Final Report

ME 450: Design and Manufacturing III

Winter 2007

Bicycle Regenerative Braking: Redesign for Manufacturability

Team 15

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1. ABSTRACT
The Environmental Protection Agency (EPA) has worked on hydraulic hybrid vehicle technology for cars and trucks, but they are currently interested in having the University of Michigan explore alternative applications, such as bicycles. During the fall of 2006, a student team built a hydraulic regenerative braking system (RBS). However, their prototype was too heavy and large to fit in the front wheel of a standard bicycle. The objective of this project is to redesign the RBS for manufacturability as a true retrofit front wheel on any bicycle. This involves reducing system weight and size, and improving efficiency. Our project will demonstrate the feasibility of this technology to interested bicycle manufacturers.

2. INTRODUCTION
The Environmental Protection Agency (EPA), an agency of the U.S. government, was established by President Nixon in 1970 to enforce federal pollution reduction laws and to implement various pollution prevention programs. Today, it is undeniable that our environment has become more hazardous than ever. The U.S. is the world leader in pollution. According to the 2006 progress report of the EPA's Clean Automotive Technology Program [1], transportation is responsible for 30% of national CO_2 emissions in the US. In an effort to decrease the amount of emissions in high-traffic areas (e.g. cities), it is noted that cycling would make a more convenient, cleaner, and in some cases, faster alternative to driving a car or taking a bus. Since cycling requires more effort than driving, it is believed that regenerative braking would make cycling a more feasible traveling option.

Since the fall of 2004, the EPA has collaborated with student teams from the University of Michigan in implementing RBS in bicycles. Earlier teams have worked on the design and made gradual improvements toward the goal of manufacturability. By the fall of 2006, the teams had collectively designed and built prototypes that were either working but large or compact but non-working. Our EPA sponsor/customer, Mr. David Swain, has assigned our team the task of reducing the weight and size of the latest prototype and ensuring that it functions properly so as to bring it a step closer to manufacturability. Future teams would then manufacture the outer casing (hub), incorporate the RBS in the bicycle front wheel, and improve the overall ergonomics of the system.

The RBS works by strategically transferring an "incompressible fluid" in a closed hydraulic system. During braking, a hydraulic pump thrusts fluid from a low pressure accumulator to a high pressure accumulator. Using bevel gears to connect the bicycle wheel to the pump shaft, the increasing difficulty in rotating the pump shaft causes the fluid to become pressurized and the bicycle to decelerate. During the launch phase, the fluid flow is reversed and actuates a hydraulic motor. This time, using bevel gears to connect the motor shaft to the bicycle wheel, the bicycle accelerates. Since the hydraulic system has to be compact, solenoid-operated 3-way valves are used to recycle the fluid in different paths during braking and launching.

3. INFORMATION SEARCH
This section presents the results of our literature and patent search [2], and relates them to the technical benchmarks to be accomplished. This section concludes with a brief description of future information sources for our project.

This project started in the Fall 2004 semester, and it has since been passed onto five successive semesters of ME 450 teams. As such, much of the present information is based on previous reports and documentation. Technical understanding of the project is mostly provided by Mr. Swain, and a student from a previous team, Jason Moore. These experienced sources supplied us with the majority of our initial information. The Internet was used to locate concurrent research and patents that are closely related to our project.

3.1. Patent Search

One patented idea involves the same concept as the RBS system in the bicycle using hydraulics. However, the hydraulic system is coupled with a traditional combustion engine and it utilizes heavy machinery. In contrast, our RBS uses electronics to operate its hydraulic components, and it is very compact. Therefore, the concept and intention of the system is similar, but the method is quite different [3].

There is another patented apparatus that combines one electronic braking unit that is positioned at each rear wheel of a vehicle and one hydraulic braking unit that is positioned at each front wheel. Similar to our RBS, energy is stored by compressing fluid during braking. However, the stored energy is used in decelerating the rear wheels instead of accelerating them [4].

There also exists, in some vehicles, a practical alternative to fully regenerative systems involving variable displacement pumps (similar to the ones used in our RBS). A fixed displacement pump, a motor, and a hydro-pneumatic accumulator system are used, but this system has a maximum efficiency of 45% [5].

According to the report by the Fall 2006 team, a electro-hydraulic/air bike design was patented in 1990 [6]. In this design, the working fluid (either hydraulic fluid or air) can be pressurized by either braking or pedaling. The system is known to be functional, but the design overwhelms the frame of the bicycle as seen in Fig. 1. The largest part of this system is a complex gear train comprising of screw gears, worm gears, and spur gears [7]. In contrast, the design of our RBS would fit inside the front wheel.

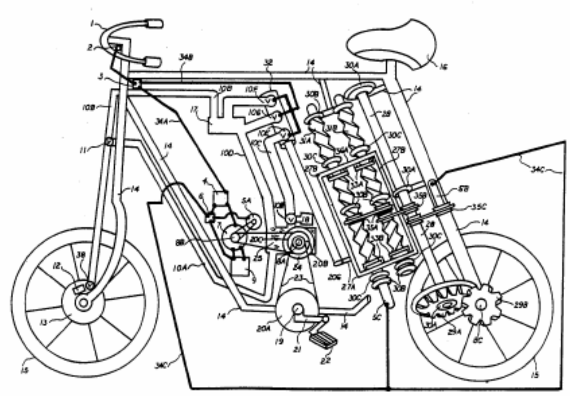


Figure 1: The Electro-hydraulic/air Bike is a complex hybrid bicycle that employs hydraulic and electric power transfer

3.2. Technical Benchmarks

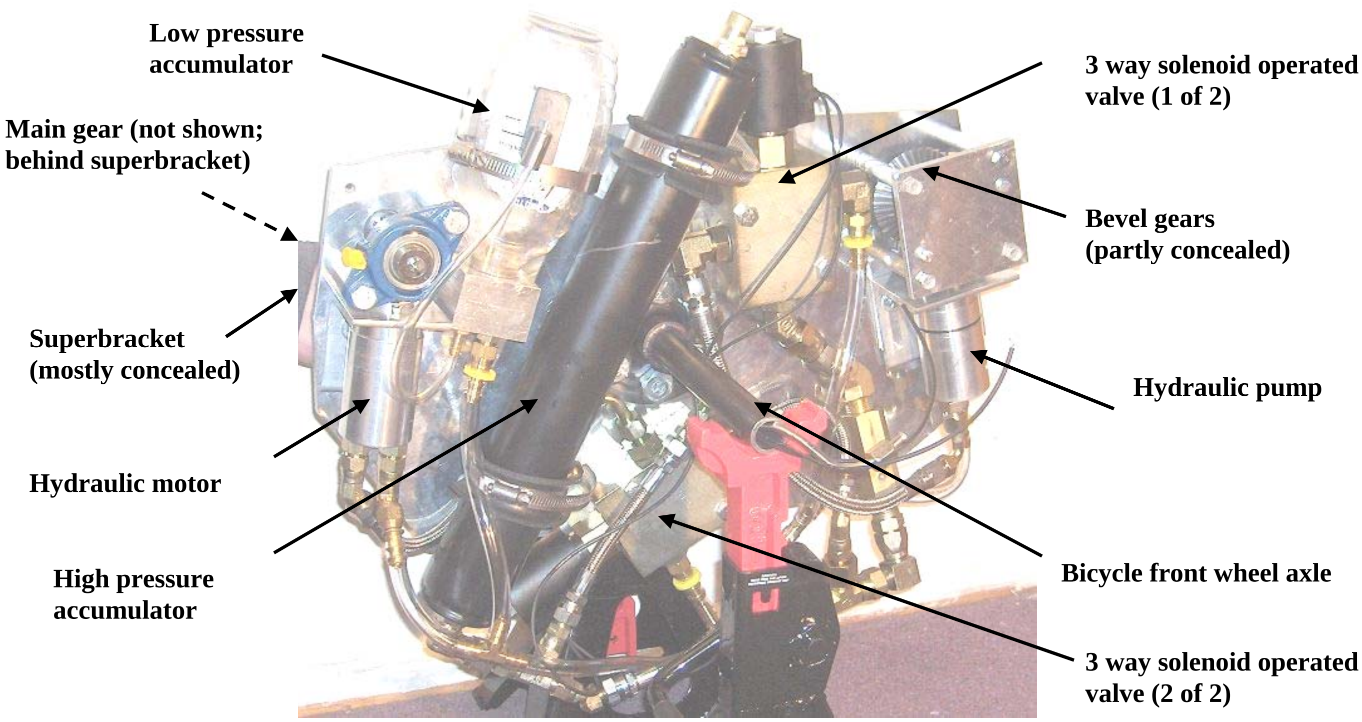
Most of these patents rely on hydraulics, which too, will be used in our bicycle. However, our project is unique in that the RBS is located within the front wheel of the bicycle. Our design requires the hydraulic and mechanical components to be compact and lightweight. This unprecedented design could encourage widespread implementation by bicycle manufacturers. Details on the requirements of the hydraulic system are described in the engineering specifications section of this report.

