9 and 10.

Figure 9: Stress Analysis for Bracket on Mirage Drive



Bearing Stress in Bracket at Bolt Hole

Thickness $t_{plate} := \frac{3}{16} \cdot in$

Projected Area $A_{proj} := D_{pin1} \cdot t_{plate} = 0.05 \cdot in^2$

Bearing Stress $\sigma_{brg} := \frac{F_{eq}}{A_{proj}} = 13759.59 psi$

Factor of Safety $n_{brg} := \frac{Sy_{alum}}{\sigma_{brg}} = 3.27$

Stress in Bracket at Critical Point - Right Side of Bracket at Bolt Hole

Force on Bracket $F := F_{rod}$

Normal Stress $\sigma_n := \frac{F \cdot \sin(\theta)}{1.5in \cdot .1875in} = 1.21 \times 10^3 \cdot psi$

Shear Stress $\tau := \frac{F \cdot \cos(\theta)}{1.5in \cdot .1875in} = 1206.796 psi$

Bending Moment $M := F \cdot \cos(\theta)(1in)$

Distance from Neutral Axis $y := .75in$

Moment of Inertia $I := \frac{\left[.1875in \cdot (1.5in)^3\right]}{12} - 2\left[\pi \cdot \frac{(.125in)^4}{4} + \pi \cdot (.125in)^2 \cdot (.375in)^2\right]$

Bending Stress $\sigma_b := \frac{(M \cdot y)}{I} = 6604.2 psi$

Combined Stress at the Critical Point $\sigma' := \sqrt{3\tau^2 + \left|\sigma_n + \sigma_b\right|^2} = 8085.8 psi$

Factor of Safety $n_{brkt} := \frac{Sy_{alum}}{\sigma'} = 5.57$

3/8" Quick Release Pin

Area $A_{pin} := \frac{\pi}{4}\cdot D_{pin3}^{2}$

Shear Stress $\tau_{pin} := \frac{F_{rod}}{A_{pin}} = 4345.99\,psi$

Factor of Safety $n_{pin} := \frac{Ssy_{ss}}{\tau_{pin}} = 5.31$

1/4" Bolts Connecting Bracket to Mirage Drive

Area $A_{bolt} := \frac{\pi}{4}\cdot D_{pin1}^{2}$

Angle of Force $\theta := 45deg$

Normal Force $F' := \frac{F_{rod}}{2}$ $F'_x := F'\cdot\cos(\theta)$ $F'_y := F'\cdot\sin(\theta)$

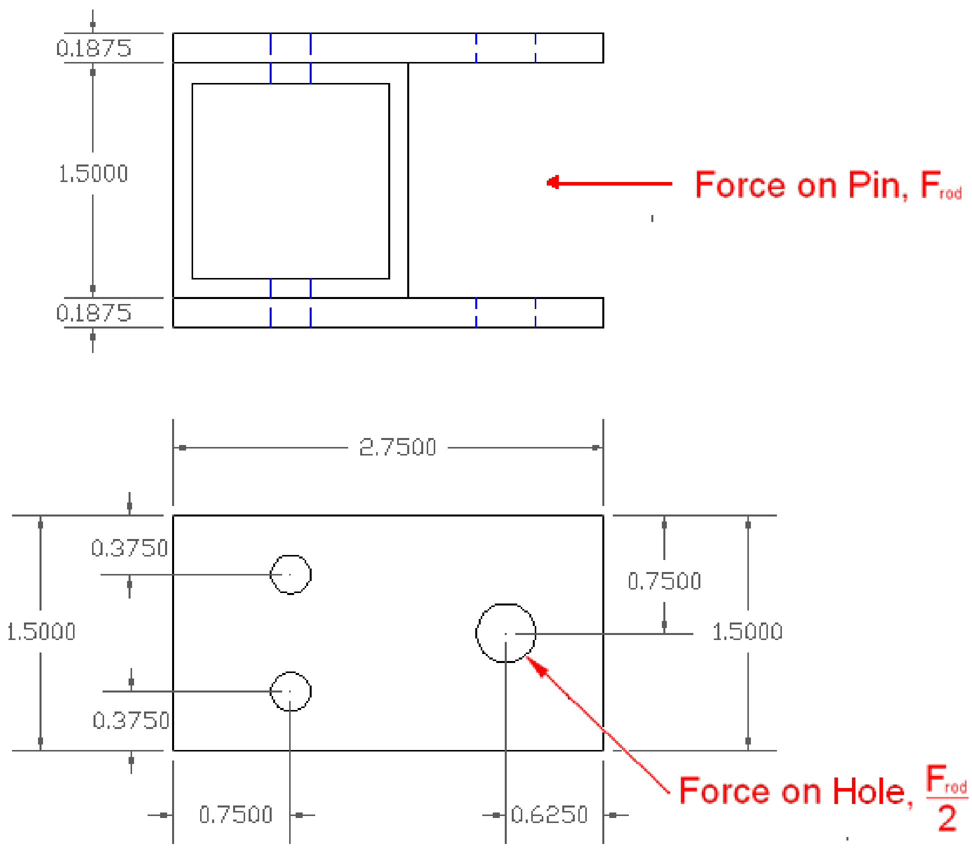
Force Due to Moment $F'' := \frac{1}{2}\cdot\frac{\left|F_{rod}\cdot\cos(\theta)\cdot 1in\right|}{.375in}$

