Hobie Cat Kayak

Midterm Report
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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

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Executive Summary

The objective of the project is to design 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 mechanism will adapt the Mirage Drive System to transfer energy from the user's arms to the foils. This will allow any user with one or two arms to efficiently propel the kayak with a simple push and/or pull motion of one or two arms. The proposed system will reduce 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 has an excellent description of the Mirage Drive System with pictures and videos to explain the mechanism's operation: http://www.hobiecat.com/kayaking/miragedrive.html

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 it still leaves behind anybody without use of their legs. The purpose of this project is to develop a mechanism which allows anybody with at least one functional arm to enjoy the same type of kayaking that we all take for granted. The value that this project can provide to disabled users far exceeds its small costs in design, fabrication, and materials.

Team Roles

All group members are 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 has also chosen 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 works 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 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 will not eliminate any functionality of the existing kayak nor the Mirage Drive System. This means that the user should be able to sit comfortably in the kayak with or without the mechanism installed. The mechanism will be simple and lightweight to minimize cost, maintenance, and transportation effort. The mechanism will 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 environment will include both freshwater as well as saltwater.

Proposed Designs

Three different designs were considered. 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.

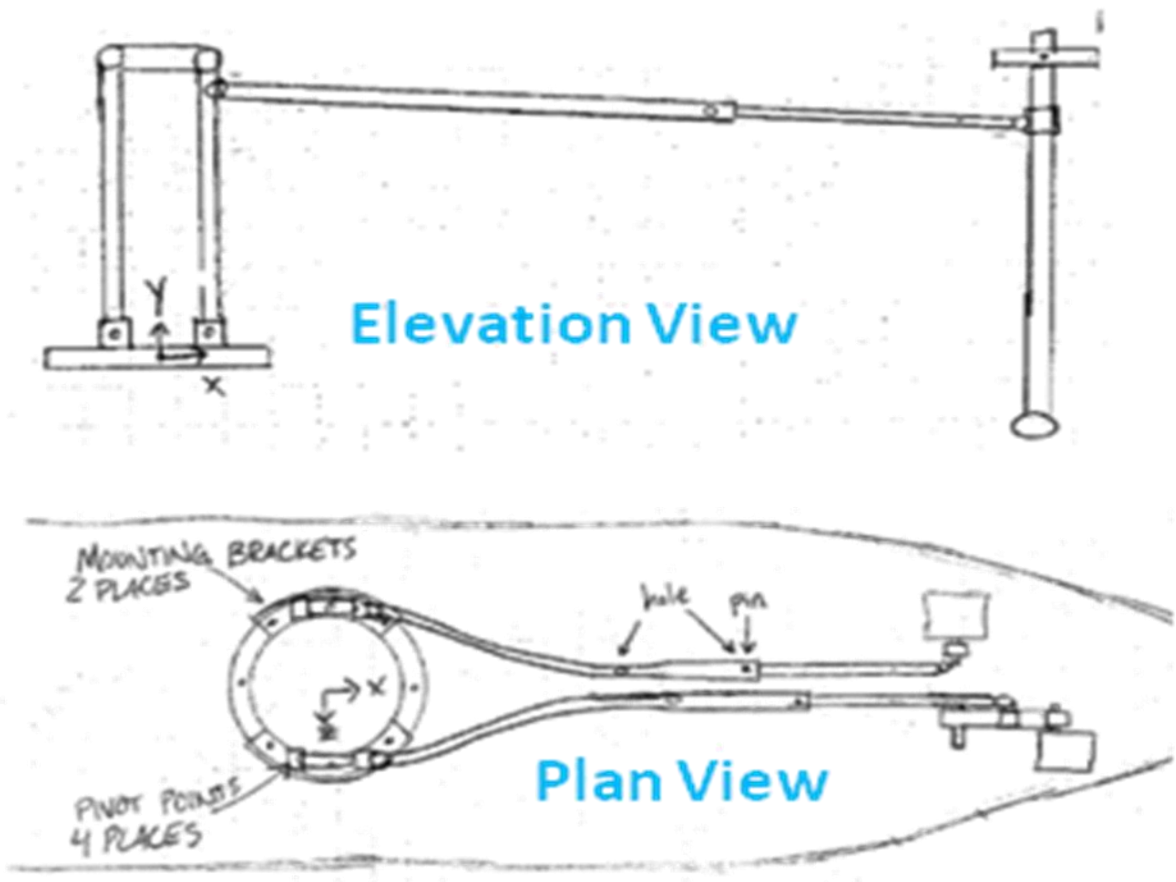


Figure 1: Rocker Mechanism

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.