3.3. Future Information Sources

Future sources of information will include fluid mechanics texts, product catalogues of hydraulic components, component distributors, and experts such as Dr. Katsuo Kurabayashi (project advisor), Mr. Swain, and a student familiar with the RBS design, Jason Moore. Regarding manufacturing issues, we could consult Bob Coury, Marv Cressey, and Steve Emanuel (machine shop staff).

4. CUSTOMER REQUIREMENTS AND ENGINEERING SPECIFICATIONS

The requirements and specifications were provided by Mr. Swain and by the work of previous teams. Since we are improving the latest prototype, several references to RBS components will be made in this section. A brief description of the main components in the hydraulic system is outlined below to familiarize the reader with the terminology used in the basic design. Figure 2 shows the latest prototype built by the team of Fall 2006, and the location of the components. Table 1 on p. 6 outlines the function of each component.



Source: ME 450 Fall 2006 Team 5 report

Figure 2: Latest (Fall 2006) prototype of the hydraulic system

Component	Function
Low pressure accumulator	Receives fluid during launching.
High pressure accumulator	Receives fluid during braking.
Hydraulic pump	Moves fluid from low to high pressure accumulator; in the process, the increasing difficulty slows the bicycle by way of bevel gear connections to main gear.
Hydraulic motor	Actuates when fluid is passed through from high pressure to low pressure accumulator; in the process, accelerates the bicycle via bevel gear connections to main gear.
3 way solenoid operated valve	Recycles fluid in the appropriate paths during braking or launching.
Bevel gears	Translate motion of the pump/motor to the bicycle wheel via main gear.
Superbracket	Mounting plate for the RBS. This is rigidly attached to the axle on the front wheel of the bicycle.
Main gear	Turns the bicycle wheel using the motion of bevel gears.

Table 1: List of component function

4.1. Engineering Specifications

From our discussion with Mr. Swain and the information from previous teams, we have obtained the engineering specifications shown in Table 2 below. The bicycle is assumed to operate at a top speed of 20 mph and to decelerate without flipping over the front wheel at a maximum braking torque of 130 N-m. The maximum launch torque is set at 90 N-m to create a comfortable acceleration. With these limitations, the maximum working pressure of the system is 5000 psi. The RBS also needs to fit inside a standard 29” bicycle wheel, and it will be enclosed in a hub that is attached to the rim. The width of the RBS is to be less than 4”, and its weight should be close to 22 lbf. For practical reasons, the maximum fluid volume is set to 1 L and the displacement for each hydraulic motor or pump is 1.5 c.c. When the entire system is integrated, the rider should not take more than an hour to learn how to operate the regenerative braking and launch assist system. These engineering specifications were made known to us from the start from Fall 2006 team’s analysis. In summary, our engineering objective is to manufacture a compact hydraulic system that meets the torque, weight, and size specifications.

Engineering Specifications	Target Value
Wheel width and diameter	4” and 29”, respectively
Maximum braking torque	130 N-m
Top operating speed of bicycle	20 mph
Approximate efficiency	> 70%
Maximum launch torque	90 N-m
Maximum system working pressure	5000 psi
Total weight of hydraulic system	< 22 lbf
Motor/pump displacement	1.5 c.c.
Maximum volume of fluid	1 L
Learning curve	~ 1 hour

Table 2: Engineering specifications

4.2. Customer Requirements

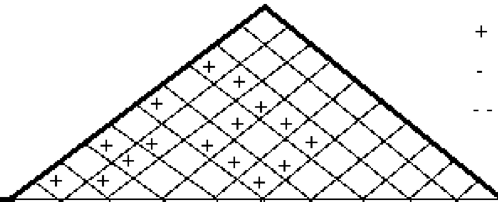
Since this project is a continuation of work from previous teams under the same customer, the customer requirements are already well established. The customer required the product to be effective (considerations are “adequate top speed”, “efficiency”, and “reliable”) and safe (considerations are “natural rate of braking”, and “safety”), with the ultimate goal of encouraging widespread adoption by bicycle manufacturers (considerations are “universal application”, “low weight”, “aesthetics”, “easy to use/service”, and “maintains bicycle functions”). Efficiency is evaluated as the fluid pressure loss in the piping system, whereby relatively low pressure loss is considered “efficient”.

In earlier teams, their emphasis was on designing the concept and refining it such that suitable engineering parameters were obtained. Our team was tasked with improving the design and creating a working prototype. Thus, the priority was to build a lightweight working hydraulic system so that future (and final) work would focus on incorporating our RBS design into the final product.

We have modified the latest Quality Function Development (QFD) from the Fall 2006 team to reflect the present challenges for our current RBS design (Fig. 3 on p. 8). The QFD was submitted to Mr. Swain for his approval and corrections, and he has indicated that the categories of highest priority are “lightweight” and “universal applications”. The weighted values expressed in percentage of their respective sums dictate the direction of our RBS design. They reflect the compromise between the technical and customer specifications in the relationship matrix. In the QFD, the weighted values that make up the larger proportions of the specifications are shaded red. The highlighted areas indicate that making the system lightweight is the top priority from the technical standpoint, while from the customer’s perspective, “safety”, “lightweight”, “efficiency”, and “universal application” are the most desired characteristics. Additional comments by Mr. Swain are referenced in the QFD with superscripts.

Relationships between technical requirements

++ Strong Positive
+ Medium Positive
- Medium Negative
- - Strong Negative



Customer requirements \ Technical specifications	Weightage	Hub width [5]	Hub diameter	Max. sustained braking torque	Max. Weight << 22lbs	Max. sustained launching torque	Motor/pump displacement	Max. pressure [4]	Max. volume of hydraulic fluid [2]	Hydraulic fluid filtration [1]	Constrained to forward movement [3]	**Total (weighted customer requirement)**	**Normalized to total**	**Importance rating (% of total)**	**Fall 06 Bike**	**Winter 05 Bike**	**Fall 05 Bike**
Universal application	9	9	9	1	9	1	9	1	7	1	1	432	0.16	16	2	1	1
Natural rate of braking	7	1	1	9	1	1	7	7	5	1	1	238	0.09	9	2	2	3
Sufficient top speed	5	1	1	1	7	9	5	9	5	1	1	200	0.07	7	2	2	3
Efficient	7	1	5	3	7	7	3	8	3	7	6	350	0.13	13	2	2	3
Lightweight	9	3	3	1	9	1	5	3	7	1	1	306	0.11	11	2	1	1
Reliable	7	1	1	5	3	5	3	1	1	9	5	238	0.09	9	2	1	3
Aesthetics	5	7	7	1	5	1	1	1	5	1	1	150	0.05	5	2	2	4
Safety	7	1	1	9	7	9	3	5	3	5	7	350	0.13	13	2	1	2
Easy to use	5	1	1	5	9	5	3	1	1	1	7	170	0.06	6	1	1	3
Easy to service	5	5	5	1	1	1	1	5	1	9	1	150	0.05	5	2	2	1
Maintains bicycle function	5	7	7	3	7	3	1	1	1	1	9	200	0.07	7	3	3	3
Measurement Unit		in	in	N-m	lb	N-m	cc.	kPsi	L	n/a	Y/N						
Present Standard Bicycle		4	29	100	20	0	n/a	n/a	n/a	n/a	Y						
Moped		5	16	n/a	100	2	n/a	n/a	n/a	n/a	Y						
Our target		4	29	130	<<22	90	1.5	5	1	n/a	Y						
Total (weighted technical specifications)		241	269	255	433	267	293	268	275	237	246						
Normalized to total		0.09	0.10	0.09	0.16	0.10	0.11	0.10	0.10	0.09	0.09						
Importance Rating (% of total)		9	10	9	16	10	11	10	10	9	9						

Weightage used throughout

1 - Not related/important
3 - Weakly related/important
5 - Somewhat related/important
7 - Moderately related/important
9 - Strongly related/important

[1] External hand-pump and in-line filter will solve this issue
[2] Need at least 6600Joules (4000joules/(60% efficiency) for 100kg to achieve 20mph)
[3] Requires electronic control and/or checkvalve
[4] Higher Pressure results in better efficiency
[5] Hub Width internals should be 3.5", completed hub with shell should be 4"

Figure 3: QFD diagram

5. CONCEPT GENERATION

The RBS has 6 major functions – mounting the hydraulic components, channeling the hydraulic fluid, directing the fluid, transmitting the torque during braking and launching, storing the energy, and capturing the energy from cycling. The storage function is limited to the use of a piston accumulator as it is more compact and robust compared to the alternative option of using a bladder accumulator; thus, we have brainstormed different concepts for the other 5 functions. In addition, we will re-use the hydraulic pump/motor (Parker HY-09 series pump motors) from the Fall 2006 prototype since information on their torque and speed specifications are well established and have already met the relevant engineering specifications. Thus, we would consider just the transmission mechanisms in our concept generation for the energy related functions. For easy reference, all the concepts are arranged in rows according to function in a morphological chart, and are shown in Table 3 on p. 10. A brief self-explanatory description is provided with each sketch.