Equivalent Force $F_{eq} := \sqrt{F'^{2}_{x} + \left|F'' + F'_y\right|^{2}}$

Shear Stress $\tau_{bolt} := \frac{F_{eq}}{A_{bolt}} = 13139.437\,psi$

Factor of Safety $n_{bolt} := \frac{Ssy_{gr1}}{\tau_{bolt}} = 4.04$

Figure 10: Stress Analysis for Bracket on Rocker Arm



Bolts Connecting Bracket to Rocker Arm

Bolt Area $A_{bolt} := \frac{\pi}{4} \cdot D_{pin1}^{\ 2}$

Normal Force $F' := \frac{F_{rod}}{2}$ $F'_x := F' \cdot \cos(\alpha)$ $F'_y := F' \cdot \sin(\alpha)$

Force Due to Moment $F'' := \frac{1}{2} \cdot \frac{\left|F'_y \cdot 1.375in\right|}{.375in}$

Equivalent Force $F_{eq} := \sqrt{F'_y{}^2 + \left|F'' + F'_x\right|^2}$

Shear Stress $\tau_{bolt} := \frac{F_{eq}}{A_{bolt}} = 8352.56psi$

Factor of Safety $n_{bolt} := \frac{Ssy_{gr1}}{\tau_{bolt}} = 6.36$

Bearing Stress in Bracket at Bolt

Projected Area $A_{proj} := D_{pin1} \cdot t_{plate} = 0.05 \cdot in^2$

Bearing Stress $\sigma_{brg} := \frac{F_{eq}}{A_{proj}} = 8746.78psi$

Factor of Safety $n_{brg} := \frac{Sy_{alum}}{\sigma_{brg}} = 5.14$

Stress in Bracket at Critical Point - Top of Bracket Above Bolt Hole

Force on Bracket $F := 0.5 \cdot F_{rod}$

Angle of Force $\alpha := 24deg$

Normal Stress $\sigma_n := \frac{F \cdot \cos(\alpha)}{1.5in \cdot .1875in} = 779.56psi$

Shear Stress $\tau := \frac{F \cdot \sin(\alpha)}{1.5in \cdot .1875in} = 347.08psi$

Bending Moment $M := F \cdot \sin(\alpha)(1.375in)$

Distance from Neutral Axis $y := .75in$

Moment of Inertia $I := \frac{\left[.1875in \cdot (1.5in)^3\right]}{12} - 2\left[\pi \cdot \frac{(.125in)^4}{4} + \pi \cdot (.125in)^2 \cdot (.25in)^2\right]$

Bending Stress $\sigma_b := \frac{(M \cdot y)}{I} = 2178.24psi$

Combined Stress at Critical Point $\sigma' := \sqrt{3\tau^2 + \left|\sigma_n + \sigma_b\right|^2} = 3018.27psi$

