GCROA Alternative Motorboat Project: Lower Unit Design

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Executive Summary

Our team is from the *College of Technology and Innovation* at Arizona State University Polytechnic. We have been contracted by the *Grand Canyon River Outfitters Association* (GCROA) to design and manufacture the lower-end of an electric motor system, which will hopefully replace the use of fossil fuels in the future. Wanting to convert the current gasoline engines to an electric drive system provides us with the opportunity to redesign the lower-end. Working in conjunction with the University of Utah these new propulsion units will be designed to have an exceptional ability to withstand the unique river conditions of the Grand Canyon that cannot be found in any off the shelf solution. The University of Utah is designing the power source for the propulsion unit, which our lower-end will be designed to adapt to and have to prove functional in order to gain consideration for future use.

By not using a gasoline engine we are able to simplify the lower-end tremendously, removing many unnecessary parts which will no longer be needed by using an electric motor. The outfitters can benefit from a redesigned lower-end suitable for an electric drive system because we are also able to address various weaknesses and failures the outfitters encounter on the river. These weaknesses were brought to our attention by receiving feedback from outfitters through surveys we created and having one on one interaction with some others. The points in question on the current design include failure to shafts, propellers, water pumps, and woodruff keys just to mention a few.

The main aspects of our new design are those in which can directly benefit the outfitters. These aspects include but are not limited to low required maintenance, able to better withstand lower-end impacts, an easily assembled and disassembled unit with fewer parts, and easy reparability both on the river and in the shop with minimal tools required. Based on the criteria gathered from the outfitters along with our own benchmarking, this new motor design will need to be affordable, durable, and fully functional with respect to the current lower-end being used all while remaining environmentally friendly and emitting as little noise as possible.

We have explored alternatives such as hydraulics, belt drive, rotary flex shafts, and direct shaft couplings. Based upon our four alternatives we have decided upon a final design using a direct connect shaft coupling. The main points of our design will encompass a torque limiting clutch, a vertical and horizontal shaft, spiral bevel gears, and thrust bearings. These will all be protected by a "bullet proof" streamline, fully designed and manufactured in house case.

From our design, we will move to the manufacturing aspect of the design where we will manufacture, construct, and test our design. We have every intention of having a fully functional prototype to test on the Colorado River at the final design review in May 2010.

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Introduction:

The Grand Canyon River Outfitters Association (GCROA) is a non-profit trade group based out of Flagstaff, Arizona[1]. GCROA represents the seventeen river outfitters who offer the public with one of the most unique and truly special backcountry river experiences available[1]. Many of these outfitters are family businesses whose ancestors, decades ago, were the first to pioneer the waters of the Grand Canyon which today is now recognized as the sport of recreational river running[1]. For the most part, these outfitters offer the same services and experience, all while maintaining their own personal experience.

Current Operations and Equipment

All of the outfitters use the same standard watercrafts. These motor rigs were first assembled sometime in the 1950's from World War II military bridge pontoons[1]. These boats are referred to as "S-rigs" which utilize large inflatable pontoons strapped to two large aluminum frames[1]. The frames provide a rigid structure and stability on the river[1]. The frame also allows for the attachment of the *Jackass Mount* which is used to hold the gearing and motor. The *Jackass* is a crucial part of the rig's body structure. It allows for the best maneuverability in the water, provides for easier maintenance with its ability to vertically lift out of the water, and its vertical movement also allows for the motor to be lifted out of the water to avoid rock strikes and plunging the propeller into a sandbar. These rigs sit between 34 and 36 feet in length and are set up to carry approximately fifteen to eighteen passengers per trip[1]. Aside from the passengers, they also need to be self-sufficient, carrying enough food and supplies to last the duration of the trip. They carry a minimum of fifteen gallons of gasoline and one fully assembled spare motor in reserve[1]. When all said and done, a fully supplied rig at the beginning of a trip can weigh between eight and twelve thousand pounds[1]. The figure below shows a general overview of the layout of the S-rigs used for the white water sections of the Colorado River.

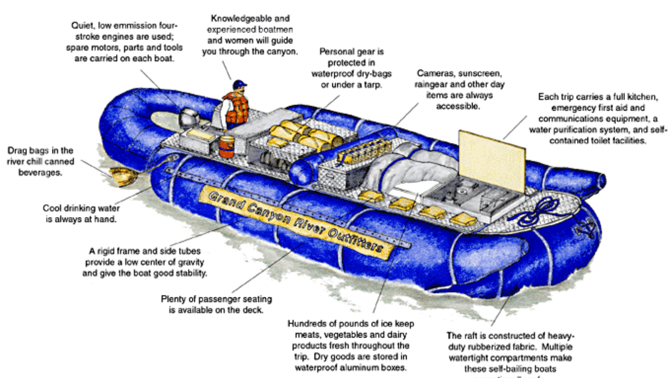


Figure (1) Standard S-rig[1]

Currently, each rig is powered by a 4-stoke outboard gasoline motor, supplied either by Honda, Tohatsu, or Yamaha, which can be seen just below. The pictures were taken straight from the manufacturer's website. These outboards weight approximately 180 lbs[1]. These motors have a maximum output of 30-horsepower[1]. They run on average four to six hours per day but at times may need to run up to eight or ten hours and burn approximately fifteen gallons of gasoline per trip, like previously stated[1]. The motorboats travel the river at eight to ten miles per hour, just slightly faster than the river speed[1]. During normal cruising operations, one-third to one-half of the throttle is used[1]. However, when rafting through rapids, full throttle is used quite often for easy maneuverability and avoiding possible hazards. Prior to 1997, outfitters used 2-stroke outboard gasoline motors[1]. These motors were an industry standard for the time period. On the other hand, these motors were very noisy, less efficient, and emitted a very high level of pollutants compared to the current 4-stroke designs used.

Figure (2) Tohatsu 30 Horsepower Motor[2]

Figure (3) Honda BF 30 Horsepower Motor[3]

GCROA in Partnership with *NPS*

GCROAs move to 4-stroke motors was the first large step taken in their effort for a more environmentally friendly and also a more customer friendly experience. Due to the fact that *GCROA* operate within the Grand Canyon National Park, it works in close juncture with the National Park Service (*NPS*). The National Park Service simply is the organization responsible for keeping our national parks clean and preserved in such a way that they were found hundreds of year's ago[4]. *GCROA* has embarked on an exciting new effort to develop, test, and implement alternative motorboat technology that will be suitable for commercial riverboat operations[1]. The river outfitters have pledged to support this project financially over the next ten years[1]. On January 1, 2008, the *NPS* issued the *Centennial Initiative*[4]. This proposal gives *GCROA* and the alternative motorboat propulsion project their full support and approval moving forward with the project. Through the *Centennial Initiative Program* the National Park Service has agreed to dollar-to-dollar funds matching of what the outfitters pay throughout their concession contracts[1]. Appendix A contains the *Centennial Initiative Program,* which describes the location, partner, total cost, and a summary in their words of the program[4].

Problem Statement:

The Grand Canyon River Outfitters Association would like to take the next step in moving towards preserving the Grand Canyon and our environment. This technology is meant to and will (when ready to implement) replace the use of fossil-fuels. The University of Utah is responsible for selecting, implementing, and testing the power source for these new systems. It is our job, working in conjunction with them, to design a durable, reliable, and easy to work on system that will need to be comparable to the current 30 horsepower gasoline outboard motors.

Methodology:

Design Process:

Last year several teams worked on finding an electric motor that could replace the internal combustion motor currently used in the outfitters' outboard motors. As a continuation of that project, our job this year is to redesign the lower end of the outboard motor to work with the electric motor. Another issue the outfitters encounter is breaking and damaging parts. The outboard motors are not designed for trips down whitewater rapids and break or damage parts when they hit objects. The major problems the outfitters encounter involve problems with shafts, propellers, water pumps, and woodruff keys. Other areas the wish to improve in are required maintenance, ability to better withstand lower-end impacts, an easily assembled and disassembled unit with fewer parts, easy reparability both on the river and in the shop with minimal tools required, being environmentally friendly and emitting as little noise as possible.

We began the project by creating a Gantt chart to outline our schedule for the semester and to give use deadlines to hold to. This breaks the project down into parts and gives a certain amount of time to each part. It also helps show which parts of the project are dependent upon another. An excerpt from the Gantt chart can be seen in Appendix B. Following the Gantt chart helps keep the group on schedule and puts the allotted tme into perspective and helps the team keep on track to meet the deadline.

We then looked into outboard motors already in use. Two variations of the vertical lower end were found. One started off with a vertical section that then connected to an angled section that leads to the prop. The angled design could be beneficial by taking impacts at an angle and thus having to deal with less force, but at the same time this increases the complexity of the motor and more parts are needed. The other used shaft that is angled down from the back of the boat and has the prop just in the water. This is used in shallow water applications or areas with a lot of debris in the water. A problem with this design is that it sprays water everywhere and the prop and is loud.

Since the outfitters are used to the current 30 horsepower motor in use we decided to use this as the control and make the new designs match the specifications of the 30 horsepower honda motor in use. The main aspects we needed to match are the thrust of the motor, its torque at the prop, and its rotations per minute at the prop. The specifications of the 30 horsepower Honda motor are:

- 30 hp
- 5500 rpm at the motor
- 2.08 gear ratio from the motor to the prop
- The torque at the prop is 28.64 ft*lbs
- Prop spins at 2644 rpm

The first step to designing a new lower end is defining the customer needs. This proved to be difficult because we had no direct contact with the outfitters until the trip to meet them and see the boats in the middle of September. We began by identifying preliminary constraints based on the project description provided at the brginning of the semester. These constraints were:

- $5000 budget
- One semester for design
- One semester to build and test the part
- Attach to the existing mount
- Must be able to steer
- Must be capable of tilting
- Must provide thrust
- Must deliver water to cool the engine.

These were constraints we made to start thre project and make some progress before visiting the outfitters. To get a more solid idea of what the outfitters were looking for we created a survey. The questions were designed to give us a better understanding of what the outfitters consider a problem. However, we also wished to know what the most common modes of failure were as well as what the outfitters liked with the current lower end and what they would like to see improved or added. When we received some of the surveys back we used the answers to the questions to create a new list of customer needs. These were:

- Easily asembled and disassembled
- Low maintenance
- Water-tight lower end
- Withstand impact
- Accessable water pump
- No cavitation
- Cooling for the engine
- Adjustable transom angles
- Uses available parts
- Be able to repair in the field
- Use the same mount size
- Can not weigh more than the current lower end
- Can not exceed the curent cost of the outboard motor
- Must have control

These gave us a better understanding of what the outfitters wished to see the project address. The surveys came back with the outfitters' answers which we then interpreted to create needs. The answers need to be written into concise statements where each statement defines on need. This allowed us to better judge what solutions were feasible, but still had several weaknesses. One is that a survey does not cover all aspects of a projct soonly specific areas are addessed. Another problem is that the question was to be interpreted a certain way, but different people may understand the question differently.

Once we met with some of the outfitters and were able to speak with them, we made our final list of customer needs and turned them into criteria and constraints. We were able to see one of the boats on running and to ask the boat operators how different things worked and what aspects and functions of the motor they preferred. This proved to be the most useful since we were able to ask them questions based on the answers from the surveys and the answers they gave us in person.

We also used the surveys, the current outboard motor, and measurements taken on the trip to create a list of known values that relate to the boat and the motor mount. These values include:

- Maximum speed is 10 mph with the river speed (4 mph above the river speed)
- The motor weighs between 150 to 160 pounds
- Approximately 80 trips per motor
- The distance from the motor mounting point to the back tube is 6 feet
- The saddle is 10in from bolt to bolt
- The distance from the top bolt to the center of the prop is 28.5 in
- Boat weight is approximately 13000 lbs fully loaded with passengers

A constraint is something that must be met for the concept to be a solution. Not meeting a constraint means the concept is not viable. A criteria has a range of values that the concept can fit into. These are used to rank different design concepts. Once several designs that meet the constraints are developed, they are ranked according to how well the fulfill the criteria. This new data was used to create a list of criteria and constraints:

Constraints and Criteria

Constraints	Criteria
1. $5000.00 budget	1. Least possible time for assembly
2. Designed by end of fall semester	2. Least possible time for Disassembly
3. Built tested by end of spring semester	3. Longest possible time before parts replacement
4. Must use current mounting system of outboard motors	4. Housing water intake over time (Or pressure) adequate
5. Must allow for water cooling of the motor	5. Highest possible toughness/strength
6. Must tilt	6. Least possible amount of air produced by propeller
7. Must have enough thrust	7. Temperature of engine remains as low as possible
	8. Motor Angle Range as large as possible
	9. Parts available by fewer suppliers
	10. Number of tools needed for assembly/disassembly as little as possible
	11. Size of motor as small as functionally possible
	12. Weight of motor as little as possible
	13. Cost of motor as little as possible
	14. Thrust after rock strike as high as possible
	15. Hours/cost/amount of new parts needed for repair as little as possible
	16. Number of parts as little as possible
	17. Time it takes to take motor off as little as possible
	18. Highest possible number of different current props in use able to attach
	19. Least possible pollutants ejected over time
	20. Least noise emissions possible
	21. As safe as possible

Table (1) Constraints and Criteria

Needs Prioritization

Through a comprehensive process, the customer needs have been established. The needs must now be used in a way to that they will define the outcome of our design since the ultimate goal is to satisfy these customer needs. In particular, there are two things that need to be done using the customer needs once they have been established; rank them, and develop metrics from them.

Ranking the needs is an important measure to take because it is necessary to realize how much of an impact a need will have on a design compared to another. For example, if you are designing a winter jacket, low weight might be a need but it might not be as important as keeping a person warm. Therefore, you might choose a material that is slightly heavier than another, but has better insulation properties because that was more important. Every member of our team met with the customers on our trip to the Grand Canyon, as well as read the responses from them to our surveys. Therefore, we used a team based process to prioritize the needs. We had established 26 total needs, and so each team member individually ranked the needs 1-26 with 26 being most important, based on what they felt was the importance of each need based on their experiences with the customers. Table (2) below shows the results of the individual team member ranks, the added total, and resultant weight associated with each need.

Needs Importance	Ranked 1-26 with 26 being the most important									
Need	**Individual Ranking**								**Total**	**Weight**
Easily assembled and disassembled	19	22	21	17	19	13	15	17	143	0.070305
Low maintenance	16	21	18	19	20	15	22	15	146	0.07178
Watertight lower end	9	20	17	8	15	16	14	19	118	0.058014
More durable lower end	22	16	22	21	22	20	21	20	164	0.080629
Withstand impact	21	15	20	22	14	19	20	22	153	0.075221
Adjustable transom angles	3	3	1	10	6	13	6	5	47	0.023107
No cavitation	17	6	4	16	11	6	7	8	75	0.036873
Engine must remain within operational temperature	10	14	8	15	7	17	13	16	100	0.049164
Uses available parts	5	5	19	9	12	5	5	14	74	0.036382
Be able to repair in the field	12	7	13	14	18	12	19	18	113	0.055556
As small as possible	7	2	6	4	9	4	4	6	42	0.020649
Weighs as little as possible	11	1	7	13	10	2	3	11	58	0.028515
Costs as little as possible	6	9	11	3	8	1	2	9	49	0.02409
Easy reparability	20	13	14	20	21	14	17	21	140	0.06883
Adequate thrust to get to shore after rock strike	8	10	12	18	5	18	18	7	96	0.047198
Limited number of parts	15	4	16	2	17	7	12	13	86	0.042281
Be able to swap failed parts in field	14	8	15	7	16	10	16	12	98	0.048181
Able to dismount motor easily	4	11	9	5	13	11	11	10	74	0.036382
Clean to the environment	18	17	3	11	2	21	8	1	81	0.039823
As little noise as possible	2	18	2	12	1	8	10	4	57	0.028024
Safe System	13	19	10	6	3	22	9	2	84	0.041298
Highly adaptable to various current props	1	12	5	1	4	9	1	3	36	0.017699
									2034	1

Table (2) Ranking of Customer Needs

The top ten needs are highlighted, with the top 3 highlighted a dark blue, the second three a light blue, and the last four a light green. From table (2), you can see that the overall theme of the most important needs was simplicity. Essentially, what it seems is most important to the customers is that the parts of the motor last long, and are easy to repair once they do fail. Needs such as the product being a safe system, and clean to the environment received low scores even though they are important because they are more of a given and will be implemented no matter what design we choose. Since these needs were going to be taken into account no matter what the design is, we did not want them overriding other needs when it came to how they would affect a design choice, which is why they received a low score. A low score indicates that the needs will have little bearing in a design decision comparatively. The weights found for each need dictate that needs level of importance, and will be used later in the design process to ensure that our design takes into account what is most important.

Metrics and Product Specifications

Developing metrics is the second task that must be completed upon the establishment of customer needs. Metrics are essential to any design because they are what will be used to evaluate the design. Metrics are a numerically quantifiable measure of customer needs. Ideally, there should be a metric for every need to be able to ensure that a design has met a customer need. Certain needs are not quantifiable themselves, such as one of our needs "Easily assembled and disassembled". To make a quantifiable metric out of this need, we split it into two separate metric "Time for assembly" and "Time for disassembly". We have decided that the amount of time it takes to assemble or disassemble our product is an adequate way to measure whether our design meets the need of our customers "Easily assembled and disassembled". We went through every need and created a metric in this way until we had a comprehensive metric list that would be able to be used to sufficiently evaluate our eventual design. The top row of Table (3) shows all of the metrics we developed.