The concepts are constrained by the requirement that the product can be manufactured or assembled using commercially available materials, therefore our concepts are conventional. In addition to the conceptual form for each major function, the choice of raw material used in manufacturing the main gear and the superbracket, and the integration of the components is also important. These will be described in a later section.

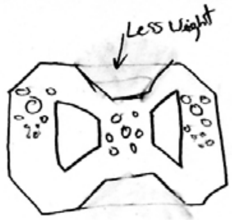
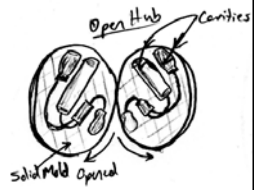
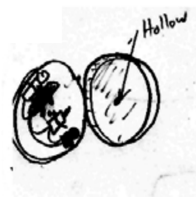
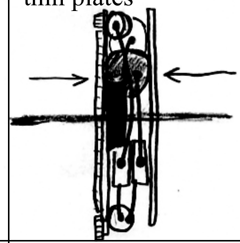
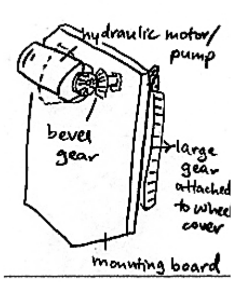
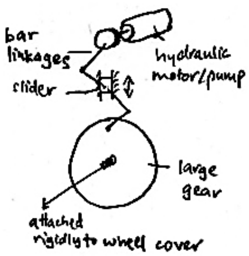
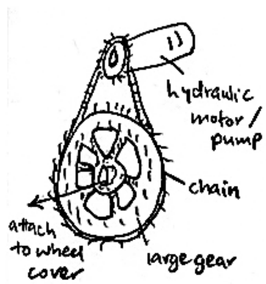
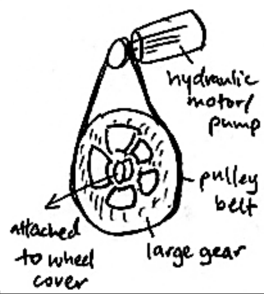
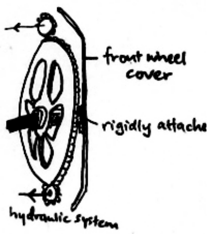
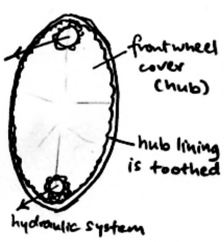
Function	Concept 1	Concept 2	Concept 3	Concept 4	Concept 5
Mount components	Metal Superbracket 	Molded plastic hub with cavities 	Attach to hub cover 	Mount directly to axle 	Sandwiched by thin plates 
Channel hydraulic fluid	Flexible plastic/ reinforced tubes 	Fixed metal pipes 			
Direct hydraulic fluid	Manually operated valves 	Solenoid operated valves 	Pressure sensing valves 		
Transmit torque	Bevel gears 	Bar linkages/cams 	Chains 	Pulley and belt 	
Capture energy	Large gear 	Toothed hub 			

Table 3: Morphological chart

6. CONCEPT EVALUATION AND SELECTION

The concepts for each function were evaluated using Pugh charts, wherein the customer requirements and weight are obtained from the QFD diagram established earlier. The first concepts are arbitrarily assigned as the datum, against which the other concepts are evaluated. "S" indicates that a value is rated the same as the datum on each Pugh chart. For easy reference, a brief explanation is presented next to each score that is different from the datum. It is important to note that “aesthetics”, “easy to use”, and “maintains bicycle function” are not influenced by any concept since the RBS will be concealed by a wheel cover (that is designed in future work) and the user will not directly interact with the internal workings of the RBS and thus not sense differences in operating the bicycle.

6.1. Mounting Components

Table 4 shows the Pugh chart for mounting components. Based on the score, the best concept is the superbracket.

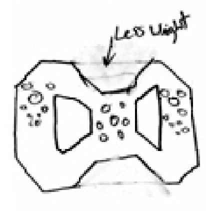
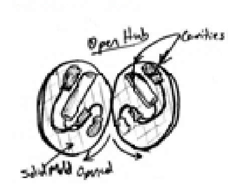
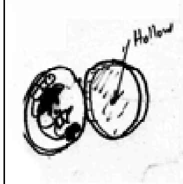
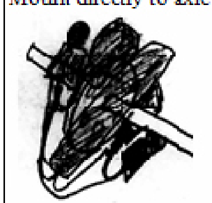
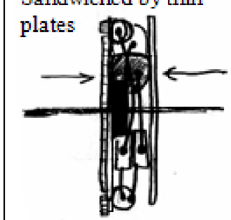
		Sketch 1 (datum)	**Sketch 2**	**Sketch 3**	**Sketch 4**	**Sketch 5**
Customer requirement	**Weight**	Metal Superbracket 	Molded plastic hub with cavities 	Attach to hub cover 	Mount directly to axle 	Sandwiched by thin plates 
Universal application	9	S	- (requires complex mold)	S	S	S
Natural rate of braking	7	S	S	S	S	S
Sufficient top speed	5	S	S	S	S	S
Efficient	7	S	- (sharp turns in piping)	S	- (sharp turns in piping)	- (sharp turns in piping)
Lightweight	9	S	- (unused space taken by mold)	+ (directly mounted to wheel cover)	+ (no mounting board)	S
Reliable	7	S	S	-	- (unstable)	S
Aesthetics	5	S	S	S	S	S
Safety	7	S	S	-	- (very close proximity of components)	+ (components are shielded on either side)
Easy to use	5	S	S	S	S	S
Easy to service	5	S	- (components are rigidly interconnected)	S	- (components are bunched in a mess)	- (components are hard to access)
Maintains bicycle function	5	S	S	S	S	S
	Total +	0	0	+9	+9	+7
	Total -	0	-30	-14	-26	-12
	Total	**0 (best)**	-30	-5	-17	-5

Table 4: Pugh chart for mounting components

6.2. Channeling Hydraulic Fluid

Table 5 shows the Pugh chart for channeling hydraulic fluid. Based on the score, the better concept is the flexible plastic/reinforced tubes.

		Sketch 1 (datum)	Sketch 2
Customer requirement	**Weight**	Flexible plastic/reinforced tubes 	Fixed metal pipes 
Universal application	9	S	- (requires complicated pipe bending)
Natural rate of braking	7	S	S
Sufficient top speed	5	S	S
Efficient	7	S	- (many sharp bends)
Lightweight	9	S	- (denser material)
Reliable	7	S	- (much pressure losses)
Aesthetics	5	S	S
Safety	7	S	+ (unlikely to burst)
Easy to use	5	S	S
Easy to service	5	S	- (require entire pipe section takeout in repair)
Maintains bicycle function	5	S	S
	Total +	0	+7
	Total -	0	-37
	Total	**0 (best)**	-30

Table 5: Pugh chart for channeling hydraulic fluid

6.3. Directing Hydraulic Fluid

Table 6 on p. 13 shows the Pugh chart for directing hydraulic fluid. All the concepts are awarded the same degree of universal application since they are all commercially available. Based on the score, the best concept is the solenoid operated valves.

		Sketch 1 (datum)	**Sketch 2**	**Sketch 3**
Customer requirement	**Weight**	Manually operated valves 	Solenoid operated valves 	Pressure sensing valves 
Universal application	9	S	S	S
Natural rate of braking	7	S	S	- (paths influenced by flow pressure and not by user)
Sufficient top speed	5	S	S	S
Efficient	7	S	S	S
Lightweight	9	S	+ (light electric wiring used for control)	+ (no additional components for controlling flow)
Reliable	7	S	S	- (might operate in unintended way)
Aesthetics	5	S	S	S
Safety	7	S	S	- (might operate in unintended way)
Easy to use	5	S	+ (user operates electric switch)	S
Easy to service	5	S	S	S
Maintains bicycle function	5	S	S	S
	Total +	0	+14	+9
	Total -	0	0	-21
	Total	0	**+14 (best)**	-12

Table 6: Pugh chart for directing hydraulic fluid

6.4. Transmitting Torques

Table 7 on p. 14 shows the Pugh chart for transmitting torques between the bicycle wheel and the hydraulic pump or motor. Based on the score, the best concept is the bevel gear system.