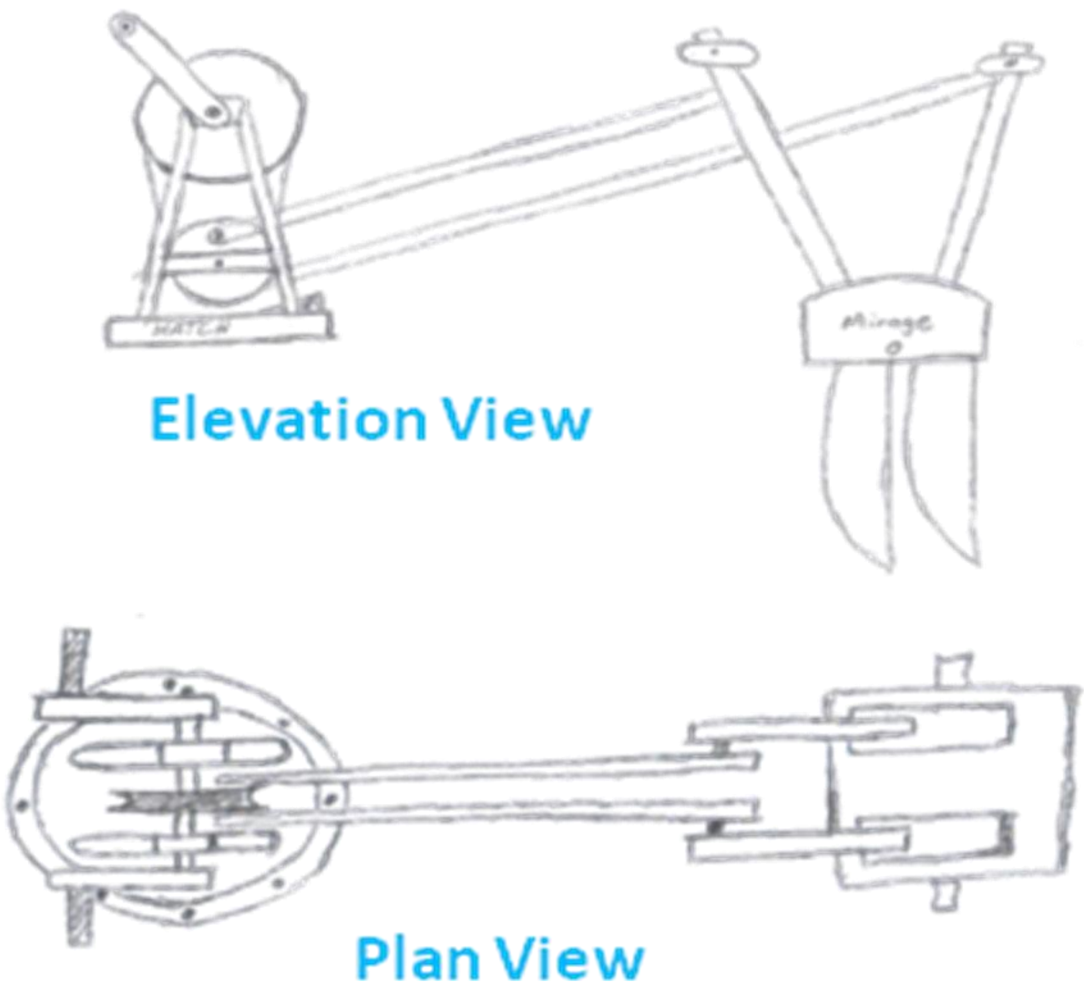


Figure 2: Hand Crank Mechanism

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.

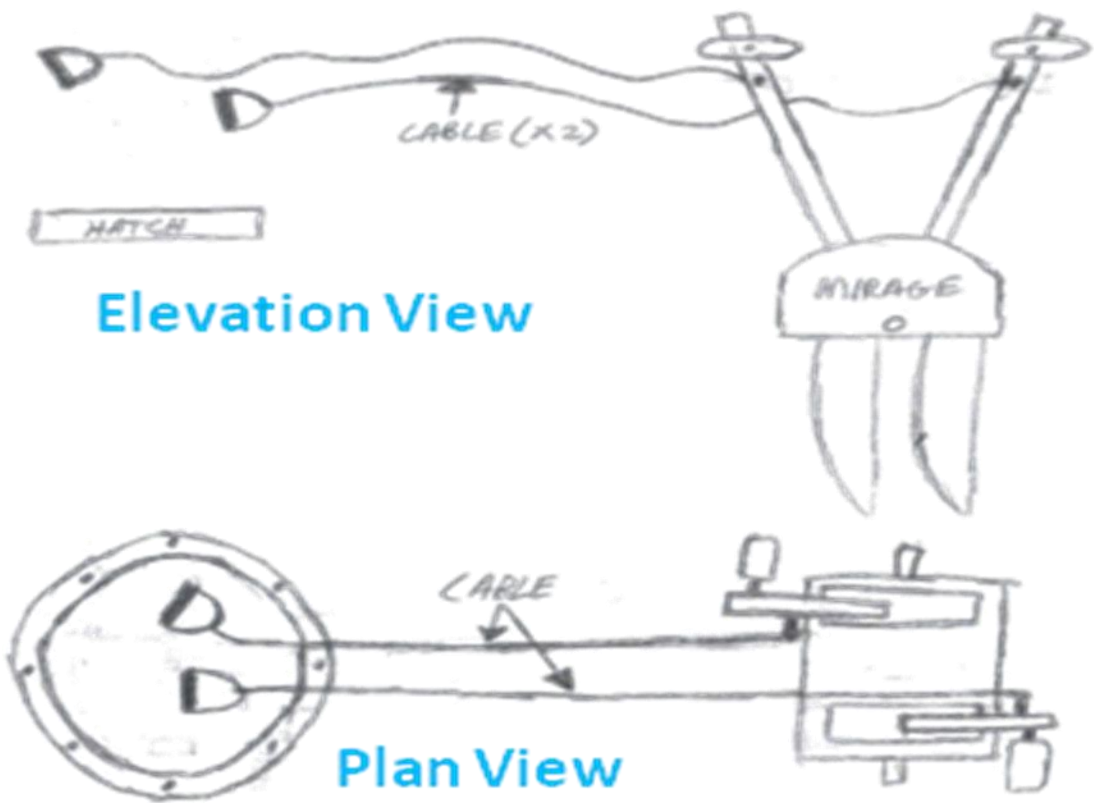


Figure 3: Direct Cable Mechanism

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 also makes it the easiest for the user to install and remove. This design 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.

	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

Table 1: House of Quality for Basic Design

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 will 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. A pin joint will be at the access hatch of the kayak, between the user's legs. 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.

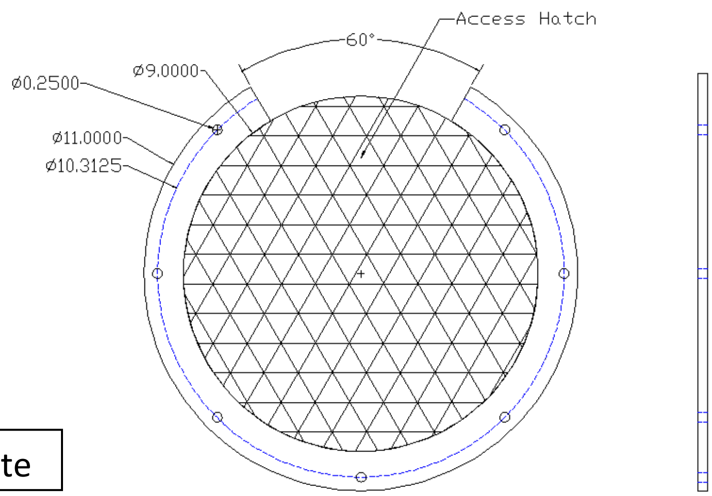


Figure 4: Base Plate

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 one of the mechanism is considered the rocker arm, while link two is referred to as the connecting rod. The existing Mirage Drive system is represented as the third link, and the body of the Kayak is the ground link of the four bar linkage.