Now that we had developed metrics which can have quantifiable values, we had to establish desired target value ranges. These target values for metrics are known as product specifications or engineering requirements. We establish these target values to be able to evaluate whether our design has attained the appropriate value desired out of each metric. For example, for the metric "Housing water intake over time" we have a unit measure of ounces of water leaked in a week while under water, and our target value is 0. The reason we chose 0 for our target value is because we have an electrical motor inside the housing, and many mechanical components that would corrode. The rest the product specifications can be seen in the bottom of Table (3).

List of product specifications in for Metric-Unit-Product Spec:

- Time For assembly- number of hours- 1 hour
- Time for disassembly- number of hours- 1 hour
- Time before parts replacement- number of hours in use- 1 season
- Housing water intake over time- ounces of water leaked in a week while under water- 0

- Toughness- Energy over volume-
- Amount of air produced by propeller- volume of air produced in an hour of use- None
- Temperature of engine- degrees Fahrenheit- N/A
- Motor angle range- angle in degrees- 0-90
- Parts available by single supplier- % of parts by single supplier over total parts- 100
- Number of tools needed for assembly/disassembly- number of tools- 1
- Size of motor- volume in inches cubed- N/A
- Weight of motor- pounds- 180 lbs.
- Cost of motor- dollars-$5,000.00
- Thrust after rock strike- pounds of thrust- 1310 lbs.
- Hours/cost/amount of new parts needed for repair- hours/dollars/number of parts- Minimal
- Number of parts- number of parts- Minimal
- Time it takes to take the motor off- seconds- 5 min.
- Number of current props in use able to attach- % able to attach over number currently used- 10
- Pollutants ejected over time- Volume in cubic inches in an hour of use- 0
- Sound Pollution- Decibels- > 85
- Safety- number of injuries due to product in a year- 0

Needs-Metrics Matrix

We developed metrics to be able to evaluate our design based on customer needs. We also prioritized our customer needs and gave a weight to each of them. Since metrics are now what will be used for the design, we want the importance that we established for our customer needs to carry over into the metrics. This is done through a need-metrics matrix. The needs-metrics matrix identifies the relationship between a metric and each of the needs, then takes into account the importance of each need in that relationship to produce a new value for the metric that shows its importance relative to the other metrics.

To accomplish this, a column of the needs and a row of the metrics we established. Each cell of the matrix then corresponds to a metric above it, and a need to the left of it. In each cell we input the relationship of the corresponding metric and need using a 0,1,3,9 scale. A score of 0 means that there is no relationship between the two; and a 9 means there is a strong relationship. We used a non-linear scale to ensure that highly important relationships stood far out from lesser relationships. Once all of the relationships were determined, we multiplied each row by the weight of the need representing that row previously found by the need ranking table. All of the values in a column were then added together after being multiplied by the needs weights to get a sum total value of importance for each metric. A weight was then given to each metric by dividing the total for that metric by the total of all metrics added together. T his results of this matrix can be seen in Table (3) with the top 10 again being highlighted in different shades of blue with darker blue representing a higher importance.

Needs	Needs Weight	Time for assembly	Time for disassembly	Time before parts replacement	Housing water intake over time (Or pressure)	Toughness/strength	Amount of air produced by propell	Temperature of engine	Motor Angle Range	Parts available by single supplier	Number of tools needed for assembly/disassembly	Size of motor	Weight of motor	Cost of motor	Thrust after rock strike	Hours/cost/amount of new parts needed for repair	Number of parts	Time it takes to take motor off	Number of different current props in use able to attach	Pollutants ejected over time	Acoustic range	Safety	Angled swamp motor	Large skag motor	Others
		Metrics																					**Benchmarks**		
Easily assembled and disassembled	0.070305	9	9	1	0	0	0	0	0	0	9	1	3	0	0	3	9	1	0	0	0	0			
Low maintenance	0.07178	3	3	9	9	9	1	9	0	0	0	1	0	3	0	9	3	1	0	0	0	0			
Watertight lower end	0.058014	0	0	3	9	9	0	0	0	0	0	0	0	1	0	0	1	0	0	9	0	9			
More durable lower end	0.080629	0	0	9	3	9	0	0	0	0	0	1	0	3	9	0	3	0	0	1	0	1			
Withstand impact	0.075221	0	0	9	9	9	9	0	0	0	0	1	0	3	9	0	1	0	0	1	0	3			
Adjustable transom angles	0.023106	0	0	0	0	0	0	0	9	0	0	3	3	0	0	0	0	0	0	0	0	0			
No cavitation	0.036873	0	0	3	0	9	9	1	3	0	0	0	0	0	9	0	0	0	0	0	1	0			
Engine must remain within operational temperature	0.049164	0	0	3	0	0	0	9	0	0	0	0	0	0	0	0	0	0	0	0	0	1			
Uses available parts	0.036382	0	0	3	0	0	0	0	0	9	0	0	0	3	0	3	3	0	3	0	0	0			
Be able to repair in the field	0.055556	9	9	3	0	0	0	0	0	0	9	0	0	0	3	9	9	9	3	0	0	0			
As small as possible	0.020649	3	3	0	0	3	0	0	1	0	0	9	9	1	0	3	3	3	3	0	0	0			
Weighs as little as possible	0.028515	3	3	0	0	3	0	0	1	0	0	9	9	0	0	0	3	3	0	0	0	0			
Costs as little as possible	0.02409	1	1	9	3	3	1	1	0	3	0	1	1	9	0	9	9	1	3	0	0	0			
Easy repairability	0.06883	9	9	9	9	9	0	0	0	9	9	3	3	3	1	9	9	9	3	0	0	0			
Adequate thrust to get to shore after rock strike	0.047198	0	0	0	0	9	3	0	3	0	0	0	0	0	9	0	0	0	0	0	0	3			
Limited number of parts	0.042281	9	9	3	0	0	0	0	0	3	3	1	1	3	0	9	9	0	0	0	0	0			
Be able to swap failed parts in field	0.048181	9	9	1	0	0	0	0	0	0	9	3	3	0	1	9	9	9	0	0	0	0			
Able to dismount motor easily	0.036382	3	3	0	0	0	0	0	0	0	0	9	9	0	0	0	1	9	0	0	0	0			
Clean to the environment	0.039823	0	0	0	9	3	0	0	0	0	0	0	0	0	0	0	0	0	0	9	0	0			
As little noise as possible	0.028024	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	3	0	0	0	9	0			
Safe System	0.041298	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	9			
Highly adaptable to various current props	0.017699	1	1	9	0	0	3	0	0	1	0	1	1	3	3	3	3	0	9	0	0	0			
	Total	59	59	74	51	75	26	20	17	25	39	43	42	32	44	66	78	45	24	20	10	26			
	Weighted Total	3.080144	3.08014	3.997537	3.137169	4.286136	1.299407	1.149459	0.50933	1.16372	2.312691	1.57227	1.48525	1.47394	2.496065	3.231567	3.803838	2.194208	0.775812	1.036383	0.289089	1.39086	43.765017		
	Metric Weight	0.070379134	0.07038	0.09134092	0.071682115	0.09793521	0.02969054	0.02626433	0.01164	0.02659019	0.05284337	0.03593	0.033937	0.03368	0.057033	0.073839044	0.086915035	0.050136117	0.017726761	0.023680626	0.006605481	0.03178	1		

Table (3) Needs vs. Metrics Matrix

Function Decomposition

The next step in the process is known as a QFD, or quality function deployment. First, functions of a product must be defined in order to perform the QFD. The process we used to identify functions is known as function decomposition. We took a 30hp Honda outboard motor that is currently used by some of the outfitters and looked at every part of it to determine what particular function that part contributes to the overall motor. The goal of this was to determine all of the functions that our outboard motor must perform. That way, if we make a motor that can perform all of these functions, then we have created a complete outboard motor. This is not to be confused by how successful the motor is, as that is determined by the metrics. We broke down our list of functions into major functions and sub-functions that make up each major function.

Functions:

Mount Power Source Adjust motor position Replace motor Connect motor shaft Orient motor Cover Motor Allow power supply connection Withstand operational forces **Allow Tilt** Adjust for storage/trim Allow manual adjustment **Provide Steering** Allow pivoting Provide geometric steering Provide control (human) Provide attitude control surfaces **Provide Thrust** Translate rotation motion (motor) to linear motion (boat)	**Encase Components** Reduce drag Reduce cavitation Sustain operational forces Prevent corrosion Resist water intrusion Allow for other functions **Transmit Power** Couple motor and thrust Withstand maximum torque **Mount to Boat** Connect to motor Hand mount to jackass Sustain operational forces Adjust trim **Provide Cooling** Intake cooling water Transfer cooling water Eject cooling water

Once the functions were determined, they were ready to be used in a QFD matrix, Table (4), to determine their importance by ranking them against metrics using the same system as the needs-metrics matrix. Ones highlighted in orange came out as the top 3 most important, and those in beige are the next 3.

		Motor Lower-End Functions																												
		Mount Power Source								Provide Steering			Transmit Power		Provide Cooling			Mount to Boat			Encase Components						Allow Tilt			Provide Thrust
Metrics	Metric Weight	Adjust motor position	Replace motor	Connect shaft	Orient motor	Cover motor	Allow power supply connection	Withstand operational forces	Allow Pivoting	Provide geometric steering	Control steering (Human)	Provide attitude control surfaces	Couple Motor and Thrust	Sustain Maximum Torque	Intake Cooling Water	Transfer cooling water	Eject cooling water	Connect to motor	Hand mount to jackass/boat	Sustain op. forces	Adjust trim	Reduce drag	Reduce cavitation	Sustain op forces	Prevent corrosion	Resist water intrusion	Allow for other functions	Adjust for storage/trim	Provide manual adjustment	Translate rotational motion to linear motion	
Time for assembly	0.0703791	9	9	3	1	0	1	0	0	0	0	0	3	0	0	1	0	3	0	0	0	0	0	0	0	3	0	0	0	1	
Time for Disassembly	0.0704	9	9	3	1	0	1	0	0	0	0	0	3	0	0	1	0	3	0	0	0	0	0	0	0	3	0	0	0	1	
Time before parts replacement	0.091341	0	9	3	0	0	0	9	0	0	0	0	3	3	0	9	0	1	0	0	0	0	9	9	9	9	0	0	0	0	
Housing water intake over time (Or pressure)	0.071682115	0	0	0	0	0	1	0	0	0	0	0	0	0	9	9	9	1	0	0	0	1	0	9	3	9	0	0	0	0	
Toughness/strength	0.0979352	0	0	9	0	0	3	9	0	0	0	0	9	9	1	1	1	9	0	0	0	1	0	9	3	3	0	0	0	3	
Amount of air produced by propellor	0.0296905	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	3	9	0	1	0	0	0	0	9	
Temperature of engine	0.026264	0	0	0	0	0	0	3	0	0	0	0	0	0	9	9	9	1	0	0	0	3	3	3	0	0	0	0	0	3	
Motor Angle Range	0.0116	0	0	0	0	0	1	1	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	
Parts available by single supplier	0.0265902	0	3	3	0	0	1	0	0	0	0	0	1	0	1	3	1	1	0	0	0	0	0	0	0	0	0	0	0	0	
Number of tools needed for assembly/disassembly	0.05284337	3	9	9	3	0	9	0	0	0	0	0	3	1	0	3	0	9	0	0	0	0	0	0	0	0	0	0	0	0	
Size of motor	0.0359	3	1	0	1	0	0	1	0	0	0	0	0	1	1	1	1	1	0	0	0	9	0	1	0	1	0	0	0	1	
Weight of motor	0.03394	3	1	0	1	0	0	1	0	0	0	0	1	1	0	0	0	1	0	0	0	0	0	1	0	0	0	0	0	0	
Cost of motor	0.0337	0	0	1	0	0	1	1	0	0	0	0	1	3	0	1	0	0	0	0	0	0	1	0	3	1	0	0	0	3	
Thrust after rock strike	0.05703	0	0	9	0	0	0	9	0	0	0	0	9	9	0	1	0	0	0	0	0	1	9	9	0	3	0	0	0	9	
Hours/cost/amount of new parts needed for repair	0.073839044	3	9	9	0	0	1	1	0	0	0	0	3	1	0	9	0	3	0	0	0	0	3	9	3	9	0	0	0	3	
Number of parts	0.086915035	3	3	9	1	0	3	3	0	0	0	0	3	3	1	3	1	3	0	0	0	1	1	3	1	3	0	0	0	3	
Time it takes to take motor off	0.050136117	0	9	0	0	0	3	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	
Number of different current props in use able to attach	0.017726761	0	0	0	0	0	0	0	0	0	0	0	3	3	0	0	0	0	0	0	0	0	9	0	0	0	0	0	0	9	
Pollutants ejected over time	0.023680626	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1	1	0	0	0	0	0	0	9	3	9	0	0	0	0	
Acoustic range	0.006605481	0	0	0	0	0	0	0	0	0	0	0	1	1	0	0	1	0	0	0	0	0	3	1	0	0	0	0	0	3	
Safety	0.0318	0	0	3	0	0	9	3	0	0	0	0	1	3	0	0	1	0	0	0	0	0	0	9	1	1	0	0	0	3	
Totals		33	62	61	8	0	34	41	0	0	0	0	44	38	22	52	25	36	0	0	0	19	47	72	27	54	0	0	0	51	
Weighted Total		2.1173	4.0908	4.2223	0.4561	0.0000	1.8249	2.8407	0.0000	0.0000	0.0000	0.0000	2.9177	2.3823	1.1289	3.2562	1.1909	2.5473	0.0000	0.0000	0.0000	0.8045	2.2028	4.4418	1.8730	3.5943	0.0000	0.0000	0.0000	2.1879	44.08
Weight		0.048033672	0.0928	0.09579	0.01035	0	0.041400632	0.064443556	0	0	0	0	0.06619	0.05404	0.02561	0.07387	0.02702	0.05779	0	0	0	0.01825	0.04997	0.10077	0.04249	0.08154	0	0	0	0.049634318	0.999990668
		0.048034	0.093	0.096	0.01	0	0.04140063	0.06444356	0	0	0	0	0.06619	0.054	0.026	0.074	0.027	0.058	0	0	0	0.018	0.05	0.101	0.042	0.082	0	0	0	0.04963432	1.00070251

Table (4) QFD Metrics vs. Functions Matrix

Some of the columns in the QFD have all zeros in them. T his is not because the functions have no relationship with the corresponding metrics, but rather because we zeroed out the column on purpose so that the particular functions would have no weight in our design choices. Although these functions are a part of an outboard motor, they were functions that we found out were not a part of our concern once we met with the customer. For example, every sub function under the major function "Provide steering" has been zeroed out because we were told that we are to use the existing motor mount and tiller. Since these components dictate the steering of the motor, we no longer have to design for these functions of the motor and therefore should not have an effect on design choices. Several other outboard motor functions became irrelevant to our design as well once we met with the customer.

Solutions Table

Once all of the functions for a product have been determined, you can create a solutions table. A solutions table contains a list of all possible solutions or ways of performing a function. A solution for every function combined together makes a product concept. In other words, a product concept contains a way to perform every function required to be performed by that product. Since the function decomposition identified all of the functions we need to perform, then a solution for each of those functions combined together will form a complete product for us.

To make our solutions table, we looked at every function found by our functional decomposition individually. We brainstormed and listed every possible solution we could think of that would be able to perform that function no matter how good or ridiculous the solution was. Every brainstormed solution under each function is our solutions table. Tables (5) and (6) show our solutions table. The major functions or written in bold with their sub-functions written in red beneath them. Below each sub-function is our list of brainstormed possible solutions for that function. The number in between the red sub-function and our list of solutions is the weight given to that function based on our QFD.