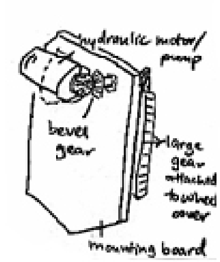
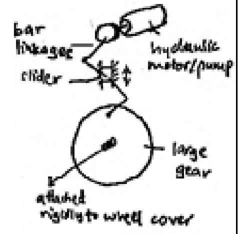
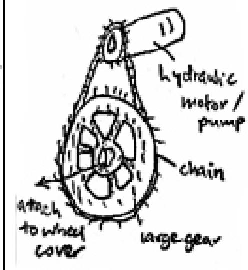
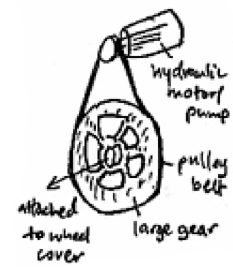
		Sketch 1 (datum)	Sketch 2	Sketch 3	Sketch 4
Customer requirement	**Weight**	Bevel gears  	Bar linkages/cams  	Chains  	Pulley and belt 
Universal application	9	S	- (require specialized bar and slider system)	S	S
Natural rate of braking	7	S	- (possible jerky motion)	- (possible backlash)	- (slippage is likely at decelerations)
Sufficient top speed	5	S	- (linkages may not catch up at high RPM)	- (inertia moving chain)	- (not suitable for high RPM)
Efficient	7	S	- (much heat losses)	- (frictional heat losses from large chain interface)	- (frictional heat losses from large belt interface)
Lightweight	9	S	S	S	- (usually lighter than gears)
Reliable	7	S	- (jerky motion)	S	- (belt may slip at high RPM)
Aesthetics	5	S	S	S	S
Safety	7	S	- (high RPM unsuitable for linkage)	S	- (belt may slip at high RPM or creep with time)
Easy to use	5	S	S	S	S
Easy to service	5	S	- (require entire linkage system takeout in servicing)	S	S
Maintains bicycle function	5	S	S	S	S
	Total +	0	0	0	0
	Total -	0	-47	-19	-42
	Total	**0 (best)**	-47	-19	-42

Table 7: Pugh chart for transmitting torques

6.5. Capturing Energy

Table 8 shows the Pugh chart for capturing energy. Based on the score, the better concept is the toothed hub, where the inner lining of the hub (front wheel cover) contains gear teeth that are directly connected to the mechanical motion from the hydraulic system. This idea was provided by our sponsor as an additional consideration above the present scope of creating the RBS. Due to the time and budget constraints (expensive custom plastic molding will be involved), we use the concept of the large gear.

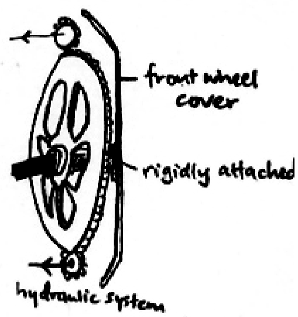
		Sketch 1 (datum)	**Sketch 2**
Customer requirement	**Weight**	Large gear 	Toothed hub 
Universal application	9	S	S
Natural rate of braking	7	S	S
Sufficient top speed	5	S	S
Efficient	7	S	S
Lightweight	9	S	+ (gears in hydraulic system is integrated directly with the hub)
Reliable	7	S	S
Aesthetics	5	S	S
Safety	7	S	S
Easy to use	5	S	S
Easy to service	5	S	S
Maintains bicycle function	5	S	S
	Total +	0	+9
	Total -	0	0
	Total	0	**+9 (better, but not in project scope)**

Table 8: Pugh chart for capturing energy

7. SELECTED CONCEPT

The selected concept of the RBS integrates the best concept for each function. Figure 4 shows the layout of the RBS on the superbracket (the main gear is located behind the superbracket and is not shown). The relatively heavy high pressure accumulator is located at the base of the superbracket to keep the center of gravity low to reduce instability during cycling. The "spread-out" design keeps the components close to the superbracket to ensure that the width of the prototype is as close as possible to the design requirement of 4". In addition, it improves efficiency by reducing pressure losses from complex piping. The shaded pipes represent the high pressure reinforced tubes while the unshaded pipes represent the low pressure vinyl tubes. Each outlet on the 3-way valve is labeled 1, 2, or 3. When the valve is not electrically activated, the valve connects 2 and 3; when the valve is activated, it connects 1 and 3. The user would activate valve A during braking, causing the pump to thrust the fluid from the low pressure accumulator to the high pressure accumulator through a check-valve to prevent back-flow. Valve B is activated during launching, causing the fluid to flow from the high pressure accumulator to the low pressure accumulator through the motor which then actuates the main gear. Only one valve can be activated at any time using a simple electric circuit that incorporates a user-controlled throw switch (not shown in Fig.). Information on the major components can be referred to in Table 1 on p. 6. A pressure gauge is connected to the side of the line to the high pressure accumulator to measure pressure changes in the RBS.

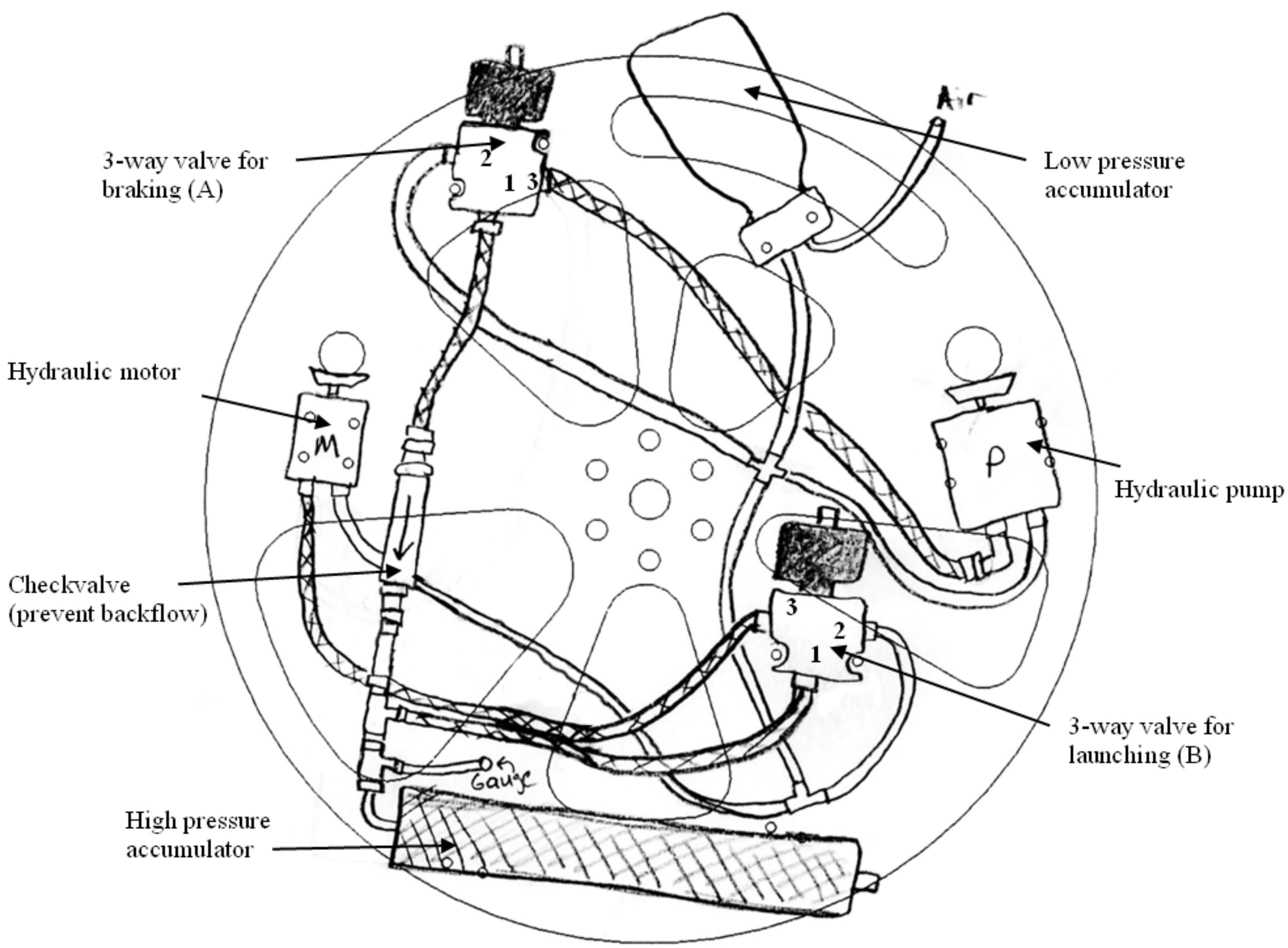


Figure 4: Layout of RBS on superbracket

Since our prototype intends to encourage mass production by bicycle manufacturers, our selected concept should include considerations for design for manufacturing and assembly (DFMA). Table 9 shows the DFMA considerations in our selected concept.