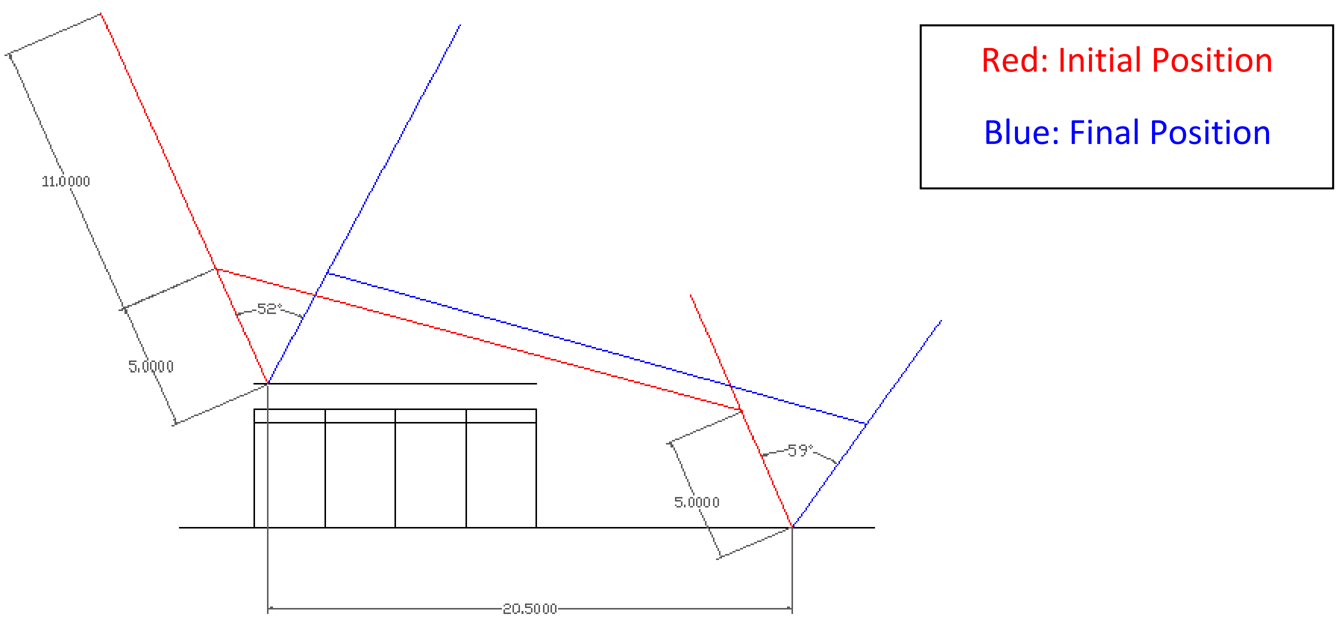


Figure 5: Parameter Analysis

A free body diagram was constructed to determine the reaction forces in each member. This analysis will be used in determining the stresses for all components of the design. Figure 6 shows this free body analysis.

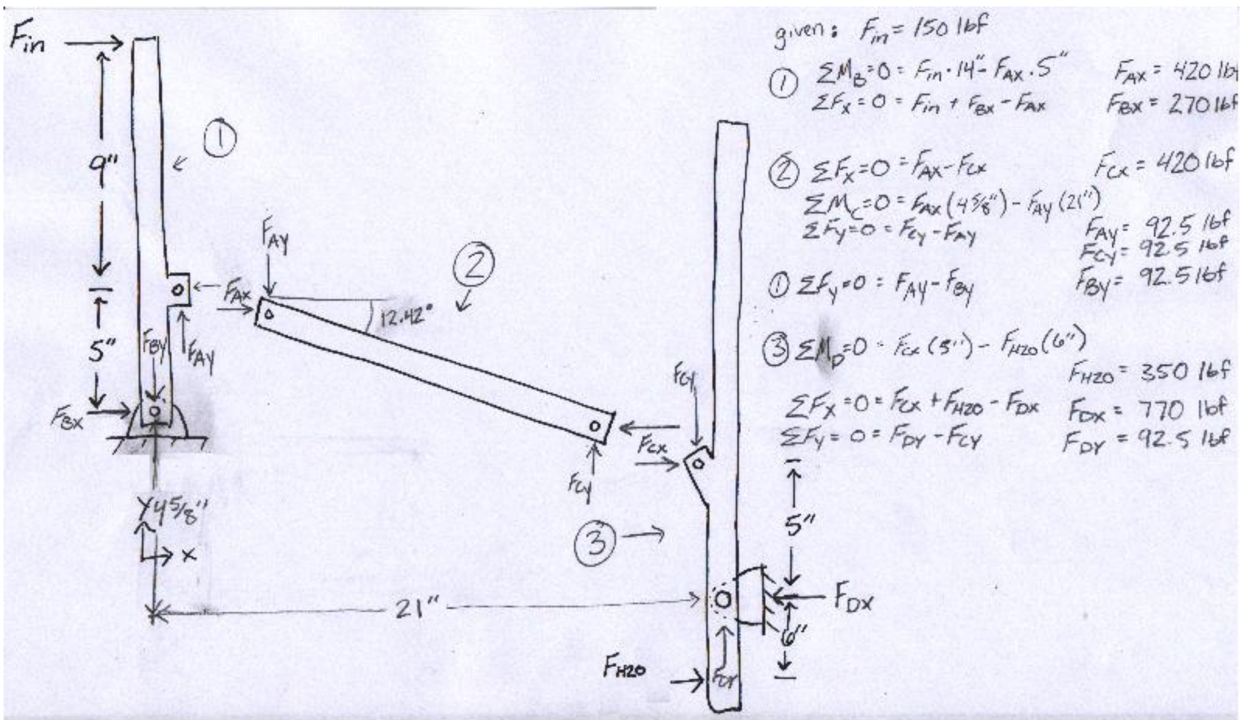
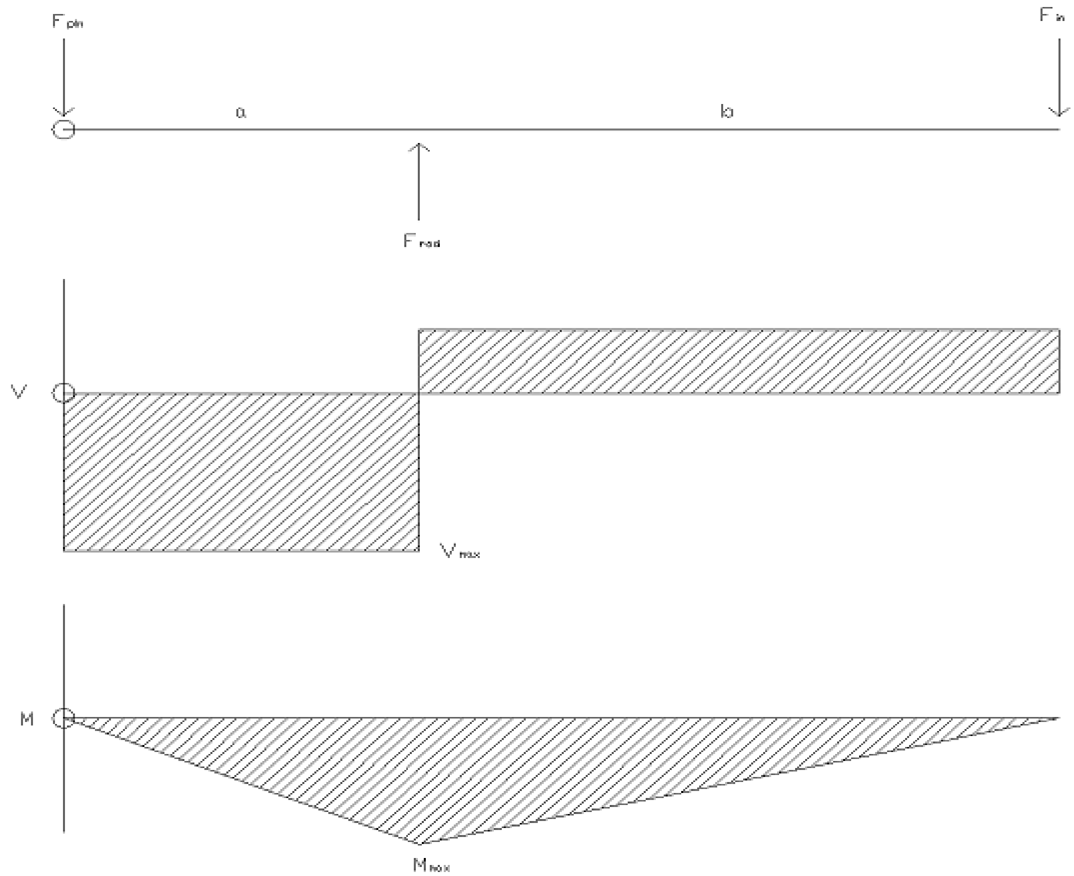