Mount Power Source							Transmit Power	
Adjust Motor Position	Motor Replacement	Connect Shaft	Orient Motor	Cover Motor	Connect to Electric Power	Withstand OP Forces	Couple Motor/Thrust	Sustain Maximum Torque
0.048034	0.093	0.096	0.01	0	0.04140063	0.06444356	0.06619	0.0
Quick Release Tab	Quick Release Tab	V-Belt/ Pulley	Horizontal	Tarp	Cable Port	Material Selection/ Geometry	Flex Shaft	CVT
Hand Wheel	Hand Wheel	Groove Belt/ Pulley	Vertical	Sheet metal	Quick Connect		Rigid Shaft	Material Selection/ Geome
Jack Screw	Jack Screw	Chain	Angle	composite			Belt	NuVinci
Bolt Down	Bolt Down	Timing Belt/ Pulley		Plastic			Chain	Torque Limiting Clutch
Set Screw	Set Screw	Gears					Gears	
Multiple Size Slots	Multiple Size Slots	Splines					Hydraulic	
	Omni Mount	Direct					Pneumatic	
		Coupling					Link Arm	
		Pin						
		Keyway						
		Weld						
		Viscous Clutch						
		Fluid Coupler						
		Plate Clutch						

Provide Steering				Provide Cooling			
Pivot	XY Dynamic Geomet	Human Control	Attitude Control Surfaces	Intake Cooling Water	Transfer Cooling Water	Eject Cooling Water	Connect to Motor
0	0	0	0	0.026	0.074	0.027	0.058
Vertical Hinge	Rudder Shape	Hydraulic	Trim Tab	Scoop	Vacuum Pump	Hole	Quick Connect
Spherical Bearing	Airfoil Shape	Tiller	Ballast	Grate	Positive Displacement Pum	Hose	Hose Fitting
Nozzle Control	Flat Surface	Cable Control	Thrust Tilt	Hole	Capillary Action		
	Positive Chord Length	Remote Control			Hand Pump		
		Electric Actuator			Electric Pump		
		Steering Wheel			Shaft Driven Pump		
		Handlebars					

Table (5) First Half of Solutions Table

Mount To Boat			Allow Tilt		Provide Thrust
Hand Mount to Jackass	Sustain OP Forces	Adjust Trim	Adjust for Storage/Trim	Provide Manual Adjustment	Translate rotational motion to linear motion
0	0	0	0	0	0.04963732
Thumbscrew	Material Selection/Geometry	Pin	Pin	Handle	Propeller
Quick Release		Hydraulic	Hydraulic	Screw System	Paddle Wheel
Magnetic		Ratcheting Gearset	Ratcheting Gearset	Cable	Impeller/ Jet
Latch		Detent Lever	Detent Lever		Radial Vacuum
Bidirectional Screws		Radius Track	Radius Track		Fin
Weld					Surface Piercing Prop
Interchange Mount					Star Wars Prop

Encase					
Reduce Drag	Anti-Cavitation	Sustain OP Forces	Prevent Corrosion	Resist Water Intrusion	Allow for other Functions
0.018	0.05	0.101	0.042	0.082	0
Airfoil Geometry/ Coatin	Anti-Ventilation Plate	Geometry	Anode,Material/ Coating	Seals/ Gasket	Geometry
	Shroud	Shroud		Solid Casing	
		Shield on Hull			
		Shield on Case			
		Spring Force Dampener			
		Break-away Detent			
		Torque Limit Clutch			
		Breakable Props			

Table (6) Second Half of Solutions Table

Concept Combinations

Creating concept combinations (Product concepts) involves ranking the solutions, reducing the number of solutions for each function, and then combining a top solution for each function to form a concept. Each team member individually ranked each solution based on how well that solution would perform the associated function, keeping in mind the importance of customer needs. This ranking was not done on a 0,1,3,9 scale but rather on a normal 0-10 scale (with 0 meaning the solutions would not work at all, and 10 meaning the solution is excellent) because the results are not being used as a weight for other measures like the results of the previous two matrices were. The scores for each team member were then averaged together to determine an average score for each solution. Weights of the functions determined by the QFD matrix were then multiplied by the average score to form a weighted averages solutions table which can be seen in Appendix C. This is how customer needs are linked to our solution choices as illustrated in Appendix C; customer needs are developed and given a weight, metrics were developed and given a weight based on their relationship with customer needs, functions were developed and given a weight based on their relationship with metrics, solutions to functions were generated and weighted based on how well they could perform a function.

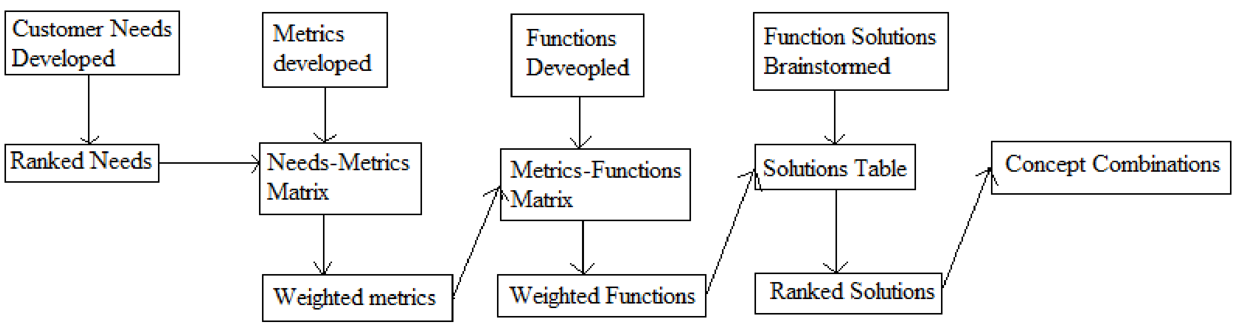


Figure (4) Big Picture of Design Process Used

Based on the averaged score results, solutions were eliminated if they received a low score by a comparatively large margin, see Appendix D. This is where some of the maybe possible but truly ridiculous brainstormed solutions get phased out. Although ridiculous, these solutions were still given a chance to go through the ranking process to avoid any closed minded design decisions. For all the functions that had only one solutions remaining, we combined their total values and created the concept variations table, see Table (7). This left a total of 72 possible concept combinations where we originally had a possible 9.6×10^{14} possible combinations with all of our brainstormed solutions. Each concept combination was ranked by adding together the weighted scores for the solutions that combination contains, see Appendix D. The top 4 solutions are shown in Table (8).

Combination Variations											
Mount Power Source				Connect Shaft (Motor)		Transmit Power		Withstand Op Forces		All Others	
Quick Release Tab	0.48034	Quick Release Tab	0.93	Splines	0.936	Flex Shaft	0.43850875	Material Selection/ Geometry	0.54	Single Solutions	6.23543971
Bolt Down	0.432306	Bolt Down	0.837	Coupling	0.924	Rigid Shaft	0.6619	Torque Limiting Clutch	0.49275		
				Keyway	0.888	Belt	0.43850875				

Table (7) Combination Variations Table

Related Function	Solution 1	Solution 2	Soltuion 3	Soltuion 4
Adjust Motor Position	Quick Release Tab	Quick Release Tab	Quick Release Tab	Quick Release Tab
Motor Replacement	Quick Release Tab	Quick Release Tab	Quick Release Tab	Quick Release Tab
Connect Shaft	Splines	Coupling	Splines	Keyway
Couple Motor/Thrust	Rigid Shaft	Rigid Shaft	Rigid Shaft	Rigid Shaft
Sustain Maximum Torque	Material Selection/ Geometry	Material Selection/ Geometry	Torque Limiting Clutch	Material Selection/ Geometry
Orient Motor	Vert/horiz	Vert/horiz	Vert/horiz	Vert/horiz
Connect to Electric Power	Quick Connect	Quick Connect	Quick Connect	Quick Connect
Withstand OP Forces (Motor)	Material Selection/ Geometry	Material Selection/ Geometry	Material Selection/ Geometry	Material Selection/ Geometry
Intake Cooling Water	Grate	Grate	Grate	Grate
Transfer Cooling Water	Electric Pump	Electric Pump	Electric Pump	Electric Pump
Eject Cooling Water	Hole	Hole	Hole	Hole
Connect to Motor	Quick Connect	Quick Connect	Quick Connect	Quick Connect
Reduce Drag	Airfoil Geometry/ Coating/ Length	Airfoil Geometry/ Coating/ Length	Airfoil Geometry/ Coating/ Length	Airfoil Geometry/ Coating/ Length
Anti-Cavitation	Anti-Ventilation Plate	Anti-Ventilation Plate	Anti-Ventilation Plate	Anti-Ventilation Plate
Withstand OP Forces (Exterior)	All in geometry	All in geometry	All in geometry	All in geometry
Prevent Corrosion	Anode,Material/ Coating	Anode,Material/ Coating	Anode,Material/ Coating	Anode,Material/ Coating
Resist Water Intrusion	Seals/ Gasket	Seals/ Gasket	Seals/ Gasket	Seals/ Gasket
Translate rotational motion to linear motion	Propeller	Propeller	Propeller	Propeller

Table (8) Top 4 Concept Combinations

Solution Alternatives

Looking at the top 4 solutions produced by the process, there is not much variation between them. The variations mainly lay in smaller details of the overall outboard motor. As a result, we kept our top solution produced by this process and dramatically changed the other 3 solutions to explore a wider range of options even though the process showed they wouldn't be the best solutions. The process could also be flawed, making it a good idea to still look into other alternatives that may not have received the top score. The major changes we made to the 3 solutions mainly came in the drive train components or the transfer of power from the electric motor to the propeller. The top 4 concept combinations from the process all came out with rigid shafts to accomplish this function, most likely because of the high importance of simplicity. Still, we decided to look into a hydraulic system, belt drive, and flex shaft drive as well. This made our final 4 alternatives to be taken into detailed design, shown in figure (5).

Revised Concept Combinations		
Solution 1	Solution 2	
Jerrod/Aaron	Matt/ Hugo	Related Function
Quick Release Tab	Quick Release Tab	Adjust Motor Position
Quick Release Tab	Quick Release Tab	Motor Replacement
Splines	Coupling	Connect Shaft
Rigid Shaft	Hydraulic	Couple Motor/Thrust
Material Selection/ Geometry	Material Selection/ Geometry	Sustain Maximum Torque
Clutch	Clutch	
Vert/horiz	Vert/horiz	Orient Motor
Quick Connect	Quick Connect	Connect to Electric Power
Material Selection/ Geometry	Material Selection/ Geometry	Withstand OP Forces (Motor)
Grate	Grate	Intake Cooling Water
Electric Pump	Electric Pump	Transfer Cooling Water
Hole	Hole	Eject Cooling Water
Quick Connect	Quick Connect	Connect to Motor
Airfoil Geometry/ Coating/ Length	Airfoil Geometry/ Coating/ Length	Reduce Drag
Anti-Ventilation Plate	Anti-Ventilation Plate	Anti-Cavitation
All in geometry	All in geometry	Withstand OP Forces (Exterior)
Anode,Material/ Coating	Anode,Material/ Coating	Prevent Corrosion
Seals/ Gasket	Seals/ Gasket	Resist Water Intrusion
Propeller	Propeller	Translate rotational motion to linear motion

Revised Concept Combinations		
Solution 3	Solution 4	
Roark/ Chris	Ryan/ Jason	Related Function
Quick Release Tab	Bolt Down	Adjust Motor Position
Quick Release Tab	Quick Release Tab	Motor Replacement
Keyway	Coupling	Connect Shaft
Belt	Flex Shaft	Couple Motor/Thrust
Material Selection/ Geometry	Material Selection/ Geometry	Sustain Maximum Torque
Clutch	Clutch	
Vert/horiz	Vert/horiz	Orient Motor
Quick Connect	Quick Connect	Connect to Electric Power
Material Selection/ Geometry	Material Selection/ Geometry	Withstand OP Forces (Motor)
Grate	Grate	Intake Cooling Water
Electric Pump	Electric Pump	Transfer Cooling Water
Hole	Hole	Eject Cooling Water
Quick Connect	Quick Connect	Connect to Motor
Airfoil Geometry/ Coating/ Length	Airfoil Geometry/ Coating/ Length	Reduce Drag
Anti-Ventilation Plate	Anti-Ventilation Plate	Anti-Cavitation
All in geometry	All in geometry	Withstand OP Forces (Exterior)
Anode,Material/ Coating	Anode,Material/ Coating	Prevent Corrosion
Seals/ Gasket	Seals/ Gasket	Resist Water Intrusion
Propeller	Propeller	Translate rotational motion to linear motion

Figure (5) 4 Alternatives

Alternatives

Hydraulic

One of the solutions our design process produced was a hydraulic system. As a team we chose not to pursue this alternative for several reasons. After reading customer surveys and gathering constraints and criteria during meetings with the individuals who will be using our product, we found that environmental issues and ease of maintenance were some of the top priorities that needed to be incorporated into the final product.

A hydraulic system would consist of a pump, motor, a storage tank for the hydraulic fluid, and a means for the fluid to travel through the system. Each of these elements would need to be connected either through piping and/ or tubing. In a hydraulic system, it is not if there will be leaks, but when. These connection points provide too great of an opportunity and location for leaks to occur. With the river rafts on the water during operation leaking in the system is not an option. As a team we decided that the environmental impact such a leak could cause would make this solution less than desirable for our customer and an option we would not pursue. *GCROA* stated they wanted these alternative solutions to remain environmentally safe, which in the case of a hydraulic fluid leak would not be.

Keeping in mind that *GCROA* also desires a system that keeps routine maintenance and repair to a minimum there are other disadvantages to a hydraulic system as compared to directly coupling the shaft to the motor. With all the extra equipment needed for a hydraulic system to run efficiently versus directly coupling the motor to the shaft, the operators and mechanics would have extra problems to be concerned with. T he gaskets seal the connection points, maintaining pressure within the system, and the operation of both the pump and motor. If a repair was needed, or failure to any of these components occurred, disassembly and assembly would be much more difficult than our other alternatives.

The size of this alternative is also a disadvantage when compared to a direct mechanical connection. The storage tank alone that is needed would add too much extra equipment which would also need to be maintained. A good estimate of the required volume is 3-5 times the pump discharge rate. For a pump that produces 15 gallons per minute that is a 45-75 gallon tank. A fairly sizeable container that would need to be placed either within a newly constructed housing for the whole system or separately on the boat somewhere. This would either take up space that is currently used for other reasons or create a much larger lower end-unit that would need to be placed differently on the vehicle. Being on the river in a raft with limited area available for customers and gear the size of the re-design is also a factor we considered.

Belt Drive

One of our design combinations called for a belt drive system for primary transmission of power from the motor to the propeller. All possible feasible and even absurd configurations were taken into consideration. The bulk of these included a motor shaft in a horizontal configuration directly transmitting power to the lower horizontal propeller shaft. Using the Lynch motor from the University of Utah 2008-09 design and an 18 inches below water line to prop, keeping the motor mounted above the water for a multitude of reasons (electrical issues, frontal drag area, gross redesign of mounting saddle…etc.) we can easily find that no center to center distance for motor to prop shaft could be less than 20.040 inches, and in good round numbers 20-24 inches for an estimate. Upon initial research we were impressed by their lack of needing lubrication, high power transmission efficiencies, and ability to resist shock loads with deformation (and slip in the case of V-belts). However this was quickly disillusioned by various factors.

To begin the design we looked at what would be required to be backwards compatible with the gas engine type performance output and were quickly discouraged by the requirement of the of a 2.01:1 ratio. This would require the prop shaft pulley to be twice the size of the driven pulley. This presented what we feel is the primary drawback of a pulley/chain design; the required frontal cross-sectional area of the drive unit. The frontal cross section of the unit directly impacts the amount of drag of the unit in the water, and it is our desire to optimize ever aspect as much as possible, thus as small as possible. To demonstrate this we found that an appropriate Gates synchronous belt would require a driven pulley of no less than Ø2.25 inches making the driven pulley nearly Ø4.5 inches. This is with the below water length of 18 inches. The absolute minimum to encase the belt would give a frontal sectional area of 72.49895481 in^2, verses our ideal shaft design with a 31.91620765 in^2 and the stock Honda motor close to 40 in^2. Nearly twice as much drag and and/or significantly increased mass required for the case geometry to create laminar flow, not to mention the diameter of the driven pulley is nearly 50% the diameter of the prop, and with most prop formulas stating that anything past 70% the diameter of a prop does not effectively contribute to thrust this means that means nearly 68% of the propeller will be in a “dirty” water flow condition.

Upon conception of a design specifically for a 1:1 drive ratio we again saw hope in the belt drive. Again using the same synchronous belt, using two of the 2.25 inch pulleys we found an area of 53.569 in^2. This is still significantly greater than even the existing case geometry. Also as before mentioned the required geometry to provide the propeller with clean water flow would be inefficient in material usage and mass to say the least.

One last option for the belt drive would include using tensioning pulleys to make the belt path profile thinner. This by nature is poor engineering, as this would require no less than two accessory pulleys. This would inherently decrease the transmission efficiency, add weight, increased parts, and operate the belt/pulley systems outside of their normal parameters by having increased contact on the primary driver and driven pulleys.

Furthermore the selection on an appropriate belt would require a belt width of 1.35 inches (Gates Corporation). This would increase the geometry lengthwise adding significantly to the volume needed to be incased hydro dynamically. At the water line this area to be enclosed is at least 3.5 in^2, versus that of our shaft .3475 in^2 inches nearly 1/10 the size.

Rotary Flex Shaft

One alternative we investigated was the use of a flexible shaft to transmit power from the electric motor unit to the propeller shaft/ propeller. Flexible power shafts have been used in many diverse industries for various applications, many of which operate under unique conditions. Common applications include those with misaligned shafts, difficult positions, or require vibration dampening and shock absorption. This wide range of applications and uses is made possible through the unique construction of the shaft itself. The shafts are constructed by tightly wrapping a wire into a cylindrical shape to form the shaft profile. Depending on the application, one or more wires may be added to this initial winding to provide additional strength in torsion and bending resistance. These additional wires are commonly wound in a direction opposite to the winding below, allowing the shaft to withstand torsion loads in both rotational directions.

For our application we looked at Bi-Directional Steady-Flex® Shafts from S.S. White Technologies[5]. This model shaft withstands torsion loading in both directions, therefore allowing us to reverse the electric motor rotation to generate reverse thrust from the propeller. While investigating the use of flexible shafts three limiting factors became apparent: minimum bending radius vs. allowable torque, and maximum surface speed[5].