DFMA guidelines	Location	Implementation
Minimal part count	Hydraulic circuit	Keep layout simple with relatively few complicated bends in the piping
Ease of assembly in open spaces		
Standardized parts to reduce part variety		Use standard SAE / JIC fittings
Facilitate orientation for assembly	Superbracket	Large irregular holes are non-symmetrical about the central axis
Use standard stocks for ease of machining		The shafts that pass through the bracket are standard 7000 series cylindrical stock with standard diameters
Standard dimensions for ease of machining		Holes can be fabricated with standard tools
Avoid long narrow holes		All holes are simple through holes that pass through the thin bracket
Avoid drilling inclined surfaces		

Table 9: DFMA considerations in prototype

8. ENGINEERING ANALYSIS

The RBS essentially comprises the superbracket, the gearing system, and the hydraulic system. Engineering analysis for each component will be described in this section.

8.1. Superbracket

From our discussion with Mr. Swain, AZ91 magnesium alloy was considered as the raw material for the superbracket, since the density of magnesium is around 2/3 that of aluminum and 1/4 that of iron [8]. In addition, magnesium alloys generally have better recyclability compared to polymers or other metallic materials [8]. Furthermore, the popularity of their use as a high-strength and light weight structural material has increased in recent years in the automobile industry. However, several machinists on campus and off-campus were reluctant to custom manufacture the superbracket from magnesium alloy due to fire risk. We managed to contact a willing company, but decided not to use this material after learning the high cost involved in coating the alloy after manufacturing to reduce its inherent susceptibility to corrosion. From our discussion with this company, we decided to use composite G10 as the choice material. This material is cheap, lightweight and easily machinable.

In considering the form of the superbracket, initial estimations of the design strength were made. We considered the hydraulic system mounted such that the heavier components are close to the wheel center. Since the width of the system was specified to be less than 4” and is small relative to the length of the bracket (around 26”), we assume that the overall center of gravity of the hydraulic system is located near the wheel center and has negligible loading effect on the superbracket. The torques produced by the hydraulic motor and pump are considered instead.

Consider for example the torque produced by the bevel gear attached directly to the shaft of the motor, Tm = 7.57 N-m. The superbracket is modeled as a cantilever beam of length L shown in Fig. 5. To balance the torque, the edge of the bracket experiences a force F, giving a deflection u expressed by eqn. (1), where E is the elastic modulus of the bracket and I is the moment of inertia of the cross section of the bracket. The maximum stress due to bending is obtained using the earlier equation, and compared to the material yield stress to check for failure. The remaining torques have to be considered in turn if such a model is used.

$$u = \frac{F \cdot L^3}{3 \cdot E \cdot I} \tag{1}$$

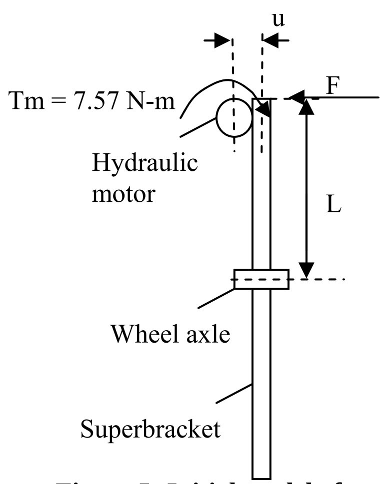


Figure 5: Initial model of superbracket

However, this model is inadequate since the superbracket will have several through holes to reduce its weight. The non-uniform cross section would be too complex to analyze by hand calculations. Thus, we would evaluate the superbracket design using FEA software. Figure 6 shows a sketch of the superbracket along with the loading conditions. The forces from the bevel gears are determined using the manufacturer's catalogue, and their directions are opposite that of separating forces in bevel gear sets [9]. The bicycle axle would run through the center large circular hole, and the mounting will be secured by an axle plate bolted to the superbracket using 6 bolts around the axle. The remaining circular holes are used to mount the hydraulic components, while the large cut-outs are used for weight reduction. Details on the dimensions are provided in Appendix A.

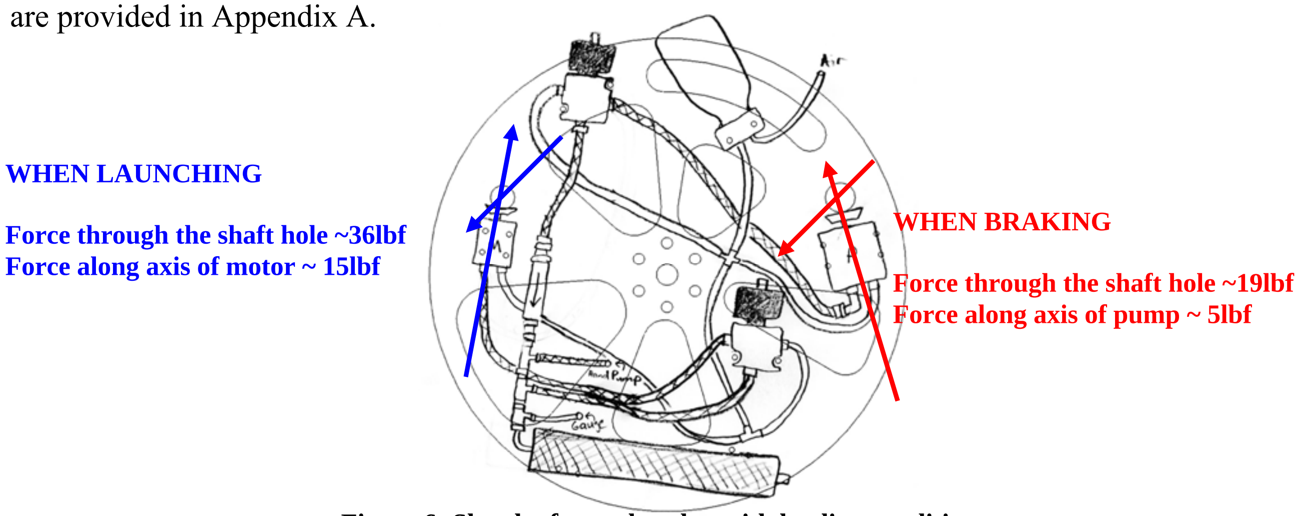
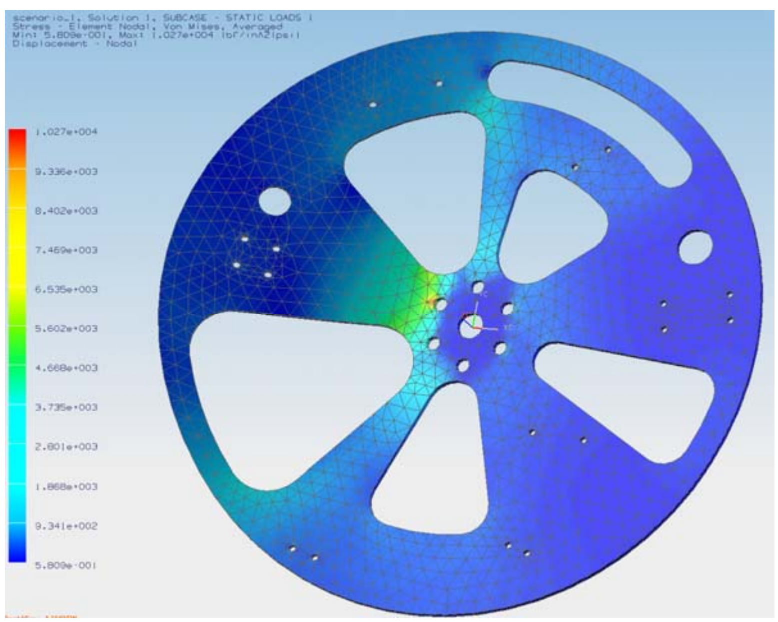


Figure 6: Sketch of superbracket with loading conditions

G10 material properties provided by the manufacturer are shown in Table 10. These properties were assigned to the superbracket in the FEA and the part was meshed with thousands of nodes as sites for stress calculations. The face of the axle hole and the surrounding 6 holes are fixed from translation and rotation as a realistic operating condition. Since either braking or launching occurs at any instance, each "side" of the superbracket was analyzed separately. Figures 7 and 8 on p. 20 show the FEA of the superbracket under loading caused by the torques from the hydraulic motor and pump respectively. The regions of the greatest stresses (shown in yellow-red) were found to be below that of the yield strength of G10. From Fig. 7, the maximum stress is around 10000 psi and from Fig. 8 on p. 20, the maximum stress is around 9600 psi. Thus, the superbracket has a safety factor of at least 5 because either of the maximum stresses is less than a fifth of the rated flexural strength.