Factor of Safety $n_{bracket} := \frac{Sy_{alum}}{\sigma'} = 14.91$

Bearing Stress in Bracket at Pin

Projected Area $A_{proj} := D_{pin3} \cdot t_{plate} = 0.07 \cdot in^2$

Bearing Stress $\sigma_{brg} := \frac{0.5F_{rod}}{A_{proj}} = 3413.33psi$

Factor of Safety $n_{brg} := \frac{Sy_{alum}}{\sigma_{brg}} = 13.18$

Solid Modeling and FEA Analysis

After completing the 2D drawings and hand calculations for the brackets, Solidworks and ProEngineer were used to conduct FEA on 3D models of the same parts. The results of the FEA are shown below in Figures 11-13. A comparison between hand calculations and FEA is also given with each figure. This comparison is necessary to confirm the validity of FEA constraints and loading for each part.

Figure 11a depicts the Von Mises stess in the bracket under a given load. Figure 11b represents the displacement in the bracket. The restraints on the bracket are representative of a bolted joint. From the free-body diagram in Figure 6, the applied force F_{ax} is a bearing load of 480 pounds at an angle 30 degrees from horizontal. The difference between hand calculations and FEA calculations for this part is 20%, with the hand calculations predicting slightly higher maximum stress than FEA. The hand calculations are shown previously in Figure 10.