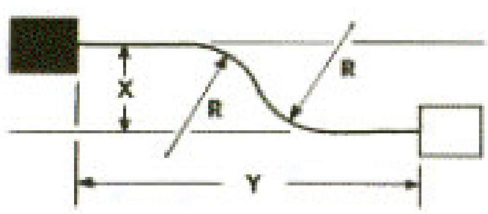


Figure (6) Dimensioning the Rotary Flex Shaft[5]

To determine the minimum bending radius we consulted the published charts provided by the manufacturer. Using 60N*m (44.25lb*ft) as our maximum torque, the required shaft diameter is 18.8mm (.750 inches) with a minimum bending radius of 50.8cm (20 inches[5]). Seen below is the chart directly from the manufacturer's website (S.S. White Technologies) [5]. This chart was used in choosing the correct operating radius needed for our application.

Bi-Directional Shafts Torque vs. Radius Chart
(in Metric Units)

Part	Stiffness	Diameter	Torque Capacity (N-m)									
						Operating Radius (cm)						
		mm	7.6	10.2	15.2	20.3	25.4	30.5	38.1	50.8	63.5	Straight
	High	15.700					25.20	33.79	42.38	50.96	56.16	76.84
750L	Low	18.800						32.88	41.36	49.72	54.69	74.92
750M	Medium	18.800						39.66	49.72	59.78	65.77	90.06
750H	High	18.800						45.65	57.18	68.82	75.82	103.74
1000L	Low	25.100							82.94	109.39	125.21	188.60
1000M	Medium	25.100							109.84	144.75	165.66	249.62
1000H	High	25.100							124.30	163.85	187.47	282.39

Figure (7) Sizing Chart Taken from S.S. White Technologies[5]

One of our design constraints was to remain comparable in overall size to the Honda BF30. With a minimum bending radius of 50.8cm, this original size cannot be maintained while still providing a horizontal propeller shaft. In addition to the small bending radius we require, the maximum surface speed of the shaft would be exceeded using our motor setup.

We calculated the surface speed as follows:

Maximum Surface Speed S_s = 152.4 m/min

$$\text{RPM} = \frac{Ss}{\pi \cdot D}$$

$$\text{RPM} = \frac{152.4\ \text{m/min}}{\pi \cdot .0188\text{m}}$$

$$\text{RPM} = 2580.34$$

With a maximum recommended speed of 2580 RPM, the shaft will not withstand operation at our required 3000 rpm for an extended period of time[5]. Internal friction buildup will cause the shaft to fail prematurely rendering our complete drive system inoperable. This failure mode is unacceptable and further eliminates the flexible shaft as a viable alternative to transmit power.

Final Design Solution

Our design process and study of our possible alternatives has brought us to the decision to choose the use of a direct shaft connection to transmit power from the motor to the propeller. We will use a modular mount to mount the motor to the lower-end. A torque limiting clutch will be used as a large factor against rock impacts and sandbars. Power will be transmitted with the use of shafts and spiral bevel gears. Thrust bearings will also be used to mount the shafts as well as withstand the thrust forces of the propeller. A case will be designed to incorporate all of the internal components on the propulsion system.

Motor Mounting

We have been supplied with the power unit for this project. The Lynch Motor (LEM-2x2-D135) will be used to supply the power. This is the same motor as was used last year, however this year in order to gain double the performance at the same operating speed, the motors will be stacked, one upon the other. Below is a rendering of the Lynch motor.

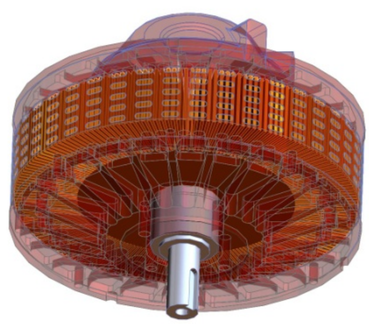


Figure (8) Lynch Electric Motor

One of the requests of *GCROA* was that the design allow for using multiple or various brands and styles of power units. We decided that if the outfitters wanted to switch power units it should be as easy as possible, especially if it *needed* to be done while on the river. The team came up with, what we felt, was the best option for running multiple setup variations as well as ease of changing the motor and disassembly. A flat modular plate will be used to connect the motor to the lower-end. The motor will bolt directly to the plate and the plate will be held on by a lock ring mechanism. The ring will slide

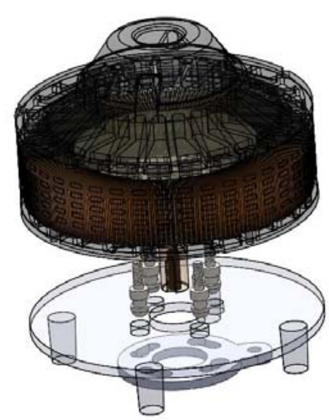


Figure (9) Modular Mounting Lock Ring Plate

over the head of the mounting bolts and then be turned, allowing the bolts to slide in the slots. The forces exerted on the plate from the motor will be enough to hold the motor in place. A clear visual representation can be seen in Figure (9).

Material Selection

Our primary course of action was geared towards learning the needs and wants of our customers; namely, our sponsors in the *Grand Canyon River Outfitters Association (GCROA).* Their requests and suggestions were held in very high priority when taking material selection into consideration. We thoroughly interviewed and surveyed the various owners and boatmen that operate the J-rigs down the rapid sections of the Grand Canyon, this led us to obtain wish-lists, collect data and notice some of the trends between different operators and outfits. With regards to material selection, we are gladly catering to the requests of *GCROA* by designing our prototype to fit the solutions we obtained while learning about our customer and applying materials engineering to them. Some of *GCROA*'s requests were as follows:

1. There was a strong desire for "beefier" shafts to transmit power from the electric engine down to the propeller. Previous and current shafting schemes on the Honda and Tohatsu platforms are adequate but tend to bend in rock-strike situations.

    a) We chose material that met those criteria and some others that our team felt most indispensible for a quality product.

    b) Our selection was a 440C series Stainless Steel. Our team chose this material to go on both of our shafts (motor and propeller shafts).
        - Stainless steel is resistant to the marine conditions that we will be submitting the metals to.
        - The yield strength and ultimate tensile strength of 440C stainless steel is exceptional, and even more so after receiving a proper heat-treating, thus allowing larger safety margins and robust design.
        - Stainless steel can treated to render surface finishes that are hard and resistant erosion due to sediments.
        - 440C is available in our area within few-days order from a reliable and trusted source here locally.
        - 440C stainless steel is expensive, but worth its value because of the enhanced properties.

2. The GCROA operators and owners also stressed the need to improve the design on the casing with regard to rock impacts. In particular our host, Steve Hatch asked for a more robust casing that "would not spill-out a quart of water, when he's supposed to be servicing out a quart of oil." Overall we speculated that the whole casing needed to be upgraded in regards to materials in order to make the unit more impact resistant. We researched metals in the vicinity of aluminum for possible candidates to replace the currently die-cast aluminum lower-end casing and we came back with a few different "marine" application metals that fit the application.

    a) Obviously we need a metal that has very little or no adverse reaction when placed in a corrosive environment such as water. We needed a metal that would not need constant attention in regards to corrosion repair and very little when it came to paint and surface coatings.

    b) We needed a material that would be tough enough to withstand the rock impacts that are incurred in the Colorado River's most dangerous rapids. The mechanical properties must present enough strength, and durability to reduce the occurrence of cracked casings and critical damage incidents by a significant percentage.

    c) There was a huge demand for a casing material that was repairable and re-usable. With that in mind, this meant we had to choose a metal that was easy to weld somewhat flexible and ductile, and versatile enough to be easily formed, machined or casted.

        - Thus far we see no better alternative than that of our well known friend and companion material 6061-T6 Series Aluminum. We are very familiar with the properties of this metal within our engineering training since we constantly utilize it for practice in machining.

        - 6061-T6 offers excellent material properties in return for a modest and readily accessible price.

        - It can be purchased in many manufactured shapes and forms and is available in almost every metal supplier in the valley.

        - The metal is completely tolerant of marine environments and its mechanical properties are decent enough for our particular use.

- We needed to choose a metal that would withstand the machining and forming tasks of our manufacturing process. 6061-T6 can be welded to a certain extent; it machines like butter and can cast readily into all sorts of complex shapes.

Overall, these are the two big areas where we are applying material selection came into major discussion. Other areas were limited to the selection of off-the-shelf parts and qualifying them for their usage ratings and limitations. Material selection was limited in a sense in the remaining components of the lower end design, we did the selection by proxy of acquiring parts that were rated for our purposes and thus did not have to do much research into those parts since they came in on the manufacturer's datasheets. Tables were constructed of various materials that were compared as potential materials to use in the final design of both the shafts and the case. These tables can be seen in Appendix F.

Clutch Design and Analysis

Torque limiting clutches eliminate the risk of a torque overload to the motor. The clutches can be ordered to disengage at predetermined loads or they are made to be adjustable. Based on the specifications provided by University of Utah, the clutch must operate in a range of 0-3000rpm and must be able to handle a load of 60Nm. Based on the motor operating characteristics two lead manufactures were looked at, R+W Coupling Technology and Mayr. Both manufactures offer high precision couplings for a large range of applications. The two companies has a large selection of clutches that operate at the given torque range, however only Mayr had clutches that will work with high-speed applications.

Mayr is number one in torque limiting clutches worldwide. The company has been producing high quality clutches since the 1950's[6]. There are three variations in clutches depending on the application, ratcheting, synchronous, and free running. The free running clutch was looked at but quickly ruled out due to the need of manual re-engagement once it is disengaged. The ratcheting and synchronous automatically re-engage when tripped by an overload[6]. All of the clutches are maintenance free[6]. They do not require regular lubricant changes, which is great for environmental impact. Torque limiting clutches also provide instant separation on overload and are backlash free[6]. This will prevent severe damage to shafts, motors and gears in the event of a rock strike at full throttle[6].

The ratcheting type clutch is easily adjustable with a readable torque adjustment. This type of clutch works by two spring creating pressure on a plate[6]. Below is a schematic of the torque limiting clutch, taken from the Mayr website, which shows how it is read and can be adjusted[6].

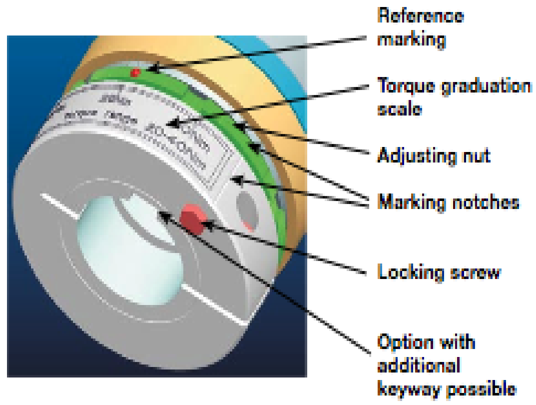


Figure (10) Adjustment Schematic of TLC[6]

The plate has 18 detents for ball bearings to seat in[6]. The spring creates pressure on the plate locking the ball bearing in the detents. As the shaft rotates under normal conditions the ball bearings remain seated transmitting power to the shaft. When there is a torque overload present, the ball bearings slip out of place allowing the assembly to rotate freely around the shaft immediately removing the overload. Every 15 degrees the clutch will try to re-engage but if the load is still present the shaft remains spinning free[6]. The cycle life of the ratcheting clutch is 150,000 disengagements, with each slip out of the detent counting as one full disengagement[6]. This style clutch has 18 ball bearings, which equates to 8,333 full revolutions under torque overload[6].

The synchronous clutch type also has readable torque adjustments. The operation of this clutch is very similar to the ratcheting except for the synchronous clutch re-engages precisely after 360 degrees[6]. With fewer ball bearing detents, the clutch life cycle is increased (no exact numbers given by manufacture). The synchronous has better dynamic characteristics with wider operating rages of 2.7-500Nm depending on the series selected[6]. This clutch also provides better vibration dampening and is more cost effective.

For this application a synchronous clutch has been selected. The clutch chosen, model 484.625.4/18/12.7[6]. The specification sheet for this particular clutch can be seen in Appendix E. This clutch has an operating speed rating of 3000rpm and a torque range of 35-65Nm[6]. In the event of a rock strike, the torque impulse created will be immediately eliminated by the clutch disengagement. When the load is no longer present the clutch will re-engage precisely after 360 degrees providing power to the output shaft[6]. This in return will prevent drive shafts from bending and twisting which are two main causes of motor failure.

A keyway connection was chosen over coupling due to higher reliability and ease of installation. Couplings often fail resulting in a free spinning shaft damaging the contact surface on the shaft/inner bore of the clutch. The inner bore of the clutch (side attached to the shaft output from the motor) will be 19mm with a 6mm key way. The inner bore for the output shaft (down to the gearset) will be 13mm also with a 6mm keyway:

$N = 3 \qquad \tau = 65\ \text{Nm} \qquad D = 19\ \text{mm}$

$w = 6\ \text{mm} \qquad S_y = 490 \times 10^6\ \text{N/m}^2$

$$Length\ of\ Keyway\ L = \frac{4 \cdot \tau \cdot N}{D \cdot w \cdot Sy}$$

$$L = \frac{4\ \cdot 65Nm \cdot 3}{19mm \cdot 6mm \cdot 490 \text{ x } 106 \text{ N/m2}}$$

$$L = 21mm$$

For the keyway, zinc plated 1040 cold drawn steel will be used. The required length of the key was calculated to be 21mm in length (with a safety factor of 3) designed to shear at a 65Nm, which is the maximum designed torque load for the clutch:

$N = 3 \qquad \tau = 65\ \text{Nm} \qquad D = 12.7\ \text{mm}$

$w = 6\ \text{mm} \qquad S_y = 490 \times 10^6\ \text{N/m}^2$

$$Length\ of\ Keyway\ L = \frac{4 \cdot \tau \cdot N}{D \cdot w \cdot Sy}$$

$$L = \frac{4\ \cdot 65Nm \cdot 3}{12.7mm \cdot 6mm \cdot 490 \text{ x } 106 \text{ N/m2}}$$

$$L = 21mm$$

The keyway is designed to shear at the maximum torque load to prevent severe damage to the motor and shafts if the clutch malfunctioned.

Shaft Design and Analysis

Vertical Shaft Design

The vertical shaft is subject to two stresses: the torsional stress from the torque of the motor and the compressive stress generated by the gears between the vertical and horizontal shaft pushing apart.

The compressive stress is the force from the gears divided by the area of the shaft:

$$\sigma = 4 * \frac{100lb}{\pi D^2}$$

Where D is the diameter of the shaft

The torsional stress is:

$$\tau = T * \frac{r}{J}$$

Where T is the torque from the motor, r is the distance from the neutral axis, and J is the polar moment of inertia. $J = \pi * \frac{D^4}{32}$ was used for a solid circular shaft.

The torque of the motor is 60 N*m, or 531 in*lbs and r is $\frac{D}{2}$.

We then used the Von Mises equation because the shaft is subject to both torsion and compressive shear stresses. This accounts for both stresses by creating a modified stress that is the combination of the two. The Von Mises equation for this case is:

$$\sigma = \left(\left(4 * \frac{100lb}{\pi D^2}\right)^2 + 3\left(532inlbs * \frac{16}{\pi D^3}\right)^2\right)^{\frac{1}{2}}$$

We then used the yield strength of the stainless steel, which is 270000 psi, divided by three as □ in order to solve for the minimum diameter needed to sustain these stresses. We divided by three to cut the allowable stress in the material by a third because we are using a safety factor of three here.

This yielded D = 0.373in for the minimum diameter.

We then checked to see what the minimum diameter required to avoid buckling is for the 100lb compressive force. The equation to find the critical load for a given beam for buckling is:

$$P = \pi^2 * E * \frac{I}{L^2}$$

Where P is the critical load, E is the modulus of elasticity of the material, I is the moment of inertia, and L is the effective length. E is 27.6 in^2lb, and, since both ends are fixed and not pinned, L is half the length of the shaft. The shaft is 24 in long, so L is 12 in. We then used P=300lb to solve for the minimum diameter needed to hold 100 pounds without buckling with a safety factor three.

$$300lb = \frac{\pi^2(27.6 * 10^6 in^2 lb)\left(\frac{\pi D^4}{64}\right)}{(12in)^2}$$

Solving for D gives D=0.317in as the minimum diameter of the shaft required to prevent buckling. Therefore the minimum diameter necessary for the vertical shaft to sustain the given conditions is 0.373 inches.

Horizontal Shaft Design

The horizontal shaft on the bottom of the lower end connect is placed between the propeller and a bearing. The propeller provides up to a maximum of 1310 lbs of thrust, which the bearing then applies in the opposite direction back onto the shaft. In addition, the gear attached to the shaft in between is providing up to 60 ft-lbs of torque, which is being resisted by the propeller in the opposite direction. Therefore, the section of the shaft in between the gear and the propeller will experience the most stress considering that it will be handling an axial load of 1310 lbs and a torque of 60 ft-lbs. Stress from the axial loading will be evenly distributed throughout the shaft, but torsional stress increases as you move out from the centroid of the part. Therefore, the surface of the rod will be experiencing the most torsional stress in combination with the stress from axial loading. To find the effective stress on the part to determine the minimum shaft diameter needed, we can use Von Mises effective stress theories.