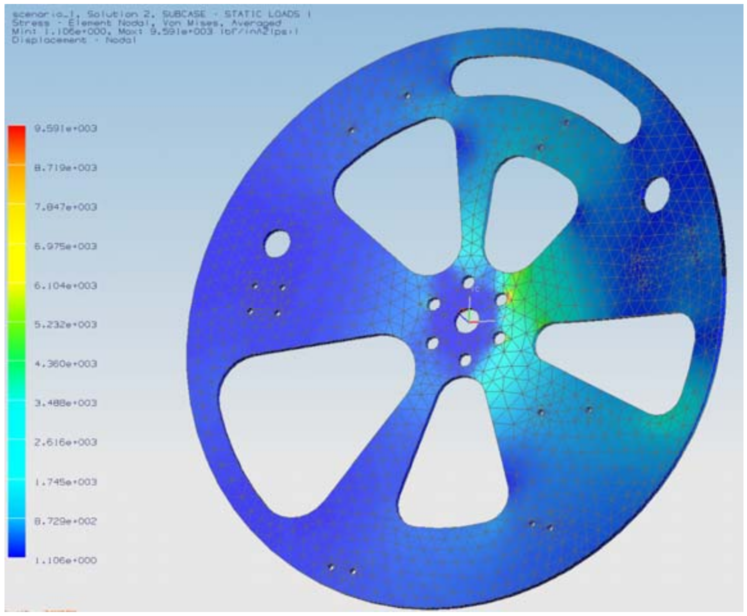
Characteristic	Value
Density [lbf/in^3]	0.067
Young's modulus [lbf/in^2]	$2.0*10^6$
Flexural strength [psi]	55000
Rockwell hardness	110
Shear strength [psi]	19000
Poisson's ratio	0.12

Table 10: G10 material properties



Solver: Structures P.E. // Analysis type: structural // Solution type: linear statics single constraint // meshing: 3D tetrahedral

Figure 7: FEA of superbracket caused by torque from hydraulic motor



Solver: Structures P.E. // Analysis type: structural // Solution type: linear statics single constraint // meshing: 3D tetrahedral

Figure 8: FEA of superbracket caused by torque from hydraulic pump

8.2. Gearing System

A schematic of the gearing system is shown in Fig. 9. The required torques (labeled as T), are established by the Fall 2006 team using the known speed and torque specifications of the hydraulic motor/pump. These values are valid since the hydraulic motor/pump will be re-used in our RBS (described in concept generation section). The numbers on adjacent circles represent the ratio of the number of teeth. When the bicycle is pedaled (the main gear is spinning) both spur gears will be spinning. The clutch bearing prevents the motor from spinning unless the motor is engaged during launch.

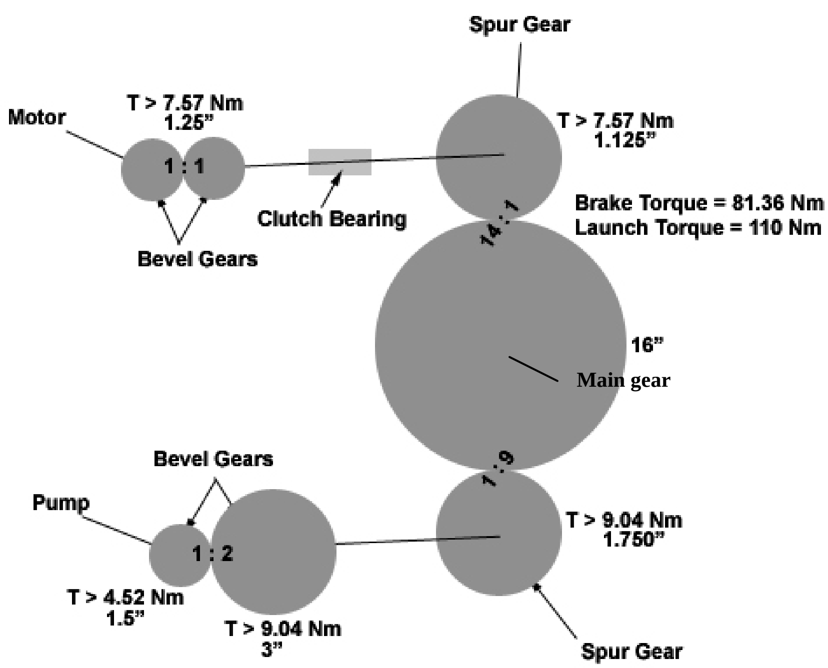


Figure 9: Schematic of gearing system

8.2.1. Spur gears and bevel gears

The gears in the Fall 2006 prototype will be re-used in our RBS as requested by our sponsor. We re-calculated the suitability of the gears to justify our decision. The Lewis equation (Barth revision) [9] given in eqn. (2) was used to get the safe tooth load W in lbf. A factor of 0.75 is applied to the calculations for bevel gears because the gears do not run under prolonged strenuous loading conditions. The maximum allowed torque that should be imposed on a gear would be the product of W and half of the pitch diameter, D. In eqn. (2), S is the safe material stress in psi given by the manufacturer; F is the face width in inches; Y is the tooth form factor given by the manufacturer; P is the diametral pitch; D is the pitch diameter in inches; and V is the pitch line velocity in feet/min calculated as 0.262 x D x RPM. The gear dimensions were obtained from the manufacturer's catalogue. The pairing gears were checked to have the same circular pitch and pressure angle to ensure meshing. Appendix B presents the details of the calculations.

$$W = \frac{SFY}{P}\left(\frac{600}{600+V}\right) \tag{2}$$

The gears were found to have several merits. They are made of suitable material (steel or hardened steel), commonly available pressure angles (14.5° and 20°), and have relatively large face width compared to the other selections in their respective class (thus reducing loading stress and ensuring longer lifespan). In addition, the bevel gears were found to have a relatively low diametral pitch of either 16 or 12, which ensures the transmission of more power [10].

8.2.2. Main gear

As requested by our sponsor, material was removed from the solid steel 16" diameter main gear used by the Fall 2006 team such that spokes would link a center hub to the gear rim. Consider the maximum braking torque, T = 130 N-m, and Ns number of spokes in the main gear. Each spoke can be represented as a cantilever beam of rectangular cross section area of breadth b (spoke thickness) and height h (spoke width) with moment of inertia I. The spoke is loaded such that the product of the force at its tip, F (the rim of the gear) and the spoke length, r_{spoke}, equals to T/Ns to balance the torque about the center (hub). Figure 10 shows the key dimensions of the main gear used in calculations.

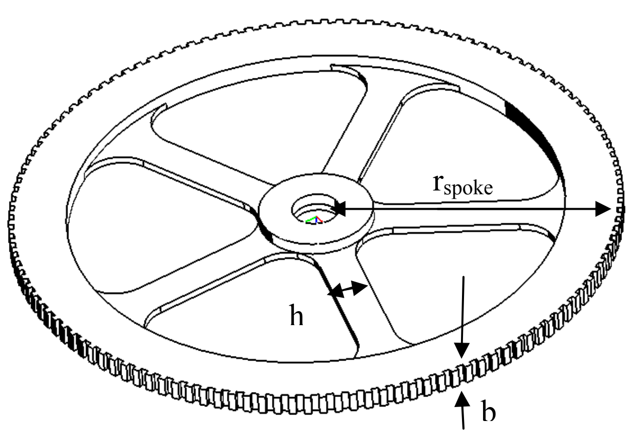


Figure 10: Key dimensions of main gear

Failure under such loading occurs at the intersection of the hub and the spokes. Consider two extreme scenarios. One is at the edge of the hub, where there is zero shear stress due to transverse loading but there is maximum normal stress due to bending. The other is at the neutral axis of the spoke, where there is maximum shear stress but zero normal stress. Consider the principal stress equal to the normal stress (with zero shear stress) in the former case, and principal stress equal to the shear stress (with zero normal stress) in the latter case as follows. These values are compared to the yield stress of material to check for failure.