Figure 6: Free Body Diagram

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 three pin joints 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 16 inches long with a wall thickness of 0.125 inches could be used for this application with a fatigue safety factor of approximately 5.

---BEGIN ROCKER ARM---



Calculate cross sectional area of bottom half (larger) rocker arm

$OD_{rock} := 1in$ $T_{rock} := .125in$ $ID_{rock} := OD_{rock} - |2 \cdot T_{rock}|$ $ID_{rock} = 0.75in$

$A_{rock} := OD_{rock}^2 - ID_{rock}^2$ $A_{rock} = 0.438in^2$ $I_{xxrock} := \frac{OD_{rock}^4}{12} - \frac{ID_{rock}^4}{12}$

Calculate maximum moment generated and max shear, bending stress

$F_{in} := 150lbf$ $a := 5in$ $b := 9in$ $F_{rod} := \frac{F_{in} \cdot (a + b)}{a}$ $F_{rod} = 420\,lbf$

$F_{pin} := F_{rod} - F_{in}$ $F_{pin} = 270\,lbf$ $\tau_{max} := \frac{F_{pin}}{A_{rock}}$ $\tau_{max} = 617.143psi$

$M_{max} := F_{in} \cdot b$ $M_{max} = 5.212 \times 10^5 \frac{lb \cdot in^2}{s^2}$ $\sigma_{bend} := M_{max} \cdot \frac{\left(\frac{OD_{rock}}{2}\right)}{I_{xxrock}}$ $\sigma_{bend} = 1.185 \times 10^4 psi$

Calculate fatigue factor of safety at 10^6 cycles

$S_f := 45000psi$ $\sigma_{total} := \left(3\tau_{max}^2 + \sigma_{bend}^2\right)^{.5} = 1.19 \times 10^4 psi$ $FS_{rock} := \frac{S_f}{\sigma_{total}} = 3.782$

Figure 7: Stress Analysis for Rocker arm

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 4, which is well above the specified minimum of 2 for this project.

---CONNECTING ROD---

Calculate factor of safety for axial stress

$OD_{rod} := 1.05in$ $\quad T_{rod} := .125in$ $\quad ID_{rod} := OD_{rod} - \left|2 \cdot T_{rod}\right|$ $\quad ID_{rod} = 0.8\,in$

$$A_{rod} := \frac{\pi \cdot OD_{rod}^{2}}{4} - \frac{\pi \cdot ID_{rod}^{2}}{4} \qquad A_{rod} = 0.363in^{2} \qquad \sigma_{axial} := \frac{F_{rod}}{A_{rod}} \qquad \sigma_{axial} = 1.156 \times 10^{3}\,psi$$

$$FS_{frod} := \frac{S_{f}}{\sigma_{axial}} \qquad FS_{frod} = 16.433$$

Caclulate factor of safety for failure by buckling

$E := 10.4 \cdot 10^{6}psi$ $\quad L_{rod} := 10.5in$

$$I_{rod} := \frac{\pi}{64} \cdot \left(OD_{rod}^{4} - ID_{rod}^{4}\right) \qquad P_{cr} := \frac{\pi^{2} \cdot E \cdot I_{rod}}{L_{rod}^{2}}$$

$$P_{cr} = 3.683 \times 10^{4}\,lbf \qquad FS_{rodbuckle} := \frac{P_{cr}}{F_{rod}} \qquad FS_{rodbuckle} = 87.692$$

Figure 8: Stress Analysis for Connecting Rod

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 seven different locations for the pivot point, in order to test different configurations to find the optimum 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.