Determining minimum shaft diameter:
Using Von Mises effective stress for uniaxial loading:
Equations:

$$\sigma' = \sqrt{\sigma_x^2 + 3\tau_{xy}^2}$$

$$N = \frac{S_y}{\sigma'}$$

$$\sigma_x = \frac{P}{A}$$

$$\tau_{xy} = \frac{TD}{2J}$$

$$A = \frac{\pi D^2}{4}$$

$$J(Circle) = \frac{\pi D^4}{32}$$

$\sigma' = Von\ Mises\ Effective\ Stress$
$\sigma_x = Nomal\ Stress$
$\tau_{xy} = Torsional\ Shear\ Stress$
$N = Safety\ Factor$
$S_y = Yield\ Strength\ of\ Material$
$P = Axial\ Load$
$A = Area$
$T = Torque$
$r = radius$
$J = Polar\ Moment\ of\ Inertia$
$D = Diameter$

Derivation for minimum diameter:
Substitute A into normal stress and J into shear stress and you get

$$\sigma_x = \frac{4P}{\pi D^2}$$

$$\tau_{xy} = \frac{16T}{\pi D^3}$$

Enter these into the uniaxial Von Mises effective stress equation

$$\sigma' = \sqrt{\left(\frac{4P}{\pi D^2}\right)^2 + 3\left(\frac{16T}{\pi D^3}\right)^2}$$

Rewritten and slightly simplified

$$\sigma'^2 = \frac{16P^2}{\pi^2 D^4} + \frac{768T^2}{\pi^2 D^6}$$

Both sides multiplied by $\pi^2 D^6$

$$\sigma'^2 \pi^2 D^6 - 16P^2 D^2 - 768T^2 = 0$$

From this equation, you can plug in values for σ' ,P, and T and solve for the minimum diameter.

Knowing the material and our desired safety factor, we can solve for a Von Mises effective stress which represents the stress that our part must be under in order to prevent failure:
$N = 3$
$S_y = 270{,}000psi$

$$\sigma' = \frac{S_y}{N} = \frac{270000psi}{3} = 90{,}000psi$$

We know the Von Mises effective stress, axial load and torque being applied to the shaft, now we can use the equation found above to find the minimum diameter required.
$P = 1310lb$
$T = 60\ ft \cdot lb$

$$(90{,}000psi)^2\pi^2D^6 - 16(1310lb)^2D^2 - 768(60ft \cdot lb)^2 = 0$$

$$D = 0.414in$$

This means that the minimum diameter our horizontal shaft should be is 0.414inches in order to prevent failure.

Bearing Design and Analysis

Why bearings are needed:

The general concept we chose involves using shafts to transmit power from the electric motor to the propeller. Minor design also indicated further that we would need a 90degree bend in rotational axes from the motor to the prop, i.e. the rotational axis of the motor is orthogonal to the rotational axis of the propeller. Investigation into how we were going to accomplish this led to a simple two shafts angled 90degress to each other, connected by a set of helical bevel gears (See gear selection). Knowing this allows us to see all of the forces acting on our shafts.

Figure (11) Vertical and Horizontal Shaft with Thrust Bearing

In addition to weight, the horizontal shaft has two forces acting on it: the thrust form the prop, and force from the gears trying to separate at their pressure angle. The vertical shaft only has the force from the gears and its weight. To be able to accommodate these forces, and still allow the shafts to rotate freely, bearings became the clear option. In order to determine what kind of bearings size them, we needed to find the magnitude of these forces.

Determining Thrust from the Propeller:

Equations for propeller coefficients as defined by NACA:

$$Thrust\ coefficient\ c_t = \frac{T}{\rho \cdot n^2 \cdot D^4}$$

$$Power\ coefficient\ c_p = \frac{P}{\rho \cdot n^3 \cdot D^5}$$

$$Advance\ Ratio\ J = \frac{v}{n \cdot D}$$

$$Efficiency\ \varepsilon = J \cdot \frac{c_t}{c_p}$$

T=thrust
ρ=density
n=revolutions per second
D=diameter of the prop
P=power
V=velocity

Solving for thrust:
Rewrite thrust coefficient equation to form thrust equation

$$T = c_t \cdot \rho \cdot n^2 \cdot D^4$$

Rewrite efficiency equation

$$c_t = \frac{c_p \cdot \varepsilon \cdot n \cdot D}{v}$$

Substitute c_t into thrust equation and simplify

$$T = \frac{c_p \cdot \varepsilon \cdot n \cdot D}{v} \cdot (\rho \cdot n^2 \cdot D^4)$$

$$T = \frac{c_p \cdot \varepsilon \cdot n^3 \cdot D^5 \cdot \rho}{v}$$

Substitute in power coefficient equation and simplify

$$T = \frac{\frac{P}{n^3 \cdot D^5 \cdot \rho} \cdot \varepsilon \cdot n^3 \cdot D^5 \cdot \rho}{v}$$

$$T = \frac{P \cdot \varepsilon}{v}$$

This equation states that the thrust produced by a propeller is equal to power times efficiency divided by velocity. Interpreting this equation, you could say that as velocity in the denominator increases, thrust decreases. This makes sense considering that a propeller will no longer produce any thrust once it has reached its maximum velocity and can no longer accelerate the fluid coming into it. Conversely, you could say that as velocity approaches 0, thrust approaches infinity. However, efficiency in the numerator is also a function of velocity, i.e. $\varepsilon=\varepsilon(v)$, and looking at efficiencies of propeller shows that as velocity approaches 0, efficiency also approaches 0[7]. Figure (12) is an example of a graph of the efficiency of propellers with respect to the advance ratio J[7]. The advance ratio is the distance a propeller has advanced after one full revolution (v/n) divided by the propeller's diameter (D)[7]. This particular graph shows several of these curves, each one representing a propeller with a different blade angle.

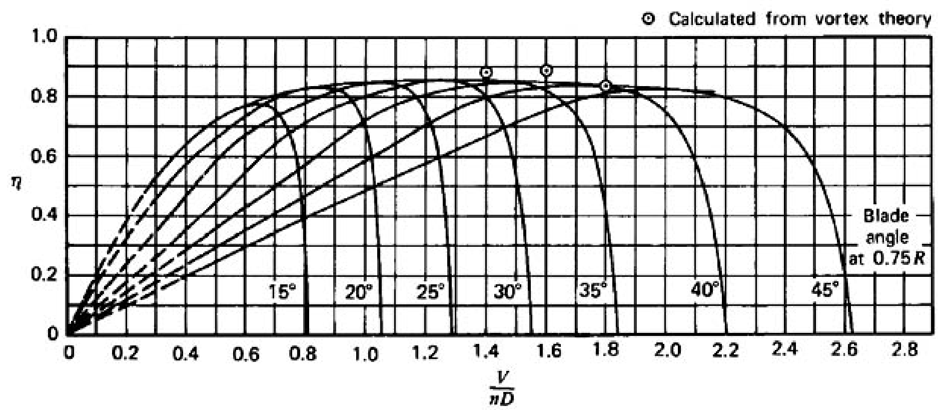


Figure (12) Example Propeller Efficiency Graph[7]

We do not have efficiency graphs for our propeller, so we had to develop an equation that could find the maximum possible thrust knowing only the physical properties of the propeller, and the power being supplied to it. Theoretically, the maximum possible thrust occurs under static conditions, i.e. v=0. This is not actually the case though when considering total efficiency (Propellers produce approximately 80-90% of their dynamic theoretical maximum, but only about 50% of their static theoretical maximum)[7] but we can use this theoretical static maximum for design, and all actual thrust values will be lower.

Solving for maximum static thrust:

Rewrite the thrust equation for efficiency

$$\varepsilon = \frac{T \cdot v}{P}$$

This equation can be adapted for what is known as the figure of merit (FOM). The FOM describes the power efficiency of a propeller and is defined as the ratio of the power being produced (Transferred to the fluid) over the power supplied to the propeller. An FOM of 1 means that 100% of the power being supplied to the propeller is being transferred to the fluid.

$$FOM = \frac{P_t}{P_s}$$

P_t=Power transferred
P_s=Power supplied

An adaptation of the equation, Power equals force multiplied by velocity, for propellers yields:

$$P_t = T(v_p + v_i)$$

v_p=Velocity of the propeller (And whatever it is firmly attached to)
v_i=Velocity induced at the propeller

Substitute P_t into the FOM equation

$$FOM = \frac{T(v_p + v_i)}{P_s}$$

Solve for T (Thrust)

$$T = \frac{FOM \cdot P_s}{v_p + v_i}$$

We are solving for static thrust, therefore $v_p = 0$ and the equation can be reduced

$$T = \frac{FOM \cdot P_s}{v_i}$$

In order to find the static thrust, we need to find what the induced velocity (of the fluid) is at the propeller. This information is not given to us, so we can look fluid mechanics equation to find a relationship between the induced velocity and thrust, beginning with Newton's second law equivalent in fluids.

$$T = \rho \cdot A \cdot v \cdot \Delta v$$

A=Area of the circle the propeller encompasses over a full rotation
v=Velocity at the propeller
Δv=Total change in velocity of the fluid by the prop

The velocity at the propeller is equal to the velocity of the propeller itself plus the induced velocity, and the induced velocity at the propeller is about half of what the total change in velocity that the fluid will experience, so

$$v = v_p + v_i$$

$$v_i = \frac{1}{2} \cdot \Delta v$$

Plug these into the above thrust equation

$$T = \rho \cdot A \cdot (v_p + {}^1/_2 \cdot \Delta v) \cdot \Delta v$$

Rewrite

$$\frac{T}{\rho \cdot A} = v_p \cdot \Delta v + {}^1/_2 \cdot \Delta v^2$$

$${}^1/_2 \cdot \Delta v^2 + v_p \cdot \Delta v - \frac{T}{\rho \cdot A} = 0$$

Use the quadratic equation to solve for Δv (Only need to consider the plus for this case)

$$\Delta v = \frac{-v_p + \sqrt{{v_p}^2 + 4 \cdot \frac{1}{2} \frac{T}{\rho \cdot A}}}{1}$$

Substitute in v_i

$$v_i = \frac{-v_p + \sqrt{{v_p}^2 + 4 \cdot \frac{1}{2} \frac{T}{\rho \cdot A}}}{2}$$

Again this is for a static case so $v_p = 0$ which reduces the equation

$$v_i = \frac{\sqrt{4 \cdot \frac{1}{2} \frac{T}{\rho \cdot A}}}{2}$$

Simplify

$$v_i = \sqrt{\frac{T}{2 \cdot \rho \cdot A}}$$

Now that we have the induced velocity, we can substitute it into the equation for thrust as a function of FOM

$$T = \frac{FOM \cdot P_s}{\sqrt{\frac{T}{2 \cdot \rho \cdot A}}}$$

Rewritten

$$\sqrt{\frac{T}{2 \cdot \rho \cdot A}} T = FOM \cdot P_s$$

Square both sides then simplify

$$T^3 = 2 \cdot \rho \cdot A \cdot P_s{}^2 \cdot FOM^2$$

$$T_s = \sqrt[3]{2 \cdot \rho \cdot A \cdot P_s{}^2 \cdot FOM^2}$$

This equation represents maximum possible static thrust able to be produced by a propeller. It depends on the density of the fluid, area of the circle a propeller encompasses over a full rotation, the power supplied to the propeller, and the figure of merit. Area can be found by the equation

$$A = \frac{\pi}{4} \cdot D^2$$

Now that we have this equation, we can solve for the maximum possible thrust that can be put on our system.

$\rho = 1000 \frac{kg}{m^3}$ (Density of water)

$D = 0.250825m$ (Diameter of propeller)

$A = \frac{\pi}{4} \cdot D^2 = \frac{\pi}{4} \cdot (0.250825m)^2 = 0.0494119m^2$

$P_s = 44742W$(Power supplied to the prop assuming no losses between the prop and the electric motor)

$FOM = 1$ (FOM is not actually 1, but we will assume it is to get the highest thrust possible for worst case scenario design)

Plugging these values in yields

$$T_s = \sqrt[3]{2 \cdot 1000 \frac{kg}{m^3} \cdot 0.0494119m^2 \cdot (44742W)^2 \cdot 1^2}$$

$$T_s = 5827N = 1310lb$$

We now have the thrust produced by the propeller, but we need to determine the force on the shafts produced by the angle of the teeth where the gears contact one another.

Determining force as a result of bevel gear mesh between the horizontal and vertical shafts:

$$F_t = \frac{T}{r}$$
$$F_s = F_t tan\alpha$$
$$\varepsilon_p = tan^{-1}\frac{Z_p}{Z_g}$$
$$\varepsilon_g = tan^{-1}\frac{Z_g}{Z_p}$$
$$F_p = F_s sin\varepsilon_p = F_{gr}$$
$$F_g = F_s cos\varepsilon_g = F_{pr}$$

$F_t = Tangential\ Force\ on\ pinion$
$T = Torque\ on\ pinion$
$r = Pitch\ radius$
$F_s = Separating\ force$
$\alpha = Pressure\ angle$
$E_p = Pinion\ pitch\ angle$
$E_g = Gear\ pitch\ angle$
$Z_p = Number\ of\ teeth\ on\ pinion$
$Z_g = Number\ of\ teeth\ on\ gear$
$F_p = Pinion\ thrust$
$F_g = Gear\ thrust$
$F_{pr} = Pinion\ radial\ load$
$F_{gr} = Gear\ radial\ load$

Solving for thrust and radial loads:
$T = 60.00N \cdot m$
$r = 0.030m$

$$F_t = \frac{60N \cdot m}{0.03m} = 2000N$$

$\alpha = 20.00\ degrees$

$$F_s = 2000N tan20$$
$$F_s = 727.9N$$

$Z_p = 20.00$
$Z_g = 20.00$

$$\varepsilon_p = tan^{-1}\frac{20.00}{20.00} = 45.00\ degrees$$
$$\varepsilon_g = tan^{-1}\frac{20.00}{20.00} = 45.00\ degrees$$

$$F_p = F_{gr} = F_s sin\varepsilon_p = 727.9N sin45$$
$$F_g = F_{pr} = F_s cos\varepsilon_g = 727.9N cos45$$

$$F_p = F_{gr} = F_g = F_{pr} = 514.7N = 115.7lb$$

Since the gear ratio is 1:1, it turns out the radial and thrust loads of both the pinion and gear will all be equal at a value of 115.7lb.

Bearing Choice

Now that we know the value of all the forces that will be acting on the shafts, we know what our bearings must support to ensure that our shafts do not move. You can see from the values found above that we have significant thrust values to account for, as well as some radial, including the weights of the shafts themselves. The weight of the horizontal shaft will be approximately will be approximately 3lb or 13N, and the weight of the vertical shaft will be approximately 5lb or 22N.

Tapered roller bearings became our choice for bearings because they can support thrust and radial loads while still allowing for high rpm ranges. A set of two, opposite facing tapered roller bearings will be used for each shaft to account for thrusts in either direction and counteract any moments present. The next thing to check was whether there were any bearings that would work within our size requirements. Looking at the Timken products page in the Appendix F, highlighted in the blue boxes are two bearings which would fit our applications well.

First bearing highlighted:
Thrust load capacity - 531lb
Radial load capacity - 1130lb
RPM limit - 4340
Inner diameter (Bore size) - 0.5in
Outer diameter- 1.5in

Second bearing highlighted:
Thrust load capacity - 2020lb
Radial load capacity - 2510lb
RPM limit - 8350
Inner diameter (Bore size) - 0.625in
Outer diameter - 1.8494in

We would need one of the second bearing to handle the thrust from the propeller, and 3 of the first bearing for the rest of the bearing positions. All bearings operate well within load and RPM ranges, and will are small enough to fit within the size requirements imposed by our design.

Gear Design and Analysis

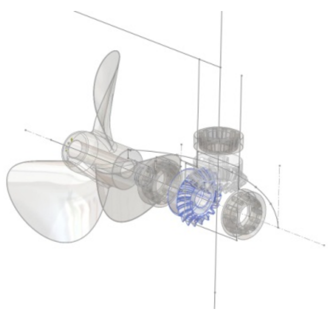
To transmit power from the vertically oriented shaft to the horizontally oriented shaft, a set of gears were chosen to complete the task. The factory gearset has a ratio of 2.08:1 to transform the 6000 rpm from the gasoline engine to the 3000 rpm necessary on the propeller shaft. Through use of an electric motor, we are given the 3000 rpm straight from the motor shaft with no further reduction necessary. Without having to change the shaft speed, we still had to change the direction of the shaft power. This was accomplished through use of a miter gearset. Miter gears are a form of bevel gears that have a ratio of 1:1 and come in a few different tooth forms. For our application, we chose spiral bevel gears due to their smooth, quiet engagement and ability to carry the necessary torque. Through research and conversations with various gear manufacturers, we concluded to choose a set of predesigned gears for simplicity and reliability. To select a gearset suitable for our application, we used the maximum output torque of the electric motor as our worst case scenario. This value of 60N*m simulates the motor operating at full capacity and the propeller shaft being held stationary. Also, due to our design having a Torque Limiting Clutch we do not anticipate encountering torque loads higher than those of the maximum motor torque.

Figure (13) CADD of Chosen Spiral Bevel Gear Assembly

We selected part number MMSA3-20R and MMSA3-20L gears from Quality Transmission Components (QTC) [8]. The gearset is rated to transmit 61.93 Nm of torque before failure due to bending[8]. Although this rated torque is very close to our expected torque load, it will be increased by reducing our lifecycle estimate. Per the manufacturer, the gearset published values is at an expected life cycle of 10,000,000 cycles, a safety factor of 1.2 and support on one end[8]. For every factor of ten we reduce the life expectancy, we can increase the torque capacity by ten percent. Therefore, with an expected life cycle of 1,000,000 cycles, we can use a rated torque of 68.123N*m[8]. By supporting our horizontal gear on both ends, and using our increased rated torque value we increase our factor of safety.