Figure 11a: Rocker Bracket Von Mises Stress

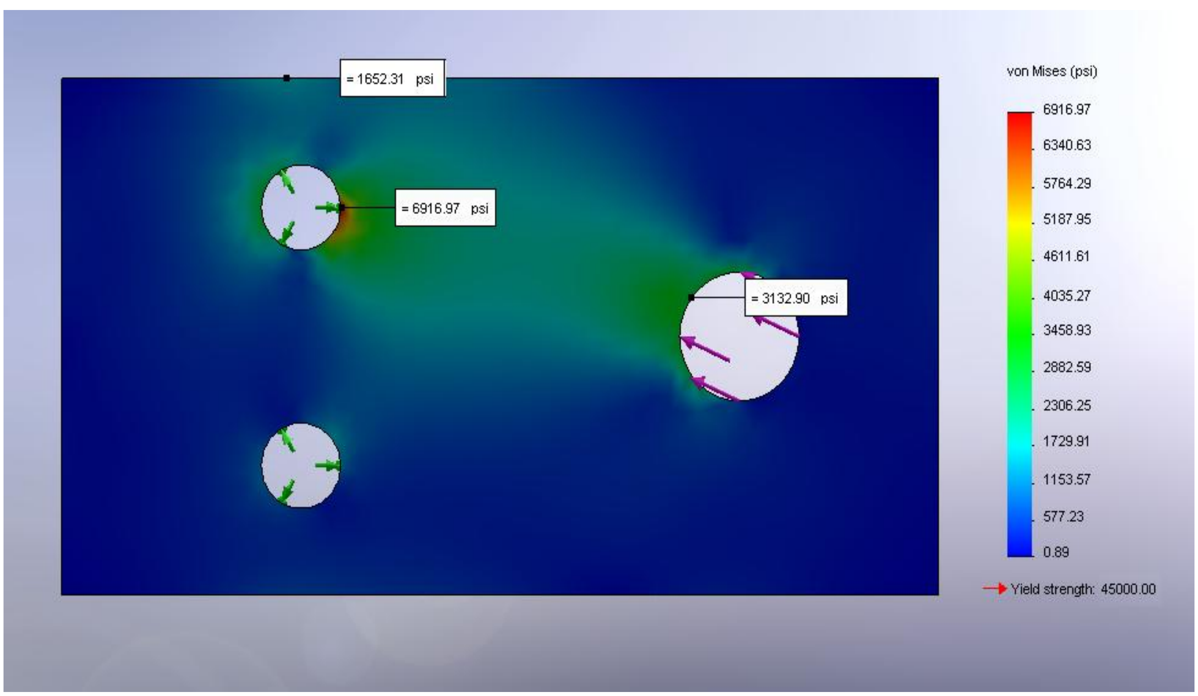


Figure 11b: Rocker Bracket Displacement

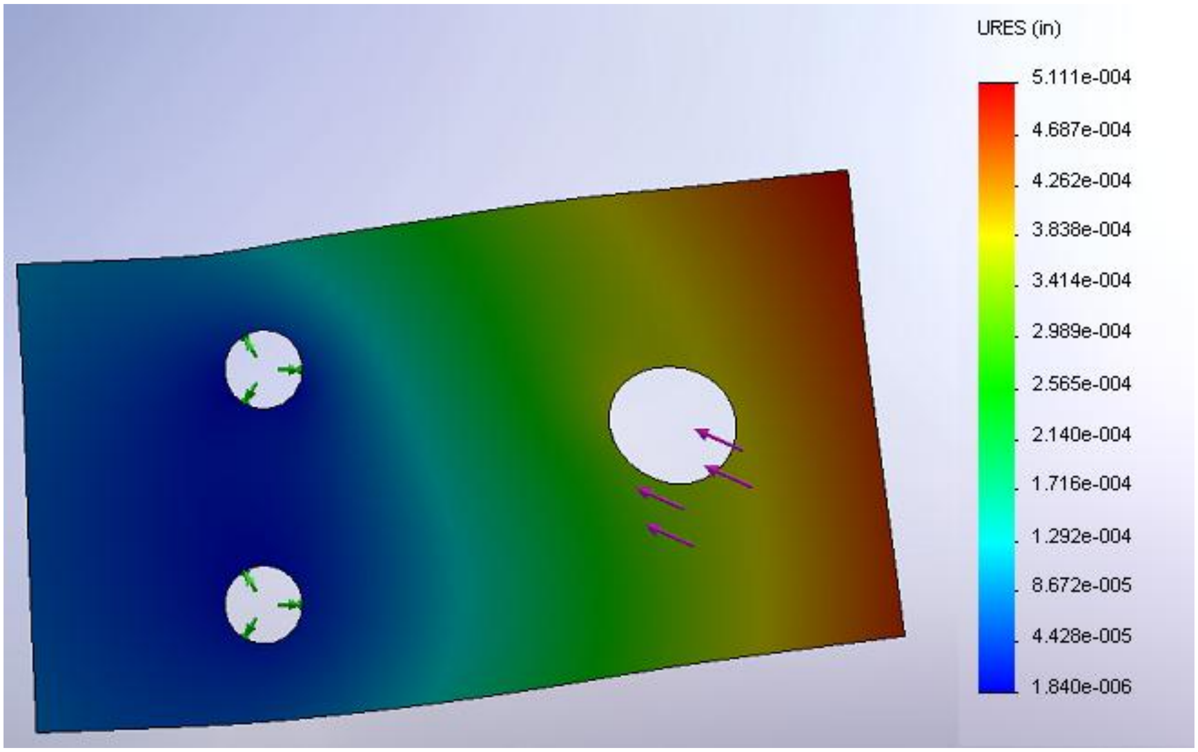


Figure 12a depicts the Von Mises stress in the bracket under a given load. Figure 12b represents the deformation in the bracket. The restraints on the bracket are representative of a bolted joint. From the free-body diagram in Figure 6, the applied force F_{ax} is a bearing load of 480 pounds at an angle 30 degrees from horizontal. The difference between hand calculations and FEA calculations for this part is 8%, with the hand calculations predicting slightly lower maximum stress than FEA. The hand calculations are shown previously in Figure 9.