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 to the Mirage Drive. Tests were executed with the kayak stationary and moving in the water. A spring scale was attached to the input arm of the Mirage Drive and different forces were applied by the tester. 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.

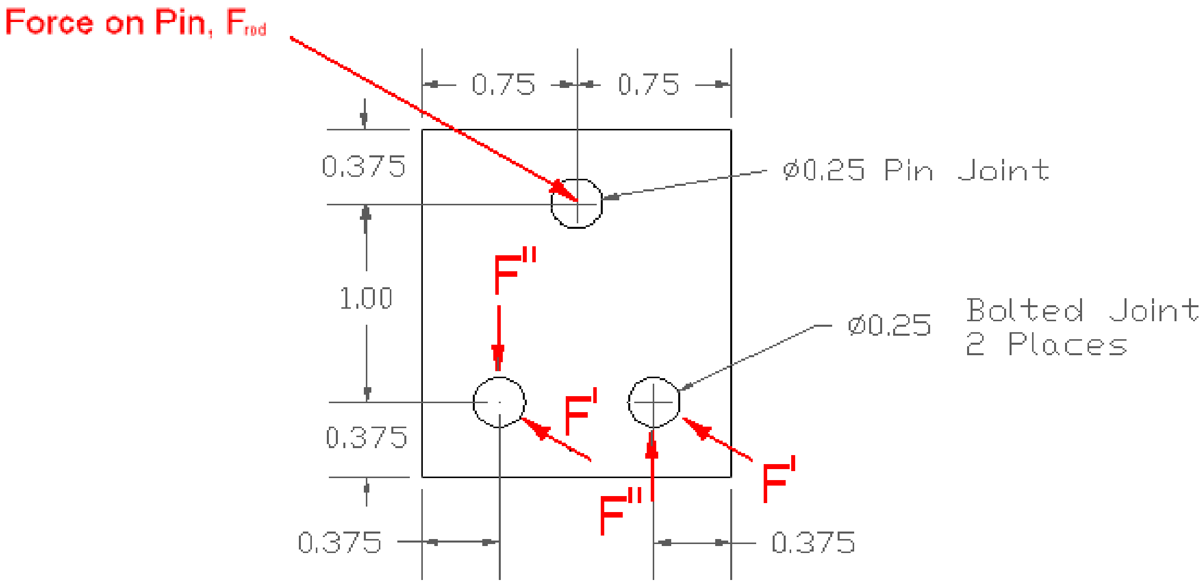
Prototype testing on Maumee River, 26-Mar-2009					
	Stationary			Moving	
	Typical	Maximum		Typical	Maximum
Test 1	21	75		20	67
Test 2	18	65		23	70
Test 3	22	82		17	76

Table 2 – Test Results

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. All of the previous calculations had to be revised to reflect this increased loading. 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.

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.

Mounting Bracket on Mirage Drive



Quick Release Pin

Stainless Steel $Sy_{ss} := 40000 \cdot psi$ $Ssy_{ss} := .577 \cdot Sy_{ss} = 23080 \cdot psi$ $F_{rod} := 420 \cdot lbf$

Diameter $D_{pin} := \frac{1}{4} \cdot in$ Area $A_{pin} := \frac{\pi}{4} \cdot D_{pin}^{2} = 0.049 \cdot in^{2}$

$\tau_{pin} := \frac{F_{rod}}{A_{pin}} = 8556.17 \cdot psi$ $n_{pin} := \frac{Ssy_{ss}}{\tau_{pin}} = 2.697$

Bolts Connecting Bracket to Mirage Drive

SAE Grade 5 Bolt $Sy_{gr1} := 92000 \cdot psi$ $Ssy_{gr1} := .577 \cdot Sy_{gr1} = 53084 \cdot psi$ $\theta := 45deg$

Diameter $D_{bolt} := \frac{1}{4} \cdot in$ Area $A_{bolt} := \frac{\pi}{4} \cdot D_{pin}^{2}$

$F' := \frac{F_{rod}}{2} = 210 \cdot lbf$ $F'_{x} := F' \cdot \cos(\theta)$ $F'_{y} := F' \cdot \sin(\theta)$

$F'' := \frac{1}{2} \cdot \frac{\left|F_{rod} \cdot \cos(\theta) \cdot 1in\right|}{.375in} = 395.98 \cdot lbf$ $F_{eq} := \sqrt{F'_{x}{}^{2} + \left|F'' + F'_{y}\right|^{2}} = 564.358 \cdot lbf$

$\tau_{bolt} := \frac{F_{eq}}{A_{bolt}} = 11497.008 \cdot psi$ $n_{bolt} := \frac{Ssy_{gr1}}{\tau_{bolt}} = 4.617$

Bearing Stress in Bracket

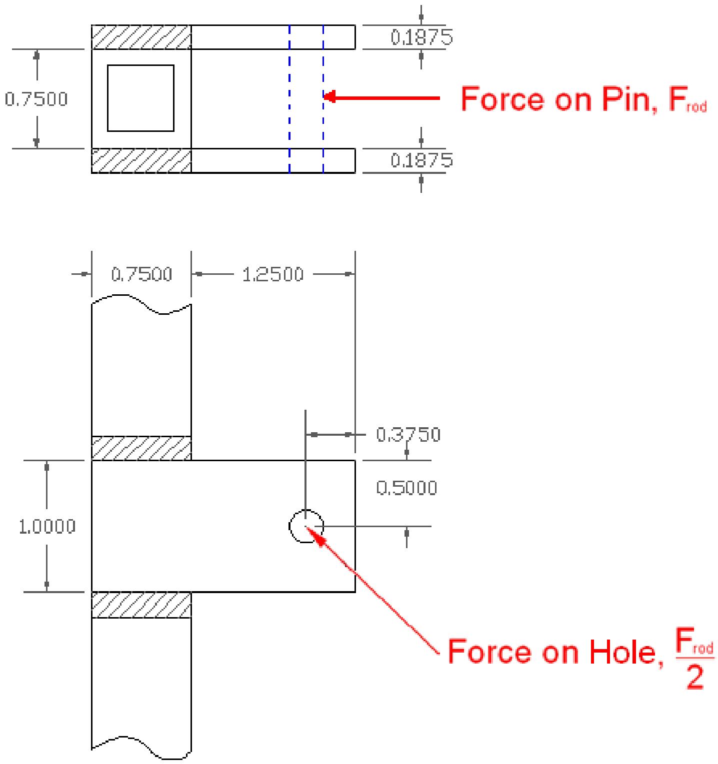
Aluminum $Sy_{alum} := 35000 \cdot psi$ $Ssy_{alum} := .577 \cdot Sy_{alum} = 20195 \cdot psi$