Housing Design and Analysis

One of the final steps in the design process is the design of the external case. This is a highly complex portion of the project which completely relies on all of the other functions of the motor, as form always follows function. However in this project a portion of the case is submerged and traveling through a viscous fluid, water. Thus furthering the complexity of the design as the form is now a function as well. The involved hydrodynamics of the project are relatively simple at the velocities in question but still play an important role. The primary design goal was to minimize the frontal cross section of the case. Using Solid modeling, and Computational stress analysis, we can optimize our design without going through extensive prototype testing as done in the past. The initial prototype is going to be made of a cast aluminum allow, just as the current



Honda is manufactured, with the added strength to accommodate the robust drive train we have developed, and taking into account the extreme environment of the Colorado river. Many of the easy repair features of the original Honda case that the *GCROA* members like, have been maintained in the new design, along increased river survivability, giving rise to a one of a kind “river grade” design. Seen below are a few pictures of the fully assembled lower-end casing, a sectioned view to show the internals within the lower-end case (left), and the water intake slots to provide cooling water.

Figure (15) Water Intake Slots for Cooling

Figure (16) Rendered Lower-end Unit

Future Work

With a solid and final design chosen, we can now look ahead to the second half of this project. First, a new Gantt chart will be created in order to keep ourselves on track to meet our deadline. The second half will entail manufacturing our lower-end design, in full, to produce a fully functional prototype. We will purchase such parts as the torque limiting clutch, bearings, and the spiral bevel gears. We will purchase our raw material in order to manufacture our shafts. The biggest challenge for our manufacturing stage is manufacturing the lower-end case, as we will explore both options of machining and casting procedures.

Once the prototype is completely assembled, we can move on to doing field testing. Ideally, we would like to allow ourselves time to test our design and make any changes or improvements as we see fit. In any case, we will have a fully functional, ready to test, lower-end prototype when we meet again this coming Spring.

References

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< http://www.tohatsu-outboards.com/tohatsu-30-hp-4-stroke.htm>

3) Honda Marine: Owner’s Manual. 01 Dec. 2009. 08 Dec. 2009.
< http://www.honda-marine.com/owners/OwnerManuals/OM.aspx?modelGroup=BF30>

4) Centennial Initiative 2016. 30 Nov. 2009 18 Oct. 2007.

< http://www.nps.gov/grca/parkmgmt/ci2016.htm>

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< http://www.sswt.com/ready_flex_with_casing.htm>

6) Mayr: Torque-Limiting Clutches. 27 Sept. 2009. 08 Dec. 2009.

< http://www.mayr.de/en/products/torque-limiting-clutches/>

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http://www.epi-eng.com/propeller_technology/selecting_a_propeller.htm

8) Quality Transmission Components: Spiral Bevel Gears. 24 Nov. 2009. 08 Dec. 2009.

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Appendix A

CENTENNIAL INITIATIVE

Grand Canyon National Park
Glen Canyon National Recreation Area
www.nps.gov/grca
www.nps.gov/glca

Environmental Leadership

The following centennial proposal is certified as eligible for Centennial Challenge funding consideration in fiscal year 2008. Funding for the Centennial Challenge requires legislation.

Accomplish Alternative Motorboat Propulsion Research

Location: Arizona – Grand Canyon and Page
Partner: Grand Canyon River Outfitters Association, which is comprised of 17 river-running concessioners in Grand Canyon National Park and Glen Canyon National Recreation Area.
Partner Website: www.gcroa.org
Total Cost: $2.8 million
Proposal #137836

Summary: This collaborative effort will develop a new generation of cleaner and quieter boat motors by using alternative fuels, reducing the environmental impacts of boating. This project is a collaborative effort among the two parks' river concessioners and the National Park Service. The ultimate goal is to develop and implement a motorboat propulsion system that offers measurable environmental gains over the conventional four- stroke outboard motors currently used. The hope is to develop a system suitable for use not only in Grand and Glen Canyons, but also in other areas within the National Park System and elsewhere. By working together, the river concessioners hope to capitalize on economies of scale to achieve a greater level of accomplishment in less time.

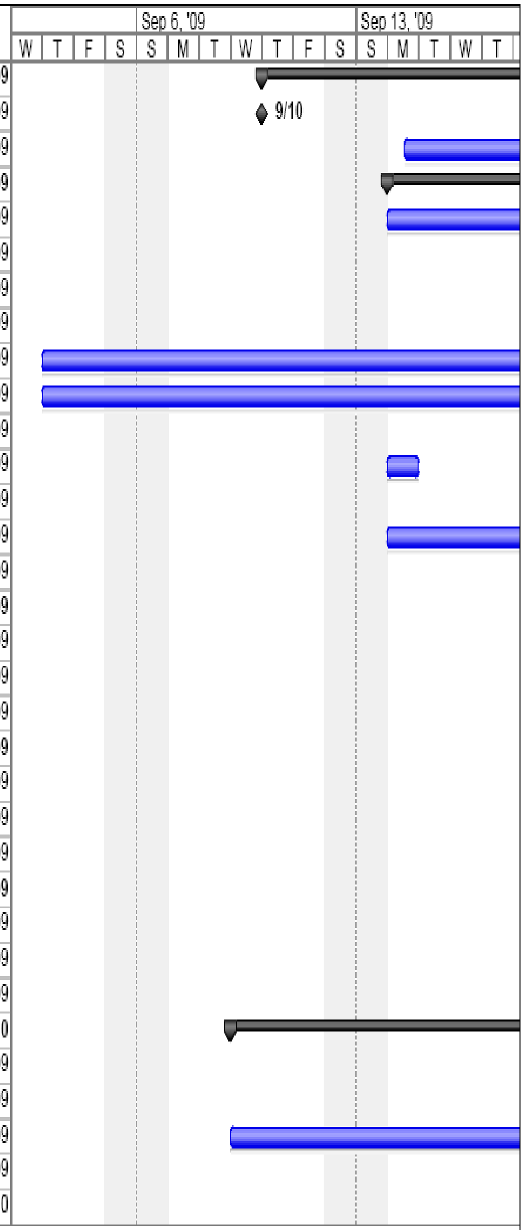
For more information contact:
Maureen Oltrogge, Grand Canyon National Park, (928) 638-7779. Maureen_Oltrogge@nps.gov

Lou Good, Glen Canyon National Recreation Area, 928-608- 6321. Lou_Good@nps.gov

Rafting the Colorado River
Grand Canyon National Park

Appendix B

ID		Task Name	Duration	Start	Finish
1		**Survey GCROA**	**12 days?**	**Thu 9/10/09**	**Fri 9/25/09**
2		E-mail Surveys	0 days	Thu 9/10/09	Thu 9/10/09
3		Collect Surveys	9.5 days?	Mon 9/14/09	Fri 9/25/09
4		**Customer Needs Assessment**	**14 days?**	**Mon 9/14/09**	**Wed 9/30/09**
5		Interpret Surveys	9 days?	Mon 9/14/09	Thu 9/24/09
6		Prioritize Needs List	1 day?	Mon 9/28/09	Mon 9/28/09
7		Finalize Constraints	3 days	Mon 9/28/09	Wed 9/30/09
8		Establish Preliminary Criteria	3 days?	Mon 9/28/09	Wed 9/30/09
9		Benchmarking and Research	71 days?	Thu 9/3/09	Wed 12/9/09
10		Develop Metrics and Specifications	24 days	Thu 9/3/09	Mon 10/5/09
11		Needs vs. Metrics Matrix	3 days	Mon 10/5/09	Wed 10/7/09
12		Function Decomposition	1 day?	Mon 9/14/09	Mon 9/14/09
13		QFD (Metrics vs. Functions Matrix)	4 days	Wed 10/7/09	Mon 10/12/09
14		Brainstorm Solutions for Functions	22 days?	Mon 9/14/09	Mon 10/12/09
15		Concept (Solution) Combinations Table	0 days	Wed 10/14/09	Wed 10/14/09
16		**Customer Meeting (GC Trip)**	**8 days?**	**Fri 9/25/09**	**Mon 10/5/09**
17		Meet with Customer	1 day?	Fri 9/25/09	Sat 9/26/09
18		Revise Design Process Details Based on Trip Info	3 days?	Mon 9/28/09	Wed 9/30/09
19		Present Criteria and Constraints	4 days?	Wed 9/30/09	Mon 10/5/09
20		**Establish 3 Alternatives**	**21 days?**	**Wed 10/7/09**	**Wed 11/4/09**
21		High Level Design	16 days?	Wed 10/7/09	Wed 10/28/09
22		Compare Alternatives with Criteria	1 day?	Wed 10/28/09	Wed 10/28/09
23		Present Design Concept	1 day?	Wed 11/4/09	Wed 11/4/09
24		**Detailed Design**	**30 days?**	**Wed 10/28/09**	**Tue 12/8/09**
25		Solidworks Modeling	30 days?	Wed 10/28/09	Tue 12/8/09
26		Engineering Analysis	30 days?	Wed 10/28/09	Tue 12/8/09
27		Manufacturing Plan	30 days	Wed 10/28/09	Tue 12/8/09
28		**Final Design**	**95 days?**	**Wed 9/9/09**	**Sat 1/16/10**
29		Create Presentation	11 days?	Mon 11/2/09	Mon 11/16/09
30		Create Poster	1 day?	Mon 12/14/09	Mon 12/14/09
31		Create Design Report	70 days?	Wed 9/9/09	Mon 12/14/09
32		Presentation to Faculty	1 day?	Mon 12/14/09	Mon 12/14/09
33		Presentation to GCROA	1 day?	Sat 1/16/10	Sat 1/16/10



Project: Preliminary RBLER Gantt
Date: Wed 11/11/09

Task | Milestone | External Tasks
Split | Summary | External Milestone
Progress | Project Summary | Deadline

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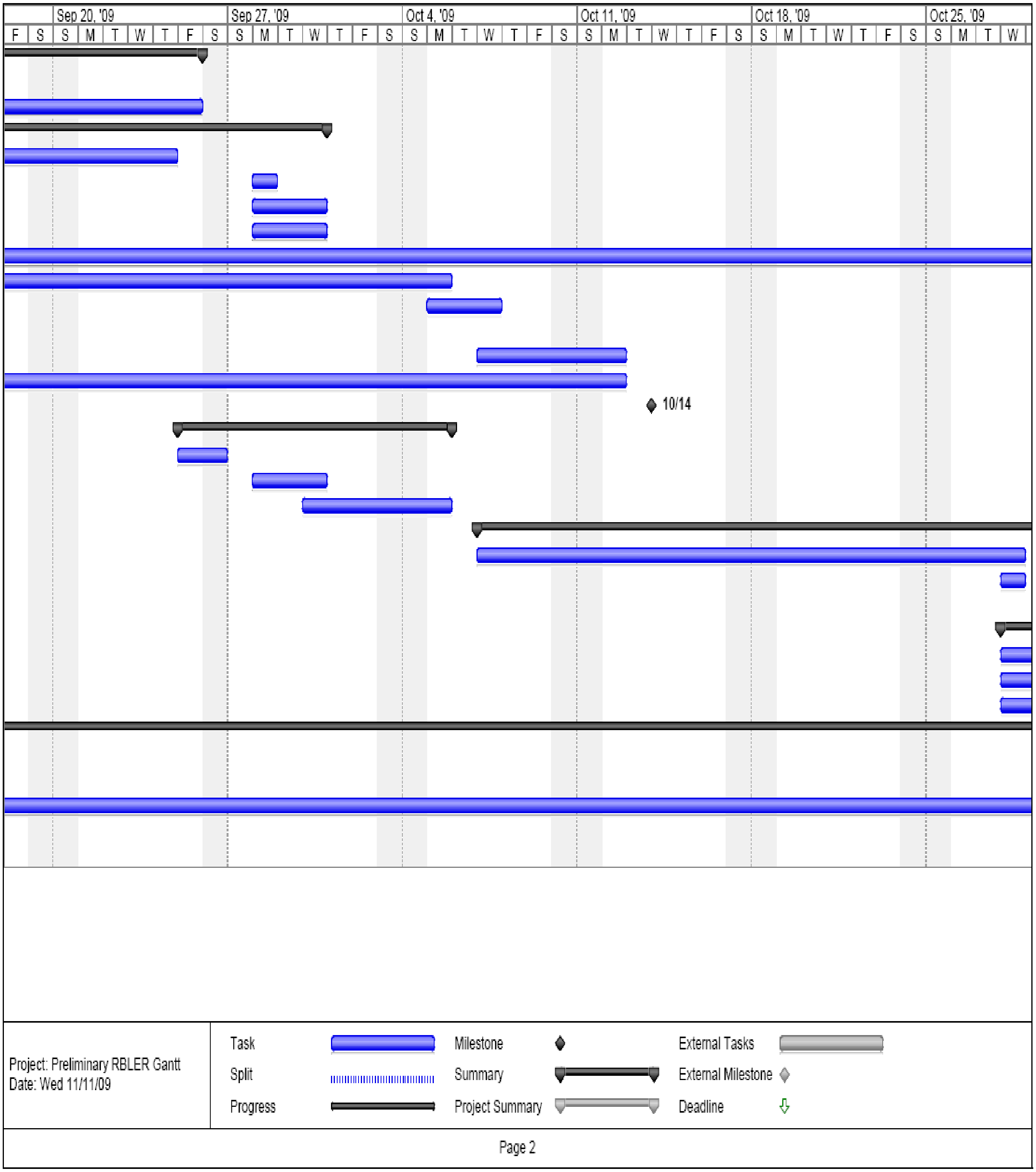

Sep 20, '09
Sep 27, '09
Oct 4, '09
Oct 11, '09
Oct 18, '09
Oct 25, '09
F S S M T W T F S S M T W T F S S M T W T F S S M T W T F S S M T W T F S S M T W
10/14
Project: Preliminary RBLER Gantt
Date: Wed 11/11/09
Task
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Summary
Project Summary
External Tasks
External Milestone
Deadline
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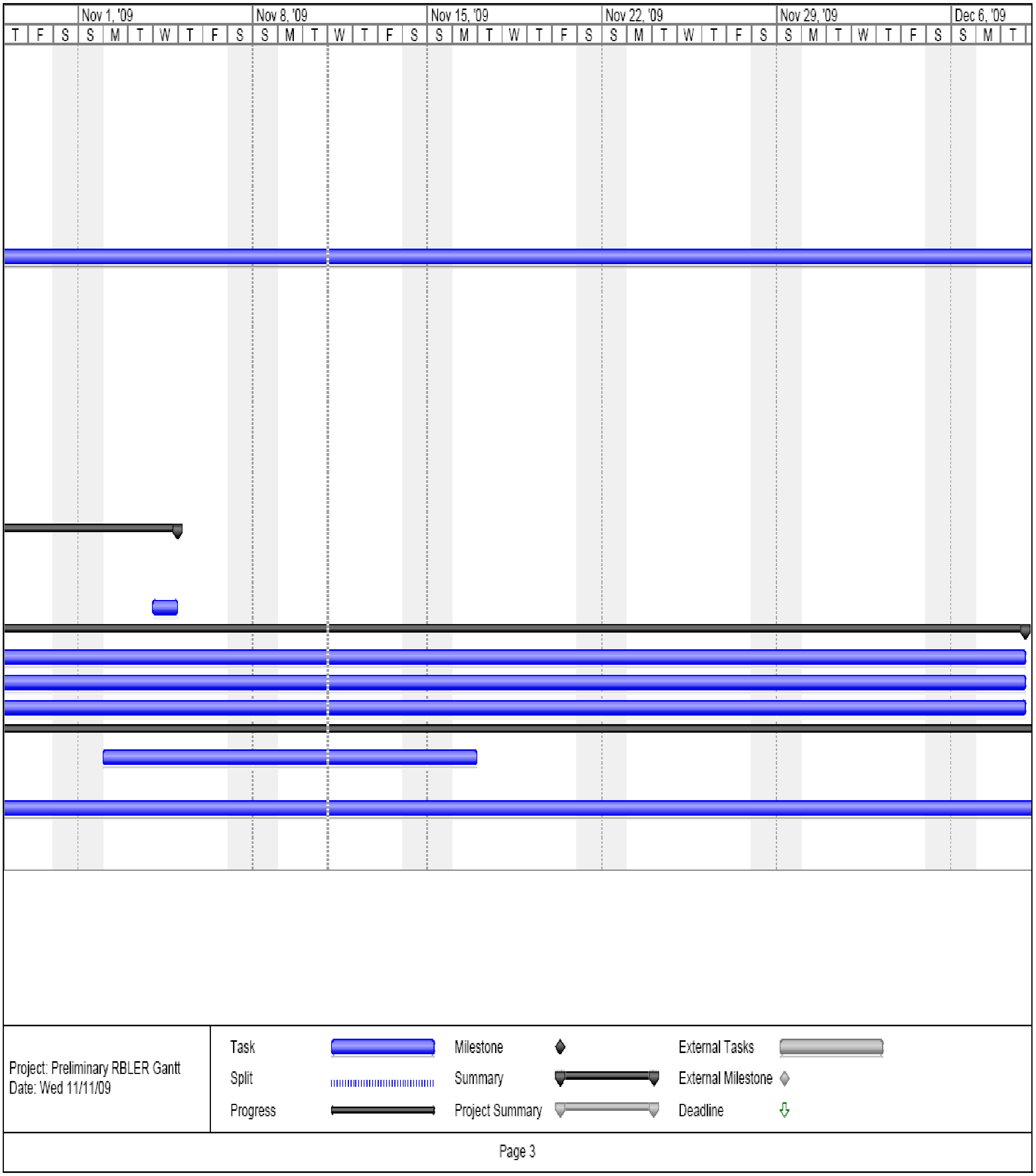