The magnitude of the maximum normal stress at the edge of the hub due to the bending moment is given by eqn. (3), where M is the bending moment, y is the furthest distance from the neutral axis, and I is the moment of inertia of the cross section of the beam.

$$\sigma = \frac{M \cdot y}{I} = \frac{\frac{T}{Ns} \cdot \frac{h}{2}}{\frac{b \cdot h^3}{12}} = \frac{6 \cdot T}{Ns \cdot b \cdot h^2} \tag{3}$$

The maximum shear stress at the neutral axis of the spoke due to F is given by eqn. (4), where V is the transverse load, Q is the first moment of the sheared section with respect to the neutral axis, I is the moment of inertia of the cross section of the beam, and t is the thickness of the beam. Details on the calculations are provided in Appendix C.

$$\tau = \frac{V \cdot Q}{I \cdot t} = \frac{F \cdot \frac{b \cdot h}{2} \cdot \frac{h}{4}}{\frac{b \cdot h^3}{12} \cdot b} = \frac{3 \cdot F}{2 \cdot b \cdot h} \tag{4}$$

To check our calculations, a FEA of the main gear was done under the loading conditions shown in Fig. 11. The thrust forces are obtained from calculations provided by the manufacturer's catalogue [9]. Lateral forces are neglected as typical of spur gear force calculations.

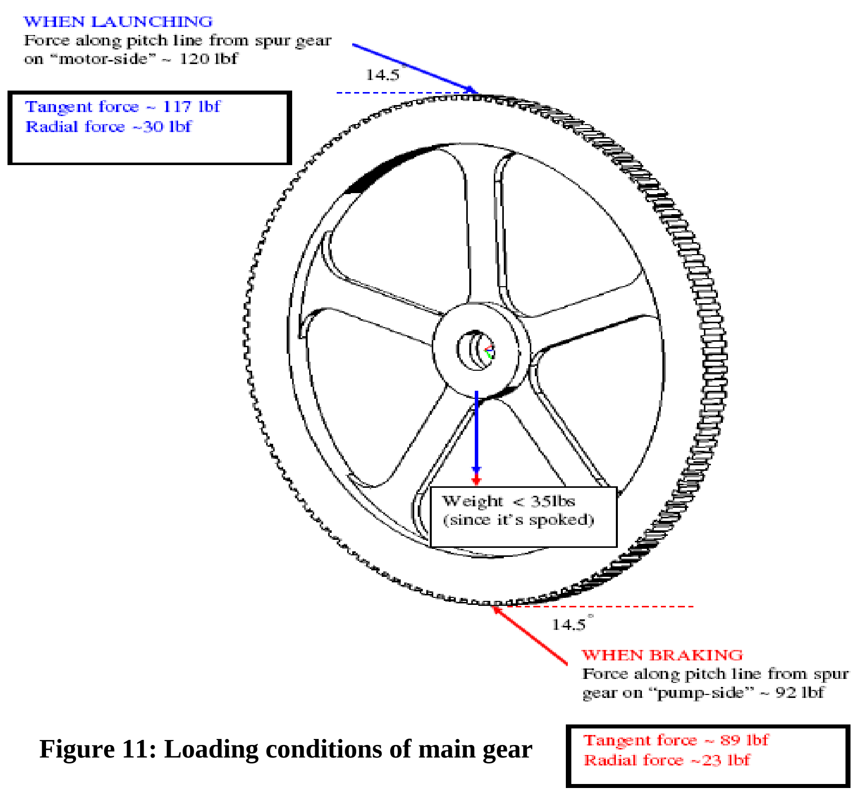
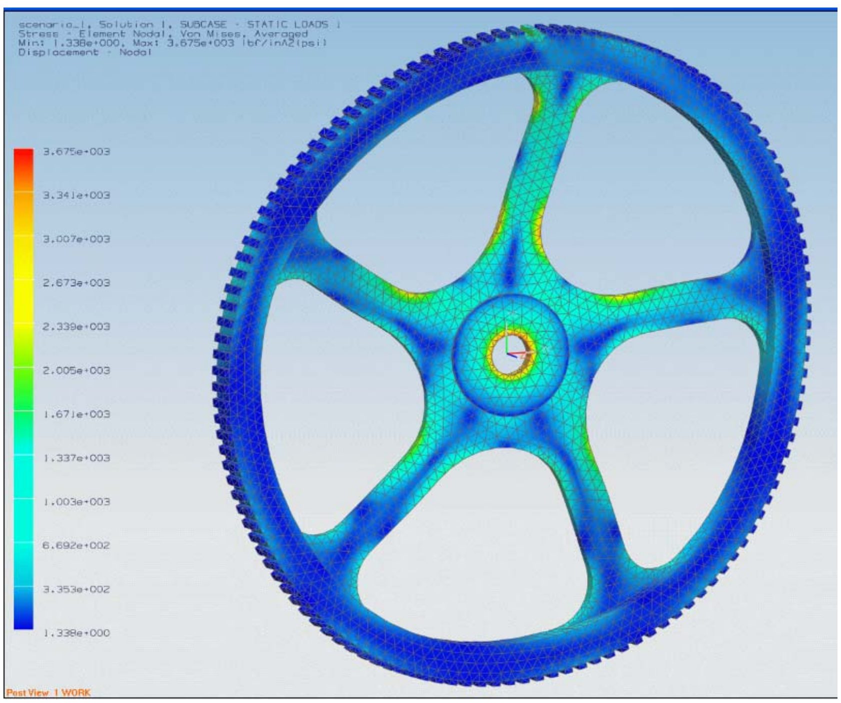


Figure 11: Loading conditions of main gear

Figure 12 shows the FEA of the main gear under loading caused by the gear connections that run through the superbracket. Common steel material properties and characteristics were assigned to the piece and the gear was meshed with thousands of nodes as sites for calculations. Since either braking or launching occurs at any instance, the larger forces under launching conditions were applied. The face of the axle hole is fixed from translation and rotation in the FEA; this provides a conservative analysis since this joint actually has rotational freedom. The regions of the greatest stresses (shown in yellow-red) are found to be below that of the yield strength of steel. From Fig. 12, the maximum stress is around 3700 psi, which is less than a tenth of the yield strength of steel which is over 40000 psi. Thus, the main gear has a safety factor of at least 10. The thicker circumferential path is intended for possible (and optional) implementation to the bicycle front wheel through rigid attachment using fasteners. Having a safety factor of 10 may be seen as an overdesign. However, this ensures that deflection is avoided. Any plastic yielding could make pedaling difficult and loud as all the connecting gears would be permanently misaligned.



Solver: Structures P.E. // Analysis type: structural // Solution type: linear statics single constraint // meshing: 3D tetrahedral

Figure 12: FEA of main gear

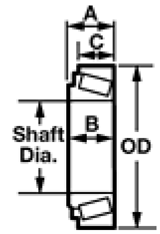
8.2.3. Bearings

The thrust and radial loads on the gears were calculated to select suitable bearings. A total of 4 bearings are used – one for each shaft connecting the spur and bevel gears on either side of the superbracket, one for the main gear, and one clutch bearing for the motor spur gear. The bevel gears produce axial and radial loads, while the main gear produces only radial loads. Thus, we have considered implementing various types of bearings depending on the loads it can take - angular bearings, tapered roller bearings, needle bearings, and ball bearings. In our search, the inner diameter of each bearing has to match the bore of the shaft involved.

Considering a 1/2” inside diameter one-way locking steel needle-roller bearing for the clutch bearing on the motor spur gear, we find that it can handle a maximum angular speed of 18,700 rpm. Therefore this bearing will be tolerable in the RBS.

Figure 13 shows the layout of a tapered-roller bearing that is chosen for each shaft connection. Assuming the bicycle is traveling at 20 mph, the main gear will rotate at 231.8 rpm. With the application of a one-way clutch bearing the maximum angular velocity on either shaft will be 2086.2 rpm. This is calculated using the gear ratio of 9:1 from the main gear to the bevel gear for the pump gear train. The specifications for each bearing are shown in Table 11.