Thickness $t_{plate} := \frac{3}{16} \cdot in$ $A_{proj} := D_{pin} \cdot t_{plate} = 0.047 \cdot in^{2}$

$\sigma_{brg} := \frac{F_{eq}}{A_{proj}} = 12039.638 \cdot psi$ $n_{brg} := \frac{Sy_{alum}}{\sigma_{brg}} = 2.907$

Figure 9: Stress Analysis for Bracket on Mirage Drive

Mounting Bracket for Connecting Rod to Rocker Arm



Quick Release Pin

Diameter $D_{pin} := \frac{1}{4} \cdot in$ Area $A_{pin} := \frac{\pi}{4} \cdot D_{pin}{}^{2} = 0.049 \cdot in^{2}$ $F_{rod} := 420 \cdot lbf$

$\tau_{pin} := \frac{0.5F_{rod}}{A_{pin}} = 4278.08 \cdot psi$ $n_{pin} := \frac{Ssy_{ss}}{\tau_{pin}} = 5.395$

Weld Strength

$\theta := 30deg$ $b := 1 \cdot in$ $d := .75 \cdot in$ $t_{weld} := .1875 \cdot in$

Moment about weld CG: $M := (.375 \cdot in + .875 \cdot in) \cdot 0.5F_{rod} \cdot \cos(\theta)$

Shear Force $V := .5F_{rod} \cdot \cos(\theta)$ $A_{throat} := 2\left[\frac{3}{4} \cdot in \cdot (.707t_{weld})\right] = 0.199 \cdot in^{2}$

$\tau' := \frac{V}{A_{throat}} = 914.614 \cdot psi$ $Ju := \frac{d \cdot (3 \cdot b^{2} + d^{2})}{6}$ $J := .707 \cdot t_{weld} \cdot Ju$

Distance from CG to critical point $r := \sqrt{(.5 \cdot in)^{2} + (.375 \cdot in)^{2}}$

$\tau'' := \frac{M \cdot r}{J} = 2406.88 \cdot psi$ $\gamma := atan\left(\frac{.5}{.375}\right) = 53.13 \cdot deg$ $\tau''_{x} := \tau'' \cdot \sin(\gamma)$ $\tau''_{y} := \tau'' \cdot \cos(\gamma)$

$\tau := \sqrt{\tau''_{x}{}^{2} + (\tau''_{y} + \tau')^{2}} = 3044.869 \cdot psi$ $n_{weld} := \frac{Ssy_{alum}}{\tau} = 6.632$

Figure 10: Stress Analysis for Mounting Bracket

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 11a-11f.

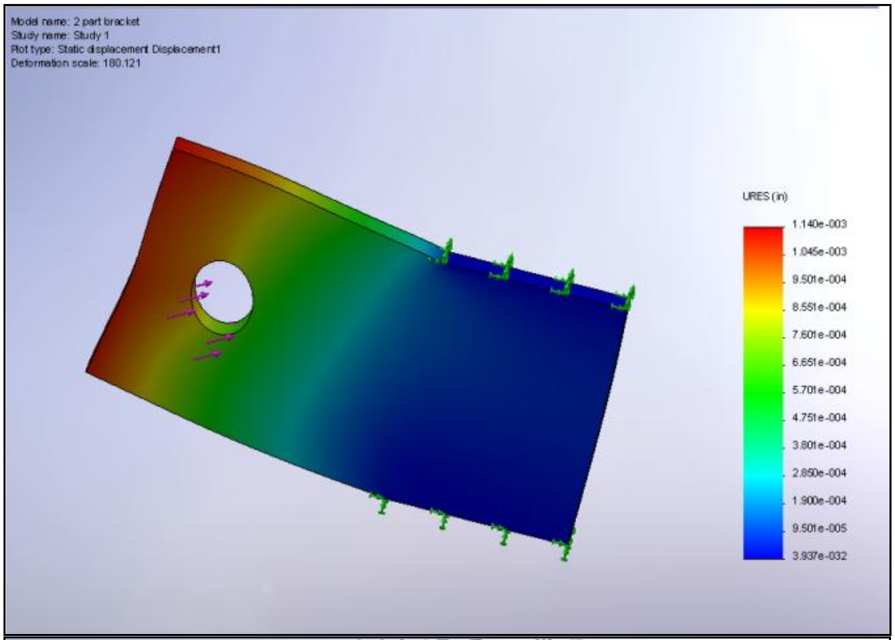


Figure 11a

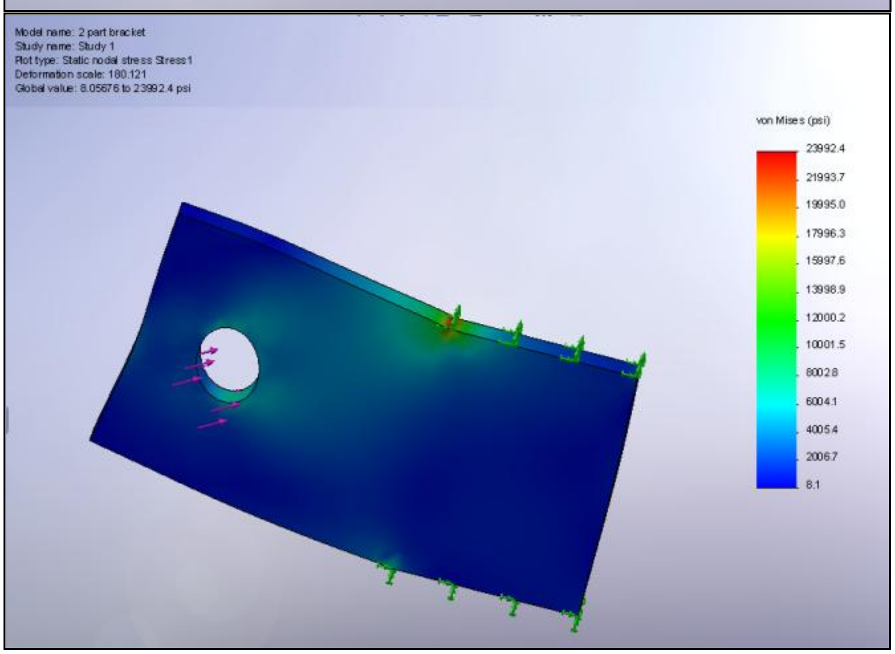


Figure 11b

Figure 11a depicts deformation in the bracket under a given load. Figure 11b represents the Von Mises stress in the bracket. The restraints on the bracket are representative of a welded joint. The applied force is a bearing load of 420 pounds at an angle 30 degrees from horizontal.