Nov 1, '09
Nov 8, '09
Nov 15, '09
Nov 22, '09
Nov 29, '09
Dec 6, '09
T F S S M T W T F S S M T W T F S S M T W T F S S M T W T F S S M T W T F S S M T
Project: Preliminary RBLER Gantt
Date: Wed 11/11/09
Task
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Summary
Project Summary
External Tasks
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Deadline
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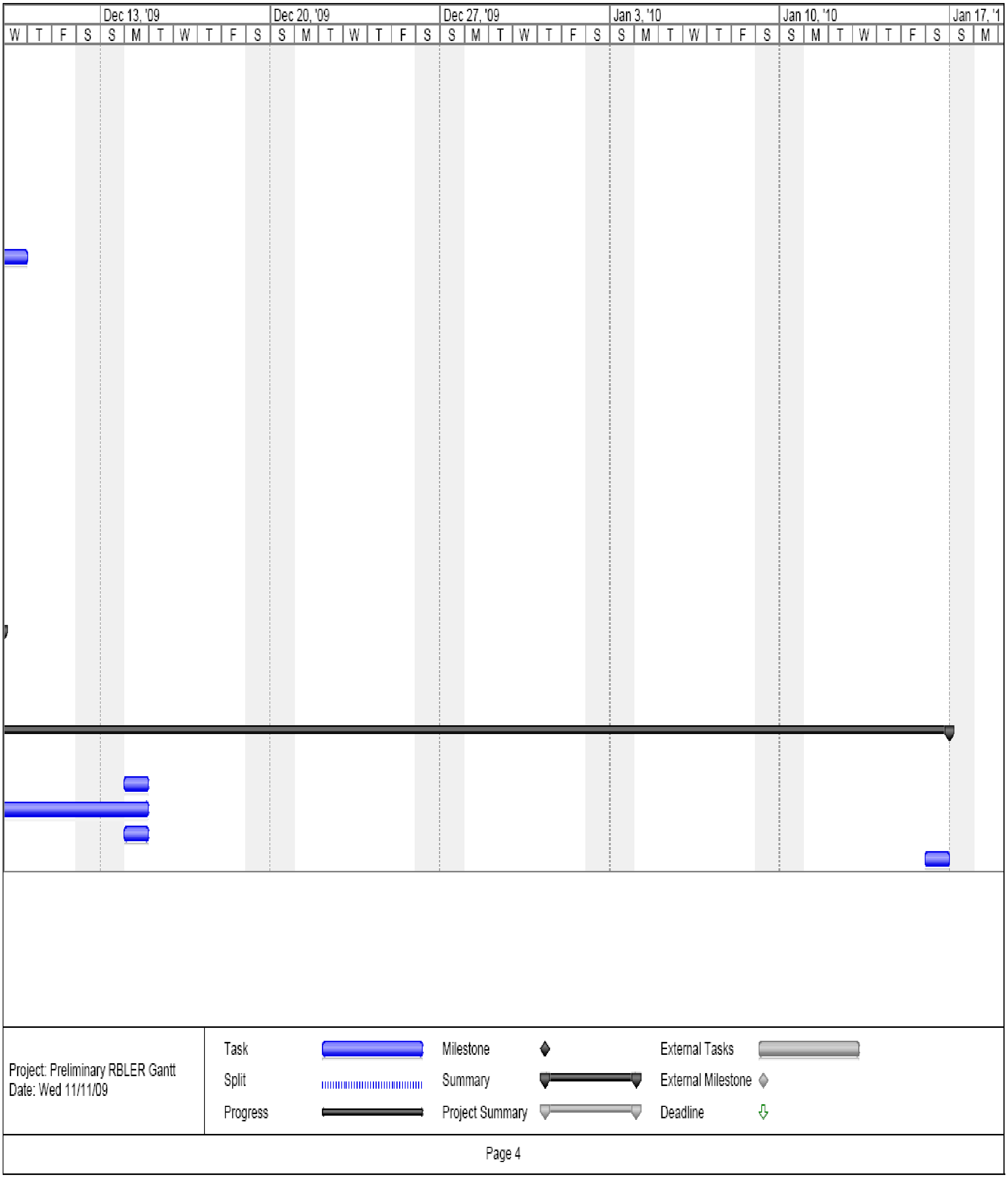
Dec 13, '09
Dec 20, '09
Dec 27, '09
Jan 3, '10
Jan 10, '10
Jan 17, '1
W T F S S M T W T F S S M T W T F S S M T W T F S S M T W T F S S M T W T F S S M
Project: Preliminary RBLER Gantt
Date: Wed 11/11/09
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Appendix C

Mount Power Source Solutions							
Adjust Motor Position		Motor Replacement		Connect Shaft		Orient Motor	
Solution Averages							
Quick Release Tab	10	Quick Release Tab	10	V-Belt/ Pulley	5.125	Horizontal	10
Hand Wheel	5.75	Hand Wheel	5.375	Groove Belt/ Pulley	6	Vertical	10
Jack Screw	3.5	Jack Screw	3.5	Chain	4	Angle	3.125
Bolt Down	9	Bolt Down	9	Timing Belt/ Pulley	6.5		
Set Screw	4.125	Set Screw	4.5	Gears	3.875		
Multiple Size Slots	6.5	Multiple Size Slots	6	Splines	9.75		
		Omni Mount	5.125	Direct	8.625		
				Coupling	9.625		
				Pin	7		
				Keyway	9.25		
				Weld	3.125		
				Viscous Clutch	2.75		
				Fluid Coupler	1.5		
				Plate Clutch	3.142857143		
Weighted Averages							
0.048034	0.48034	0.093	0.93	0.096	0.492	0.01	0.1
	0.2761955		0.499875		0.576		0.1
	0.168119		0.3255		0.384		0.03125
	0.432306		0.837		0.624		
	0.19814025		0.4185		0.372		
	0.312221		0.558		0.936		
			0.476625		0.828		
					0.924		
					0.672		
					0.888		
					0.3		
					0.264		
					0.144		
					0.301714286		

Cover Motor		Connect to Electric Power		Withstand OP Forces		Pivot	
Tarp	1	Cable Port	7.625	Material Selection/ Geometry	10	Vertical Hinge	10
Sheet metal	5.125	Quick Connect	9.375			Spherical Bearing	6.875
composite	9.375					Nozzle Control	4.375
Plastic	9.625						
0.0	0	0.04140063	0.315679804	0.06444356	0.6444356	0	0
	0		0.388130906				0
	0						0
	0						
	0						

Appendix C

Provide Steering Solutions						Transmit Power Solutions					
XY Dynamic Geometry		Human Control		Attitude Control Surfaces		Couple Motor/Thrust		Sustain Maximum Torque		Intake Cooling Water	
Rudder Shape	7	Hydraulic	4.125	Trim Tab	9.75	Flex Shaft	6.625	CVT	2.875	Scoop	5.625
Airfoil Shape	9.857142857	Tiller	10	Ballast	3.125	Rigid Shaft	10	Material Selection/ Geometry	10	Grate	9.125
Flat Surface	4.714285714	Cable Control	4.875	Thrust Tilt	5.75	Belt	6.625	NuVinci	3.25	Hole	7.625
Positive Chord Length	5.285714286	Remote Control	2.875			Chain	4.875	Torque Limiting Clutch	9.125		
		Electric Actuator	3.5			Gears	3.875				
		Steering Wheel	5.5			Hydraulic	2				
		Handlebars	5			Pneumatic	1.5				
						Link Arm	1.875				
0	0	0	0	0	0	0.06619	0.43850875	0.054	0.15525	0.026	0.14625
	0		0		0		0.6619		0.54		0.23725
	0		0		0		0.43850875		0.1755		0.19825
	0		0				0.32267625		0.49275		
			0				0.25648625				
			0				0.13238				
			0				0.099285				
							0.12410625				

Provide Cooling Solutions						Mount To Boat Solutions					
Transfer Cooling Water		Eject Cooling Water		Connect to Motor		Hand Mount to Jackass		Sustain OP Forces		Adjust Trim	
Vacuum Pump	6.25	Hole	9.375	Quick Connect	9.375	Thumbscrew	9.75	Material Selection/Geometry	10	Pin	9.7
Positive Displacement Pump	5	Hose	5.875	Hose Fitting	6.125	Quick Release	8.25			Hydraulic	2.7
Capillary Action	2.875					Magnetic	1.125			Ratcheting Gearset	5.62
Hand Pump	3.375					Latch	4.875			Detent Lever	7.37
Electric Pump	9.875					Bidirectional Screws	4.125			Radius Track	4.62
Shaft Driven Pump	7.75					Weld	1.125				
						Interchange Mount	4				
0.074	0.4625	0.027	0.253125	0.058	0.54375	0	0	0	0	0	
	0.37		0.158625		0.35525		0				
	0.21275						0				
	0.24975						0				
	0.73075						0				
	0.5735						0				
							0				

Appendix C

Encase Solutions									
Reduce Drag		Anti-Cavitation		Sustain OP Forces		Prevent Corrosion		Resist Water Intrusion	
Airfoil Geometry/ Coating/ Length	10	Anti-Ventilation Plate	10	Geometry	9.125	Anode,Material/ Coating	10	Seals/ Gasket	10
		Shroud	5.875	Shroud	5			Solid Casing	7
				Shield on Hull	5				
				Shield on Case	7.625				
				Spring Force Dampener	5				
				Break-away Detent	6.714285714				
				Torque Limit Clutch	9				
				Breakable Props	9.25				
0.018	0.18	0.05	0.5	0.101	0.921625	0.042	0.42	0.082	0.82
			0.29375		0.505				0.574
					0.505				
					0.770125				
					0.505				
					0.678142857				
					0.909				
					0.93425				

		Allow Tilt Solutions				Provide Thrust Solutions	
Allow for other Functions		Adjust for Storage/Trim		Provide Manual Adjustment		Translate rotational motion to linear motion	
Geometry	10	Pin	9.875	Handle	9.875	Propeller	10
		Hydraulic	3	Screw System	5	Paddle Wheel	5.625
		Ratcheting Gearset	5.125	Cable	4.75	Impeller/ Jet	8.25
		Detent Lever	8			Radial Vacuum	2.5
		Radius Track	5.875			Fin	1.375
						Surface Piercing Prop	7
						Star Wars Prop	1.625
0	0	0	0	0	0	0.04963732	0.4963732
			0		0		0.279209925
			0		0		0.40950789
			0				0.1240933
			0				0.068251315
							0.34746124
							0.080660645

Solution Averages and Weighted Solutions Scores

Appendix D

						Mount Power Source Solutions	
Adjust Motor Position		Motor Replacement		Connect Shaft		Orient Motor	
Reduced Solutions							
Quick Release Tab	0.48034	Quick Release Tab	0.93	Splines	0.936	Vert/horiz	0.1
Bolt Down	0.432306	Bolt Down	0.837	Coupling	0.924		
				Keyway	0.888		

Cover Motor		Connect to Electric Power		Withstand OP Forces		Pivot	
0	0	Quick Connect	0.388130906	Material Selection/ Geometry	0.6444356	0	0

Provide Cooling Solutions						Mount To Boat Solutions			
Transfer Cooling Water		Eject Cooling Water		Connect to Motor		Hand Mount to Jackass		Sustain OP Forces	
Electric Pump	0.73075	Hole	0.253125	Quick Connect	0.54375	0	0	0	0

		Encase Solutions							
Adjust Trim		Reduce Drag		Anti-Cavitation		Sustain OP Forces		Prevent Corrosion	
0	0	Airfoil Geometry/ Coating/ Length	0.18	Anti-Ventilation Plate	0.5	All in geometry	0.921625	Anode,Material/ Coating	0.42

Provide Steering Solutions						Transmit Power Solutions					
XY Dynamic Geometry		Human Control		Attitude Control Surfaces		Couple Motor/Thrust		Sustain Maximum Torque		Intake Cooling Water	
0	0	0	0	0	0	Flex Shaft	0.43850875	Material Selection/ Geometry	0.54	Grate	0.23725
						Rigid Shaft	0.6619	Torque Limiting Clutch	0.49275		
						Belt	0.43850875				

				Allow Tilt Solutions				Provide Thrust Solutions	
Resist Water Intrusion		Allow for other Functions		Adjust for Storage/Trim		Provide Manual Adjustment		Translate rotational motion to linear motio	
Seals/ Gasket	0.82	0	0	0	0	0	0	Propeller	0.49637

Reduced Solutions (In Green) and Their Scores (In Blue)

Appendix E

Concept Combinations			
1	9.560288456	37	9.512254456
2	9.513038456	38	9.465004456
3	9.783679706	39	9.735645706
4	9.736429706	40	9.688395706
5	9.560288456	41	9.512254456
6	9.513038456	42	9.465004456
7	9.548288456	43	9.500254456
8	9.501038456	44	9.453004456
9	9.771679706	45	9.723645706
10	9.724429706	46	9.676395706
11	9.548288456	47	9.500254456
12	9.501038456	48	9.453004456
13	9.512288456	49	9.371254456
14	9.465038456	50	9.417004456
15	9.735679706	51	9.687645706
16	9.688429706	52	9.640395706
17	9.512288456	53	9.464254456
18	9.465038456	54	9.417004456
19	9.467288456	55	9.419254456
20	9.420038456	56	9.372004456
21	9.690679706	57	9.642645706
22	9.643429706	58	9.595395706
23	9.467288456	59	9.419254456
24	9.420038456	60	9.372004456
25	9.455288456	61	9.407254456
26	9.408038456	62	9.360004456
27	9.678679706	63	9.630645706
28	9.631429706	64	9.583395706
29	9.455288456	65	9.407254456
30	9.408038456	66	9.360004456
31	9.419288456	67	9.371254456
32	9.372038456	68	9.324004456
33	9.642679706	69	9.594645706
34	9.595429706	70	9.547395706
35	9.419288456	71	9.371254456
36	9.372038456	72	9.324004456

The 72 Possible Concept Combinations and Their Scores

Appendix F

Material Designation for Drive Shafts on Lower End.									
Material Grade	Benefits to Lower End	ultimate strength	yield strength	Hardness	Density	Mod. Elas	Thermal Exp.	Cost	Suppliers
303 Stainless	high machinability	500	190	262	8.03	193	17.3	$1.10	Speedy Metals
austenitic	good corrosion resistance	Mpa	Mpa	Brinell B	g/cm3	Gpa	(µm/m/°C)	per inch	www.Speedymetals.com
	good strength/toughness			max				3/4" Round X 12'	
	reasonable cost							call	Industrial Metal Supply (IMS)
304	most versatile stainless	515	205	201	8000	193	17.2	$0.68	metals depot
austenitic	high availability	Mpa	Mpa	Brinell B	Kg/m3	Gpa	(µm/m/°C)	per inch	www.metalsdepot.com
	very weldable							3/4" Round X 12'	
	high forming								
	high machinability							call for quote	IMS
	high hardenability							local, no shippin cost	
316	high toughness	515	205	217	8000	193	16.3	$1.37	Speedy Metals
austenitic	very weldable above 6mm	Mpa	Mpa	Brinell B	Kg/m3	Gpa	(µm/m/°C)	per inch	www.speedymetals.com
	alternative alloys available							3/4" round	
	solution hardening							call for quote	
	already used in marine ap.							1.50lbs/ft	IMS
								$1.17	IMS quote
								per inch	
410	high hardenability	1240	960	325	7750	200	9.9	must get quote	IMS
martensitic	machinable	Mpa	Mpa	Brinell B	Kg/m3	Gpa	(µm/m/°C)	call or special order	Speedy
316C	fresh water applications								
	shaft applications								
	used in ballistics and blades								
420	poor weldability	655	345	241	7750	200	10.3	call for quote/online	yarde metals online
martensitic	lowered corrosion resistance	Mpa	Mpa	Brinell B	Kg/m3	Gpa	(µm/m/°C)		420 F RD
annealed condition	lowered ductility								
	high hardenability								
316C		1580	1365	444	same	same	same		IMS
		mpa	mpa	Brinell B					
440		758	448	269	7650	200	10.1	$1.38	IMS
martensitic		Mpa	Mpa	Brinell B	Kg/m3	Gpa	(µm/m/°C)	per inch	quote by phone
annealed								3/4" round X12'	440C grade
316C		1860	1740	56	same	same	same		
				Rockwell C					
440C	**great in strength**	**297**	**280**	**60**	**2.760E-01**	**29007**	**same**	**$1.38**	**IMS**
hardened at 150C OQ	**low impact resistance**	**kpsi**	**kpsi**	**rockwell C**	**lbs/in3**	**kpsi**		**per Inch**	
in English units				**9**				**3/4" Round X 12**	
				charpy					