Figure 12a: Mirage Bracket Von Mises Stress

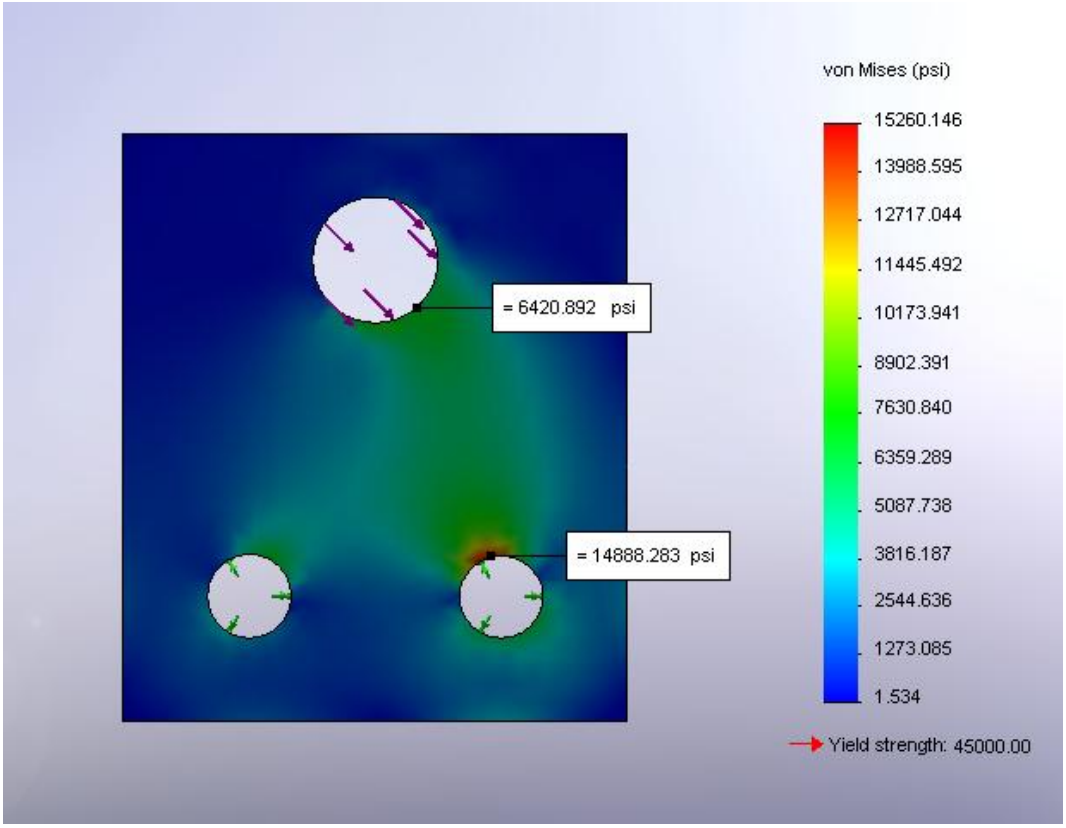


Figure 12b: Mirage Bracket Displacement

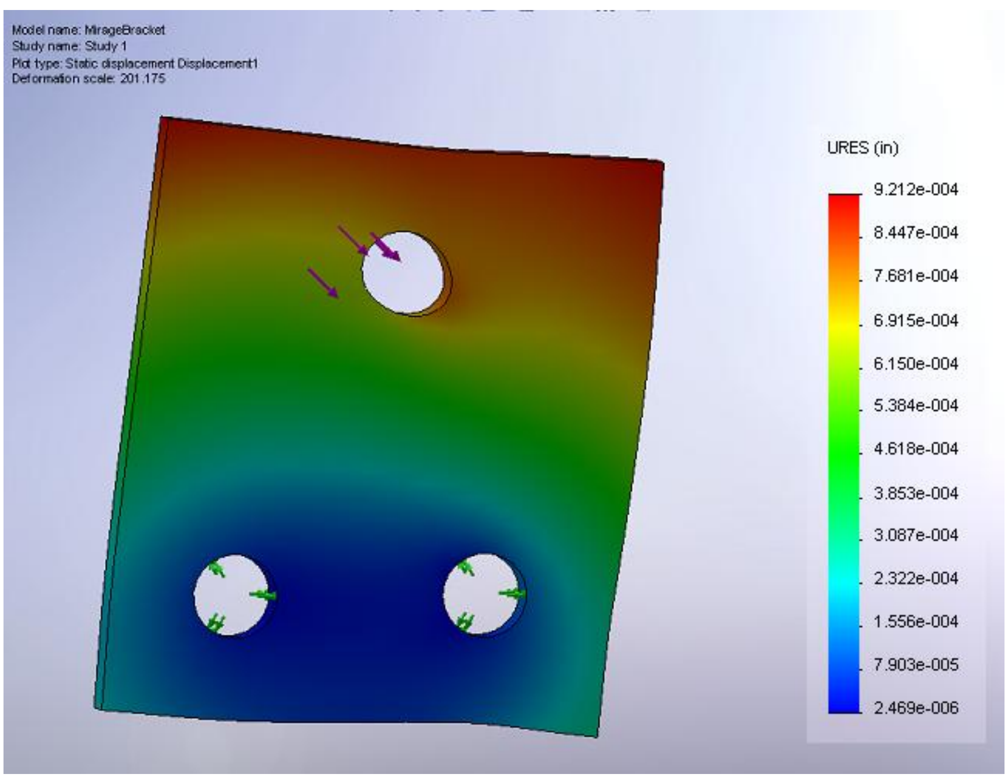


Figure 13a: W Bracket Von Mises Stress

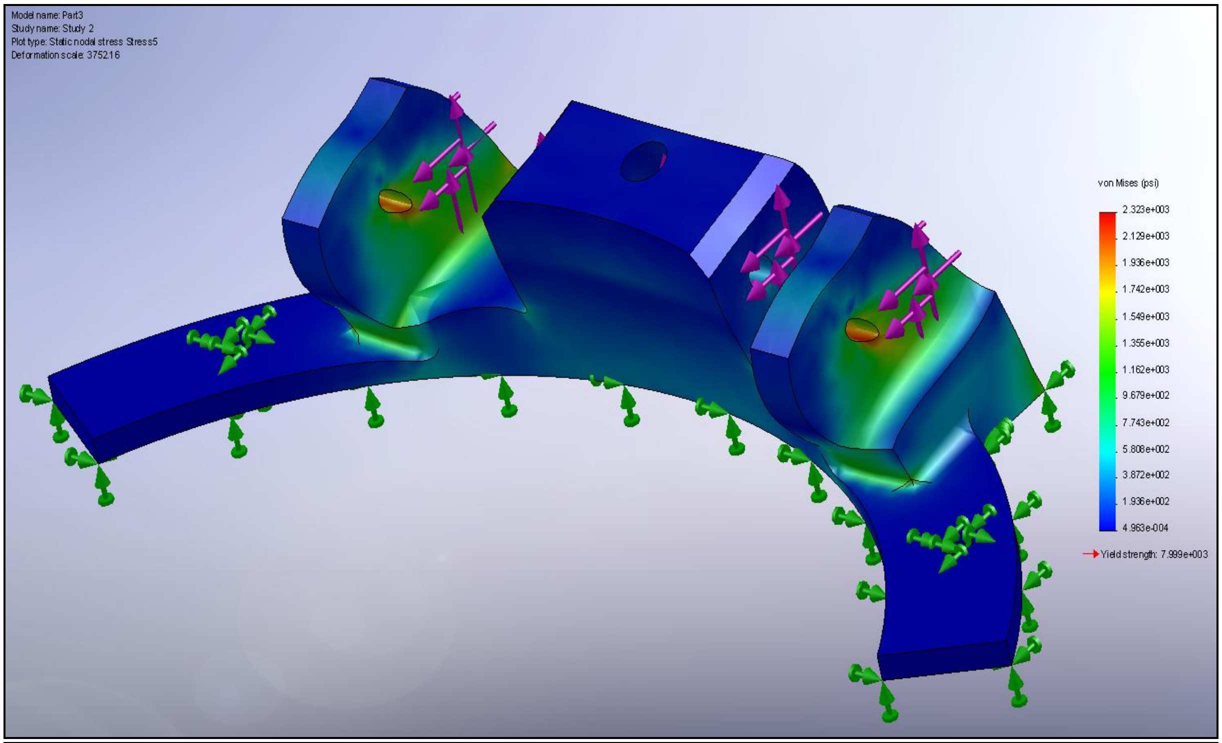


Figure 13b: W Bracket Displacement

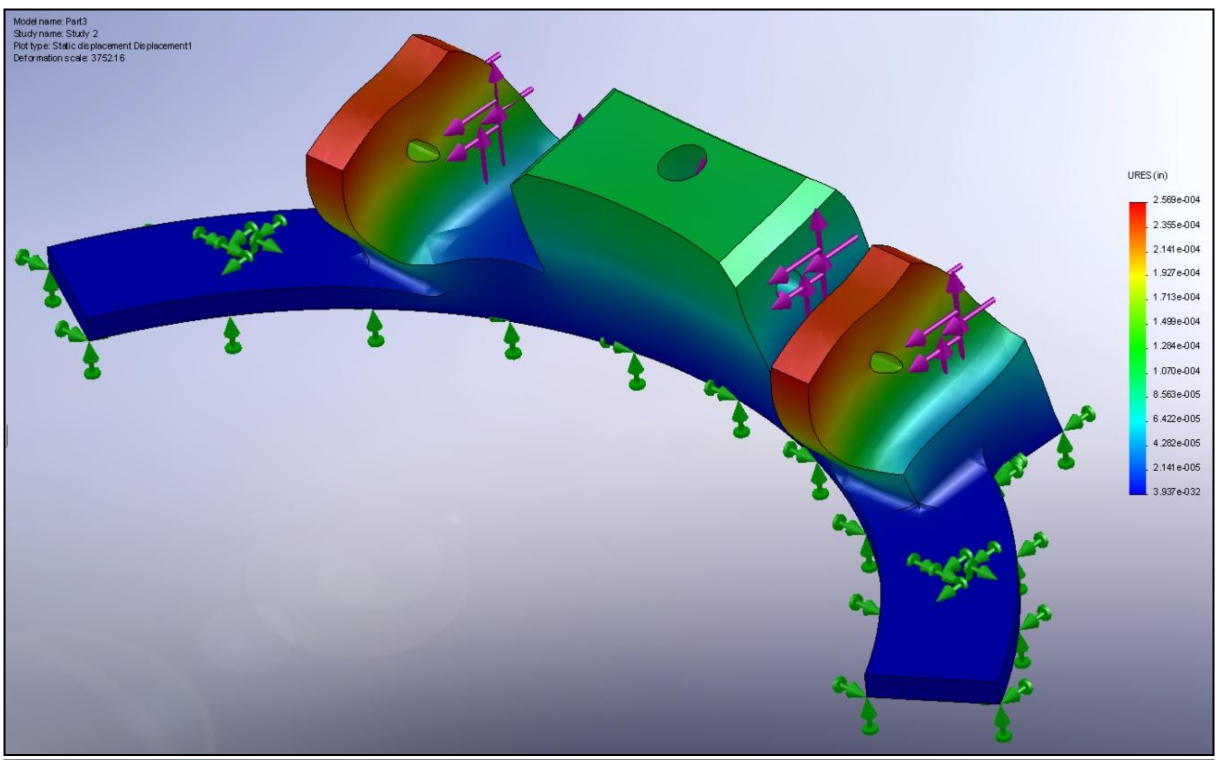


Figure 13a depicts the Von Mises stress in the bracket under a given load. Figure 13b represents the deformation in the bracket. The restraints on the bracket are representative of a welded base. From the free-body diagram in Figure 6, the applied force F_{bx} is a bearing load of 330 pounds in the horizontal direction.

Project Deliverables

The final product meets the client specifications as well as specifications created throughout the design and analysis phase. The material for all major components is 6061-T6511 aluminum, while all fasteners are stainless steel. Dimensions are as given in the calculations shown in the Design & Analysis section of the report. These dimensions were confirmed during prototype testing, and the final product has been successfully tested for fit-up. The kayak has also been successfully tested on the Maumee River with the final product mechanism attached. The client requested a mechanism to allow users with at least one functional arm to enjoy Hobie Cat kayaking, and the final product safely and efficiently meets this request.

In preparation for the design exposition, the group also has several small but important tasks to complete. These tasks include: constructing a suitable display stand for the kayak with the final mechanism attached and functioning, as well as constructing a display board outlining the project in its entirety. The group has kept logs of all activities, detailed meeting notes, and pictures of all machining, testing, and assembly. A selection of these items is included on the project display board.

Timeline

This project allows just 16 weeks total for completion. At the beginning of the project a tentative timeline was established to give focus on the most important tasks. Throughout the duration of this project, each member of the group understood the importance of communication, both with each other and with the clients and advisors, and the importance of hard work in order to complete the project of time. Now, the project has been completed, and the hard work of the team has been realized. The team was able to follow the original timeline of work closely, and a representation of how the work progression actually occurred, is shown in Table 4 below.