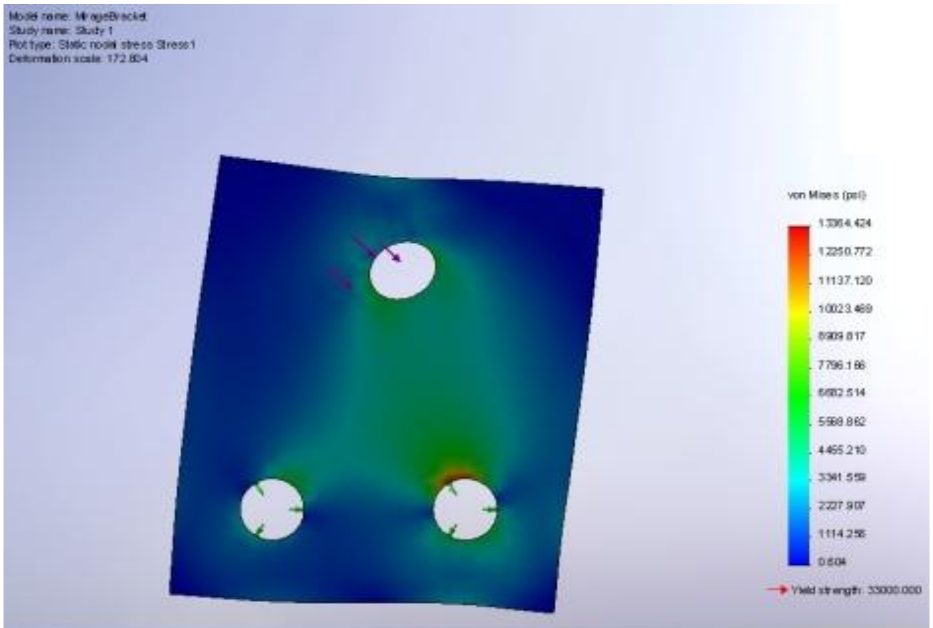


Figure 11c

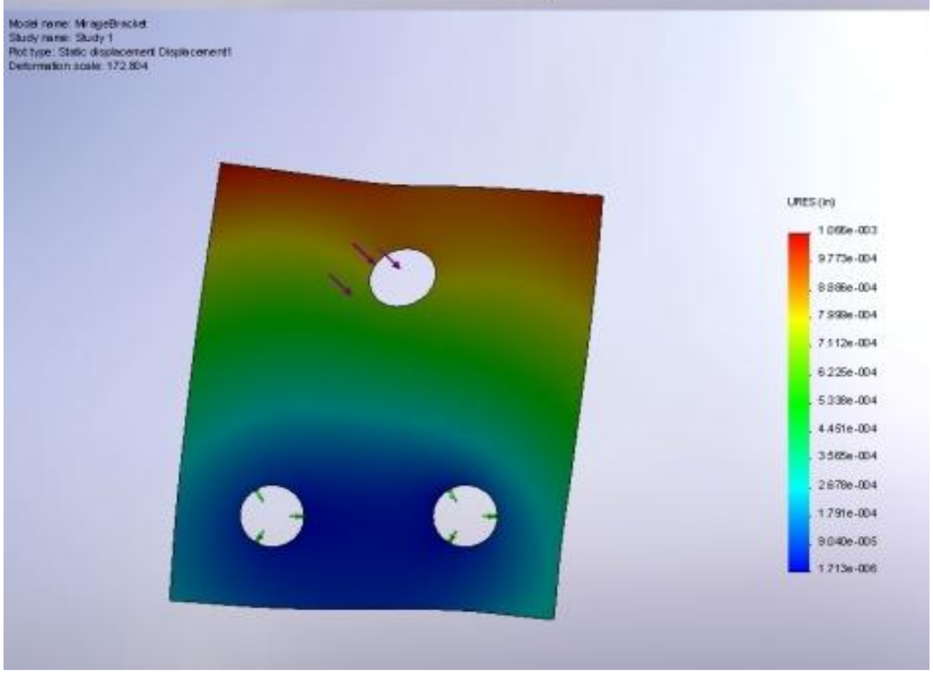


Figure 11d

Figure 11c depicts the Von Mises stress in the bracket under a given load. Figure 11d represents the deformation in the bracket. The restraints on the bracket are representative of a bolted joint. The applied force is a bearing load of 420 pounds at an angle 30 degrees from horizontal.