Appendix F

Casing material selection and criteria for lower End Design				
Material	**pros**	**cons**	**yield (ksi)**	**ultimate (ksi)**
5086-H112	weldable, high corrosion resistance	non heat-treating	19	39
	Utilized in boat hulls	cracking & chipping		
5083-H112	readily wealdable	non heat-treating	27	43
	Utilized in boat hulls	cracking & chipping		
6061-T6	versatile use	limited enhancement	40	45
	readily available			
	workable, high strength			
	weldable, high corrosion resistance			
	relatively affordable			
Alustar ™	marine alloy, for hull design	expensive	39	53
	superior weld strength/ corr resistance	special order		
	high repair/weld properties	overkill		
	outperforms 5xxx ser. Corr. Resistance			
	high strength, high impact resistance			
FERRALIUM® alloy 255	superior mechanical strength	very expesive	98	126
	erosion, pitting, cracking resistance	heavy/ overkill		
	highest strength of group	special order		
		ferritic/ treatments		

Appendix G

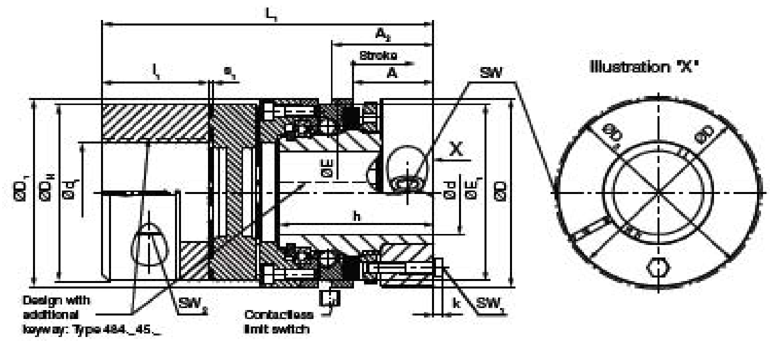
mayr®

EAS®-smartic® synchronous clutch Type 484._ _5._

EAS®-smartic® lastic backlash-free

Type 484._35._
Clamping hub both sides

Type 484._45._
Clamping hub and keyway both sides



Sizes 01 to 2
ROBA®-ES-side: clamping hub, EAS®-smartic®-side: clamping hub

Dimensions	Size 01	Size 0	Size 1	Size 2
A	29	29	34	38
A_1	14	15	17	19
Ø D	55	70	85	100
Ø D_1	57	70	85	105
Ø D_2	50	65	78	91
Ø D_H	55	65	80	105
Ø E	55	70	85	100
Ø E_1	50	65	80	95
G	M5	M6	M8	M8
h	51	56	65	75
h_1	36	42	48	56
k	2,8	2,8	3,5	4
k_1	1,5	2,8	3,5	3,5
L_1	107	118	142	170
L_2	92	104	125	151
l_1	30	35	45	56
s_1	2	2,5	3	3,5
SW	6	6	8	10
SW_1	7	7	8	10
SW_2	5	6	6	10
SW_3	5	7	8	8
t	10	15	15	25

1) Up to ø 19 keyway acc. DIN 6885/1, over ø 19 keyway acc. DIN 6885/3
2) Up to ø 27 keyway acc. DIN 6885/1, over ø 27 keyway acc. DIN 6885/3
3) Up to ø 38 keyway acc. DIN 6885/1, over ø 38 keyway acc. DIN 6885/3

Bores				Size 01	Size 0	Size 1	Size 2
EAS®-smartic® side	Type 484._25._	Ø d_2^{H7}	min.	10	14	19	20
			max.	22 5)	30 2)	38 3)	45 4)
	Type 484._35._	Ø d^{H7}	min.	10	14	19	20
			max.	22	32	42	50
	Type 484._45._	Ø d^{H7}	min.	10	14	19	20
			max.	20 5)	30 2)	38 3)	45 4)
ROBA®-ES side	Type 484._25._	Ø d_2^{H7}	min.	8	10	13	20
			max.	28	38	45	60
	Type 484._$\frac{3}{4}$5._	Ø d_1^{F7}	min. 6)	15	19	20	35
			max. 6)	28	35	45	55

We reserve the right to make dimensional and constructional alterations.

Accessory parts (hook wrench for torque adjustment)		
Size	Article number hook wrench Type 484._25._	Article number hook wrench Types 484._35._ / 484._45._
01	8170662	8170663
0	4084939	4084158
1	4084939	4084158
2	4084940	4084159

4) Up to ø 43 keyway acc. DIN 6885/1, over ø 43 keyway acc. DIN 6885/3
5) Up to ø 17 keyway acc. DIN 6885/1, over ø 17 keyway acc. DIN 6885/3
6) The transmittable torques on the flexible coupling "T_{KN}" are dependent on factors such as e.g. temperature factor, torsional rigidity factor etc., (please contact the manufacturers). Furthermore, the transmittable torques are dependent on the bore diameters d or d_1 (see Table below: Preferred bores and respective transmittable torques).

Preferred bores and respective transmittable torques [Nm] on diameters d and d_1 of frictional locking on hubs for shaft tolerance k_6 ROBA®-ES-side and h_6 / h_8 EAS®-smartic®-side

Size	Ø 10 Ø d	Ø 11 Ø d	Ø 12 Ø d	Ø 14 Ø d	Ø 15 Ø d	Ø 15 Ø d_1	Ø 20 Ø d	Ø 20 Ø d_1	Ø 25 Ø d	Ø 25 Ø d_1	Ø 28 Ø d	Ø 28 Ø d_1	Ø 32 Ø d	Ø 32 Ø d_1	Ø 35 Ø d	Ø 35 Ø d_1	Ø 42 Ø d	Ø 42 Ø d_1	Ø 45 Ø d	Ø 45 Ø d_1	Ø 50 Ø d	Ø 50 Ø d_1	Ø 55 Ø d_1
	Torque ranges 2 up to 7 (Types 484.235._, 484.335._, 484.435._, 484.535._, 484.635._ and 484.735._)																						
01	23	27	30	37	40	34	53	54	-	57	-	63	-	-	-	-	-	-	-	-	-	-	-
0	-	-	-	42	45	-	64	83	80	104	90	116	102	133	-	145	-	-	-	-	-	-	-
1	-	-	-	-	-	-	88	83	110	104	124	116	142	133	155	145	186	174	-	187	-	-	-
2	-	-	-	-	-	-	140	-	175	-	210	-	240	-	266	350	320	455	343	505	381	600	705
	Torque range 8 (Type 484.835._)																						
01	37	43	48	59	64	54	85	86	-	91	-	101	-	-	-	-	-	-	-	-	-	-	-
0	-	-	-	67	72	-	102	133	128	166	144	186	163	213	-	232	-	-	-	-	-	-	-
1	-	-	-	-	-	-	141	133	176	166	198	186	227	213	248	232	298	278	-	299	-	-	-
2	-	-	-	-	-	-	224	-	280	-	336	-	384	-	426	408	512	531	549	589	610	700	823

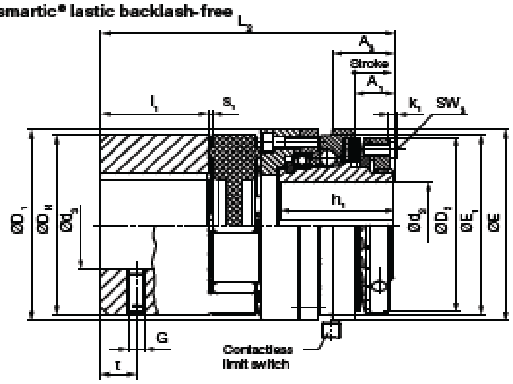
6

EAS®-smartic® synchronous clutch Type 484._ _5._

mayr®

EAS®-smartic® lastic backlash-free

Type 484._25._
key hub both sides



Sizes 01 to 2
ROBA®-ES-side: key hub, EAS®-smartic®-side: key hub

Technical Data					Size 01	Size 0	Size 1	Size 2
Limit torques for overload	Type 484.2_5._	(Torque range 2)	M_G	[Nm]	2,7 - 5	5 - 10	10 - 20	20 - 40
	Type 484.3_5._	(Torque range 3)	M_G	[Nm]	5 - 10	10 - 20	20 - 40	40 - 80
	Type 484.4_5._	(Torque range 4)	M_G	[Nm]	8 - 15	15 - 30	30 - 60	60 - 120
	Type 484.5_5._	(Torque range 5)	M_G	[Nm]	11 - 20	20 - 40	40 - 80	80 - 160
	Type 484.6_5._	(Torque range 6)	M_G	[Nm]	18 - 33	35 - 65	70 - 125	140 - 250
	Type 484.7_5._	(Torque range 7)	M_G	[Nm]	32 - 40	60 - 80	120 - 160	240 - 320
	Type 481.8_5.0 [8]	(Torque range 8)	M_G	[Nm]	35 - 60	70 - 120	150 - 240	300 - 500
Nominal and maximum torques,[7] flexible backlash-free shaft coupling	92 Shore A		T_{KN}	[Nm]	35	95	190	310
			$T_{K\,max}$	[Nm]	70	190	380	620
	98 Shore A		T_{KN}	[Nm]	60	160	325	525
			$T_{K\,max}$	[Nm]	120	320	650	1050
Maximum speed			n_{max}	[rpm]	3000	3000	2500	2000
Thrust washer stroke on overload				[mm]	0,9	1,1	1,3	1,5
Tightening torques, clamping screws	SW		T_A	[Nm]	40	40	83	140
	SW_2	Torque ranges 2 up to 7	T_A	[Nm]	10	25	25	120
		Torque range 8	T_A	[Nm]	17	40	40	140
Permitted misalignments, flexible backlash-free shaft coupling	Axial displacement	92/98 Shore A	ΔK_a	[mm]	1,4	1,5	1,8	2,1
	Radial misalignment	92 Shore A	ΔK_r	[mm]	0,14	0,15	0,17	0,21
		98 Shore A	ΔK_r	[mm]	0,1	0,11	0,12	0,16
	Angular misalignment	92 Shore A	ΔK_w	[°]	1,0	1,0	1,0	1,0
		98 Shore A	ΔK_w	[°]	0,9	0,9	0,9	0,9
Mass moments of inertia [7]	Type 484._25._	EAS®-smartic® hub-side	J	[kgm²]	0,00010	0,00034	0,00086	0,00200
		ROBA®-ES-side	J	[kgm²]	0,00028	0,00056	0,00149	0,00773
	Type 484._35._	EAS®-smartic® hub-side	J	[kgm²]	0,00017	0,00056	0,00151	0,00320
		ROBA®-ES-side, Torque ranges 2 up to 7	J	[kgm²]	0,00024	0,00058	0,00140	0,00490
		ROBA®-ES-side, Torque range 8	J	[kgm²]	0,00038	0,00088	0,00228	0,00490
Weights [7]	Type 484._25._			[kg]	0,78	1,31	2,27	5,89
	Type 484._35._	Torque ranges 2 up to 7		[kg]	1,01	1,62	2,75	6,72
		Torque range 8		[kg]	1,29	2,06	3,59	6,72

7) The mass moments of inertia and weights refer to clutches with maximum bore. 8) Maximum speed: 250 rpm

Order Number

__ / 4 8 4 . __ __ 5 . __ / __ / __ / __ / __ / __

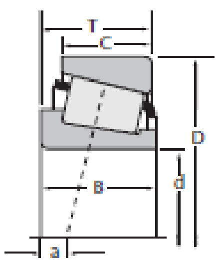
Size	Torque range		92/98 Shore A	Bore	Bore	Bore	Bore	With limit switch
01	Torque range 2	2 Keyway	92 Shore A 3	Ø d H7	Ø d_1 F7	Ø d_2 H7	Ø d_3 H7	see page 10
0	Torque range 3	3 Clamping hub	98 Shore A 4					
1	Torque range 4	4 Clamping hub + keyway						
2	Torque range 5							
	Torque range 6							
	Torque range 7							
	Torque range 8							

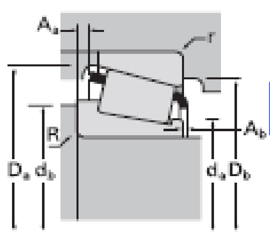
Example: 1 / 484.535.4 / 35 / 35 / plus limit switch 055.002.5

7

TS

SINGLE-ROW





Dimensions, mm (inches)			Load Ratings, N (lbf.)							Part Number	
			Dynamic [1]	Factors [5]		Dynamic [2]	Factors [5]		Static		
d	D	T	C_1	e	Y	C_{90}	C_{a90}	K	C_0	Inner	Outer
7.938 0.3125	31.991 1.2595	10.008 0.3940	10600 2380	0.41	1.48	2750 618	1910 429	1.44	9230 2070	A2031	A2126
9.525 0.3750	31.991 1.2595	10.008 0.3940	10600 2380	0.41	1.48	2750 618	1910 429	1.44	9230 2070	A2037	A2126
11.112 0.4375	34.988 1.3775	10.998 0.4330	12200 2740	0.45	1.32	3160 710	2450 550	1.29	11500 2580	A4044	A4138
11.987 0.4719	31.991 1.2595	10.008 0.3940	10600 2380	0.41	1.48	2750 618	1910 429	1.44	9230 2070	A2047	A2126
12.680 0.4992	34.988 1.3775	10.998 0.4330	12200 2740	0.45	1.32	3160 710	2450 550	1.29	11500 2580	A4049	A4138
12.700 0.5000	34.988 1.3775	10.998 0.4330	12200 2740	0.45	1.32	3160 710	2450 550	1.29	11500 2580	A4050	A4138
12.700 0.5000	38.100 1.5000	13.495 0.5313	19300 4340	0.28	2.18	5010 1130	2360 531	2.12	17100 3840	00050	00150
14.987 0.5901	34.988 1.3775	10.998 0.4330	12200 2740	0.45	1.32	3160 710	2450 550	1.29	11500 2580	A4059	A4138
15.875 0.6250	34.988 1.3775	10.998 0.4330	14100 3160	0.32	1.88	3650 820	1990 447	1.83	13900 3130	L21549	L21511
15.875 0.6250	39.992 1.5745	12.014 0.4730	12400 2790	0.53	1.14	3220 724	2900 653	1.11	12300 2770	A6062	A6157
15.875 0.6250	41.275 1.6250	14.288 0.5625	22200 5000	0.31	1.93	5770 1300	3070 690	1.88	21300 4790	03062	03162
15.875 0.6250	42.862 1.6875	14.288 0.5625	17400 3910	0.70	0.85	4510 1010	5430 1220	0.83	17400 3920	11590	11520
15.875 0.6250	42.862 1.6875	16.670 0.6563	29100 6540	0.33	1.81	7550 1700	4280 962	1.76	29200 6560	17580	17520
15.875 0.6250	47.000 1.8504	14.381 0.5662	24700 5560	0.36	1.68	6420 1440	3920 881	1.64	25400 5720	05062	05185
15.875 0.6250	49.225 1.9380	19.845 0.7813	39700 8920	0.27	2.26	10300 2310	4680 1050	2.20	40500 9100	09062	09195
15.875 0.6250	49.225 1.9380	23.020 0.9063	39700 8920	0.27	2.26	10300 2310	4680 1050	2.20	40500 9100	09062	09194
15.875 0.6250	53.975 2.1250	22.225 0.8750	43000 9670	0.59	1.02	11200 2510	11300 2540	0.99	42500 9560	21063	21212
15.987 0.6294	46.975 1.8494	21.000 0.8268	37100 8350	0.55	1.10	9630 2170	9000 2020	1.07	39300 8840	HM81649	HM81610
16.993 0.6690	39.982 1.5741	12.014 0.4730	12400 2790	0.53	1.14	3220 724	2900 653	1.11	12300 2770	A6067	A6157A
16.993 0.6690	39.992 1.5745	12.014 0.4730	12400 2790	0.53	1.14	3220 724	2900 653	1.11	12300 2770	A6067	A6157
16.993 0.6690	47.000 1.8504	14.381 0.5662	24700 5560	0.36	1.68	6420 1440	3920 881	1.64	25400 5720	05066	05185
17.455 0.6872	36.525 1.4380	11.112 0.4375	12100 2720	0.49	1.23	3130 704	2610 587	1.20	11600 2600	A5069	A5144
17.462 0.6875	39.878 1.5700	13.843 0.5450	22900 5160	0.29	2.10	5950 1340	2910 655	2.04	23400 5260	LM11749	LM11710
17.462 0.6875	44.450 1.7500	12.700 0.5000	19900 4460	0.48	1.25	5150 1160	4220 950	1.22	20600 4640	4C	6
17.462 0.6875	44.450 1.7500	15.494 0.6100	24700 5560	0.36	1.68	6420 1440	3920 881	1.64	25400 5720	05068	05175
17.987 0.7082	47.000 1.8504	14.381 0.5662	24700 5560	0.36	1.68	6420 1440	3920 881	1.64	25400 5720	05070XS	05185-S
18.000 0.7087	47.000 1.8504	14.381 0.5662	24700 5560	0.36	1.68	6420 1440	3920 881	1.64	25400 5720	05070X	05185-S
19.004 0.7482	56.896 2.2400	19.368 0.7625	42000 9450	0.31	1.95	10900 2450	5740 1290	1.90	45300 10200	1774	1729

[1] Based on 1 x 10^6 revolutions L_{10} life, for the ISO life calculation method.
[2] Based on 90 x 10^6 revolutions L_{10} life, for The Timken Company life calculation method. C_{90} and C_{a90} are radial and thrust values.
[3] Negative value indicates effective center inside cone backface.
[4] These maximum fillet radii will be cleared by the bearing corners.
[5] These factors apply for both inch and metric calculations. Consult your Timken representative for instruction on use.