Table 4: Actual Timeline

	January			February				March					April				May
	12	19	26	2	9	16	23	2	9	16	23	30	6	13	20	27	4
Establish Group																	
Assign Roles																	
Meet with Client																	
Meet with Client Advisors																	
Meet with Faculty Advisors																	
Brainstorming Sessions																	
Establish Multiple Designs																	
Design Selection																	
Proposal Presentation/Report																	
Design Modeling																	
Order Materials																	
Assemble/Test																	
Midterm Presentation/Report																	
Finished Product																	
Final Presentation/Report																	
Design Expo																	
NSF CD and Abstract																	
Evaluations/Final Paperwork																	

From this timeline, it is easy to see that the project progressed on time with all reports and presentations completed on time. There was a small delay in prototype fabrication through the UT machine shop. This delay forced the group to order materials approximately one week later than projected. The machining of final parts, however, had a much lower lead time than that of the prototype parts. This improved lead time is due to the group technical liaison, Brian Back, having contact with a third-party machine shop that has donate machining time and resources to the group as described in the budget. Due to this arrangement, the small delay did not affect the final delivery date or quality of the finished product.

The group compared actual progress with the original timeline at every group meeting. This process allowed the group to focus work on critical path items so the final deadline was met with the most efficient use of resources.

<u>Budget</u>

Common materials and commodity fasteners will be used to build the mechanism. Some of the fabrication processes was moderately complex, namely the machining of the “W” bracket, but tolerances were relatively loose without sacrificing safety due to simple motions and low forces required. The project design allows reasonable costs and easy fabrication in addition to the client requirements. The proposed budget was initially compiled from a draft bill of material and has been refined throughout the project once exact materials and products that were used were found. The breakdown of the actual budget is shown in Table 5 below:

Table 5: Actual Budget

Aluminum Tube	$87
Aluminum Stock	$43
Aluminum Plate	$53
Quick Release Pins	$40
Welding Rod	$13
Ball Joint Rod Ends	$63
Miscellaneous Expenses	$35
Travel	$182
Shipping	$85
Machining Costs	$65/hr
<u>Total Cost of Project</u>	<u>$516 + Machining Cost</u>

The total time for machining of the prototype and final product is estimated at 36 hours total. This machining time was donated by the Reliable Mold and Pattern Inc. of Marshallville, Ohio, and the University of Toledo machine shop, and thus is not included in the total cost of this project. The total cost for machining would have accrued to an estimated $2400.

Summary

The design team was able to successfully design and fabricate a final product that met client specifications while staying under budget in the allotted 16 week time period. Several designs were considered, a formal design and stress analysis procedure followed, and a well-engineered, efficiently constructed mechanism was produced. Opportunities for improvement are primarily in the area of welding, as a higher weld quality was desired for the final product. The group recommends that future generations of the product should include proper welds to better meet design requirements and ensure longevity under the most extreme operating conditions.

Acknowledgements

First, the group would like to thank Reliable Mold and Pattern Incorporated of Marshallville, Ohio, and Benjamin Back, for their generous contribution of facilities and time for machining of parts of the final product[2]. The group would like to acknowledge our client, Mr. Steve Grudzien of Patriot Products, for bringing this project to the attention of the university and his support by lending the kayak to the group for testing and also a variety of handles to attach to the mechanism[3]. The group would like to thank Ms. Jill Caruso of the Ability Center for her support and perspective on the needs of users with disabilities. The group would also like to thank Dr. Chris Beins for his advice based on his 30 years of engineering experience. The group would like to thank Dr. Pourazady and Dr. Hefzy for their technical guidance and advice and feedback on all graded materials for this project. The group would like to acknowledge the assistance provided by Mr. Randy Reihing during machining at the UT machine shop.

References

1. Hobie Cat Website – Description of Mirage Drive Operation

   http://www.hobiecat.com/kayaking/miragedrive.html

2. Reliable Mold and Pattern Inc. – Marshallville, Ohio

3. Patriot Products – Toledo, Ohio

   http://dmetree.com

4. McMaster-Carr

   http://www.mcmaster.com

5. Shigley, Charles R., and Mischke, Joseph Edward. Mechanical Engineering Design. New York: McGraw Hill, Inc., 2002.