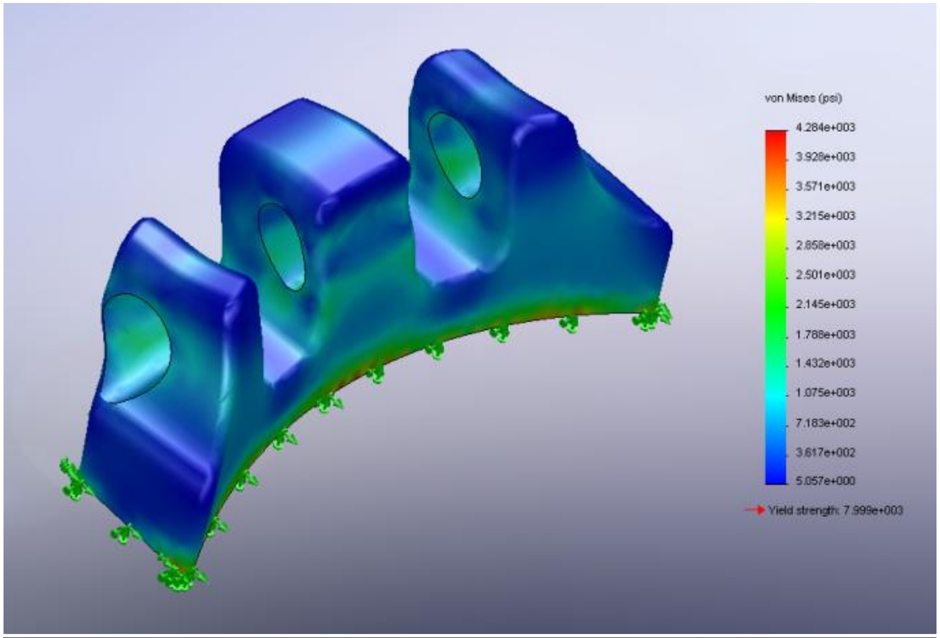


Figure 11e

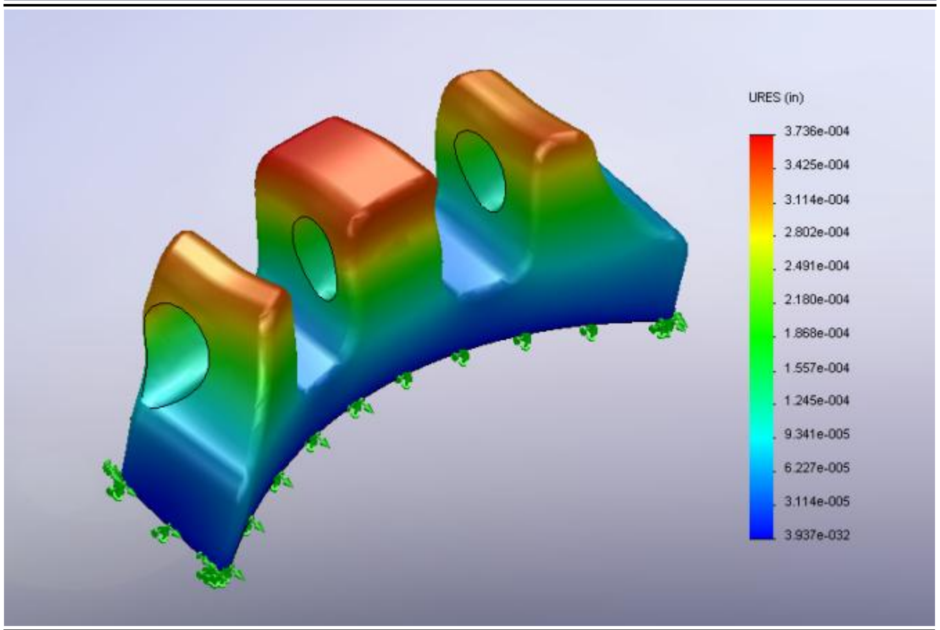


Figure 11f

Figure 11e depicts the Von Mises stress in the bracket under a given load. Figure 11f represents the deformation in the bracket. The restraints on the bracket are representative of a welded base. The applied force is a bearing load of 250 pounds in the horizontal direction.

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.

	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

Table 3:
House of Quality for
Material Selection

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. Adviser 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 this decision will likely be governed by supplier availability.

Current Progress

On-the-water testing in the kayak to determine exact loads and dimensions has been the focus of the group for the past two weeks. Several methods were used to obtain these measurements. Prototyping was the most beneficial method of visually seeing clearance issues and gathering specific dimensions. After recording these dimensions, 2D drawings were generated in AutoCAD with accompanying stress calculations in MathCAD. The experimental loading did not significantly change preliminary design drawings, but did require increased thickness in all hinges to account for the slightly increased loading. The calculations attached to this report are based on actual loading determined during testing on the water.

After revising 2D drawings and calculations, Pro-Engineer and Solidworks were used to generate 3D models and perform FEA to confirm all hand calculations. For example, the FEA performed on the hinge between the Mirage Drive and connecting rod showed a maximum stress of 11ksi compared with 9ksi for bearing stress. This is a difference of 19%, which would be unacceptable to use as a basis for design, but it confirmed that the FEA loading was likely accurate. The FEA also highlighted easy fixes for areas of high stress concentrations in our hinges. This allowed refinement of the detailed design relatively quickly without performing any kind of destructive testing.

Future Goals

The main focus of the next week will be to order all the necessary material and parts. To do this, the group will need to finalize the overall design of the system and have all designs approved by both the client and faculty advisors. Once these designs have been approved, the materials and parts can be ordered.

Once the group has received the necessary materials, construction of the final product will commence immediately. The machining of the final parts should have a much lower lead time than that of the prototype parts. This improved lead time will be due to the group technical liaison, Brian Back, having contact with a third-party machine shop that will donate machining time to the group. Once the finished parts are obtained, the system will be assembled and fitted to the kayak to ensure proper functionality for the design exposition.

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 will be 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. 11 weeks have elapsed and a large portion of the project is completed. Each member of our group understands the importance of communication, both within the group and with the clients and advisors in order to complete the project on time. A current timeline, with a few modifications based on actual data, is shown as Table 4 below.

	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																	

Table 4: Current Timeline

From this timeline it is easy to see that the project is progressing on time in every area except for one: ordering materials. The primary delay in ordering materials has been unexpected delays in machining prototype parts which delayed prototype testing. The prototype testing was essential to confirming stress calculations before ordering materials. Now that testing has been completed, the final design is nearly complete and parts and materials can be ordered first thing next week (week of March 30).

The group has been comparing actual progress with the timeline shown above in Table 4 at every group meeting. This process has allowed the group to focus work on critical path items so the final deadline is 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 may be moderately complex, but tolerances can be 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 will be used were found. The current breakdown of the budget is as follows:

Aluminum Bar Stock	$50
Aluminum Plate	$100
Aluminum Tube	$60
Threaded Fasteners	$10
Quick Disconnect Pin	$25
ER 5356 Aluminum Welding Rods	$15
Rod Ends	$80
Compression Springs	$10
Shipping	$85
Anodizing	$65
Machining Costs	$65/hr
<u>Current Total</u>	<u>$500 + Machining Cost</u>