Inner Ring and Roller Assembly



Complete Bearing

Figure 13: Layout of tapered-roller bearing

	Shaft Dia. (in)	OD (in)	Wd. (A)	Bearing No.	Wd. (B)	Radial (lbf)	Thrust (lbf)	Part #
Inner/Outer Ring Pair	5/8	1 11/16	9/16	11590	9/16	1010	1220	5709K11
					3/8			5709K51
	1/2	1 3/8	7/16	A4050	7/16	710	550	23915T11
					11/32			23915T71

Table 11: Specifications for chosen tapered roller bearings

Due to the sponsor’s request to re-use the gears from the previous term, 5/8” and 1/2” bearings were needed to match the bore of the gears. However, this is not ideal for manufacturing and assembly. For uniformity and ease of manufacturing, similar bearings will be used on each shaft for the different gear trains. While the prototype will use two different size shafts, the actual design will use two 1/2” shafts and two 1/2” bore bearings. Given an L10 value of 1 million revolutions for a 550 lbf thrust load and knowing that the product will be handling about 36 lbf, eqn. (5) was used to determine the approximate lifetime of the bearing assuming 90% efficiency.

$$L_D = L_{10}\left(\frac{C_{10}}{F_D}\right)^a \tag{5}$$

Where ‘a’ is 10/3 for roller bearings and 3 for ball bearings. First the design load needs to be calculated from the combination loading of thrust and axial forces. Eqn. (6) is first used to determine the ‘e’ value, and is then compared to the condition shown in eqn. (7).

$$\frac{F_a}{C_o} \tag{6}$$

$$e \geq or < \frac{F_a}{VF_r} \tag{7}$$

$$F_e = X_i VF_r + Y_i F_a \text{ ; V=1 for rotating inner race} \tag{8}$$

For the 1/2" tapered roller bearing, X_i and Y_i were found to be 0.56 and 1.63 respectively because 'e' was less then the ratio shown in eqn. (7) on p. 24. Using these results in eqn. (8) resulted in an equivalent radial load of about 78.84 lbf. This was calculated using the worst case scenario of a thrust load of about 36 lbf and an axial load of about 36 lbf (the approximate total weight of the prototype). Due to the moderate impact involved in the application of the prototype on a bicycle a load factor of about 1.5 was used to calculate a design load of 118.26 lbf through eqn. (9).

$$F_D = 1.5 F_e \quad (9)$$

$$L_{hr} = \frac{L_D}{n_{rpm} 60 \frac{hr}{\min}} \quad (10)$$

Substituting this design load into eqn. (5) on p. 24, the 1/2" roller bearing was found to have a lifetime of $393.31 * 10^6$ revolutions or 3142 hrs assuming 2086.2 rpm and using eqn. (10). Similarly, for the 5/8" roller bearing X_i and Y_i were found to be 0.56 and 1.99 respectively using eqn. (7) and the equivalent load was found to be about 137.7 lbf using a similar load factor of 1.5. Using eqn. (5), this results in a design lifetime of $766.7 * 10^6$ revolutions or 6125 hrs assuming 2086.2 rpm and using eqn. (10).

Figure 14 shows the layout of a flanged ball bearing chosen for the main gear.

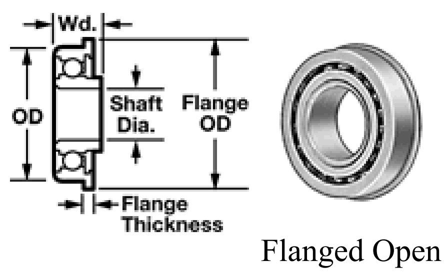


Figure 14: Layout of flanged ball bearing

With the main gear rotating at 231.8 rpm and considering the application of a one-way clutch bearing, the maximum angular velocity of the main gear will be 231.8 rpm. The flanged double sealed bearing can operate at a maximum speed of 1000 rpm so the designed operating speed is significantly less. The specifications for the bearing for the main gear are shown in Table 12.

Type	Shaft Dia. (in)	OD (in)	Width (in)	Radial (lbf)	Part #
Flanged Double Sealed	1.0	2.000	9/16	672	6384K373

Table 12: Specifications for chosen flanged double sealed bearing

Therefore for an L10 value of 1 million revolutions for a 30 lbf radial load and knowing that our product will be handling about 10 lbf, we can use the lifetime eqn. (5) on p. 24 to determine an approximation lifetime of the product assuming 90% efficiency. Using an 'a' value of 3 for ball bearings this gives us a lifetime of about 83 years. Therefore, this bearing will be more than acceptable.

8.2.4. Gear shafts

The steel shafts that connect the spur gear to bevel gear for either of the pump and motor in the Fall 2006 prototype were replaced with aluminum to reduce their weight by at least two thirds (density of aluminum is about a third that of steel). The bulky gear housings were over-designed in preventing deflection. They were removed since each shaft is supported in three places - a bevel gear on the superbracket side, a spur gear on the main gear side, and a bearing at the shaft hole of the superbracket.

Since deflection is not a major concern for the restrained shafts, we checked for yielding of the shafts. The maximum stress of each shaft at the superbracket shaft hole support is determined using eqn. (11), where the normal axial force and the bending stress caused by the separating forces (P_1 and P_2 respectively) of each of the bevel gear set were added (mentioned in superbracket analysis section). L is the distance between the end of the shaft connected to the bevel gear and the support, and r and A are the radius and the cross sectional area of the shaft respectively. The denominator in the second term on the RHS of eqn. (11) is the moment of inertia for a circular cross section. The maximum stresses were found to be far below that of a typical yield stress of aluminum (larger of the two maximum stresses is in the region of 1400 – 1500 psi, compared to a yield stress of over 70000 psi for 7000 series aluminum).

$$\sigma_{\max} = \frac{P_1}{A} + \frac{P_2 \cdot L \cdot r}{1/4 \cdot \pi \cdot r^4} \tag{11}$$

8.2.5. Bolts

Standard 1/4” steel bolts were used to mount the supporting structures for the gearing system onto the superbracket for manufacturability. In turn, the superbracket is mounted on a display bicycle axle using 5/8” steel bolts re-used from the previous prototype. These large bolts have been determined by the previous team to be sufficient for the design strength. It would be tedious and unnecessary to analyze the stresses on every 1/4” bolt on the superbracket. Instead, the bolts with the greatest potential to fail were considered in our analysis. Most bolts were used to secure stationary components onto the superbracket and these components are lightweight enough to intuitively know that the bolts do not shear. The bolts in concern, however, would be those that are used to mount the bracket holding the motor/pump because these components experience separation thrust forces when the bevel gears are in operation (see Fig. 6 on p. 18). Since two bolts are used to mount either pump or motor bracket, the largest shear force experienced by any one 1/4" bolt would be 7.5 lbf (half of the 15 lbf as shown in Fig. 6 on p. 18). This gives an average shear stress of around 153 psi, which is much less than the listed strength for the standard ASTM A307 bolt, which is 60000 psi [11].

8.3. Hydraulic System

8.3.1. Accumulators

A PETE plastic low pressure accumulator will be used since it is light weight and suitable for use with standard hydraulic fluid. After analyzing the high pressure accumulator for three different sizes from Parker (0.32*l*, 0.5*l*, and 0.75*l*) the 0.5*l* accumulator was found to be more than adequate for storing the required amount of energy, and this will be used as the high pressure accumulator. More information on the accumulators is presented in Appendix D.

8.3.2. 3-way valves

Two solenoid operated 3-way Parker valves (model DSH083-N-omit-omit-D012-L-P-R-A-6T) were used in our design. This model is selected among its class because it is relatively lightweight, has suitable maximum operating pressure of 5000 psi, relatively low pressure drop across different flow rates, suitable operating flow rates, and universal compatibility in attaching piping to its outlet using SAE 6 fittings and operating it with a 12V power source. Technical details are omitted in this report due to its page length; more detailed information can be obtained from commercial catalogues [12].

8.3.3. Fittings

The pressure drops related to the geometry of the fittings vary depending on the types of fittings to be used [13]. To predict the pressure drop, we make an initial assumption that each type of fitting is connected to the piston accumulator, containing hydraulic fluid. Then we run the fluid with an initial velocity (V_1) ranging from 1 to 4 m/s as the piston gets pushed inside the accumulator. Using eqn. (12), we calculate the fluid velocity running at the fittings, V_2. The accumulator has a diameter of 0.064 m. Then using eqn. (13), we calculate the loss coefficient (h_L) in each case. Each fitting has different minor loss coefficient (K_L); The JIC 90 deg sharp bend fitting has a K_L of 0.3, JIC 90 deg smooth bend has a K_L of 0.2, JIC tee-flanged line flow has a K_L of 0.9, JIC tee-flanged branch flow has a K_L of 1.0. Then using eqn. (14), we obtain the pressure drop (Δp) for each fitting, where γ is 10.45 kN/m^3.

$$V_2 = \frac{V_1 A_1}{A_2} \tag{12}$$

$$h_L = K_L \frac{V_2^{\,2}}{2g} \tag{13}$$

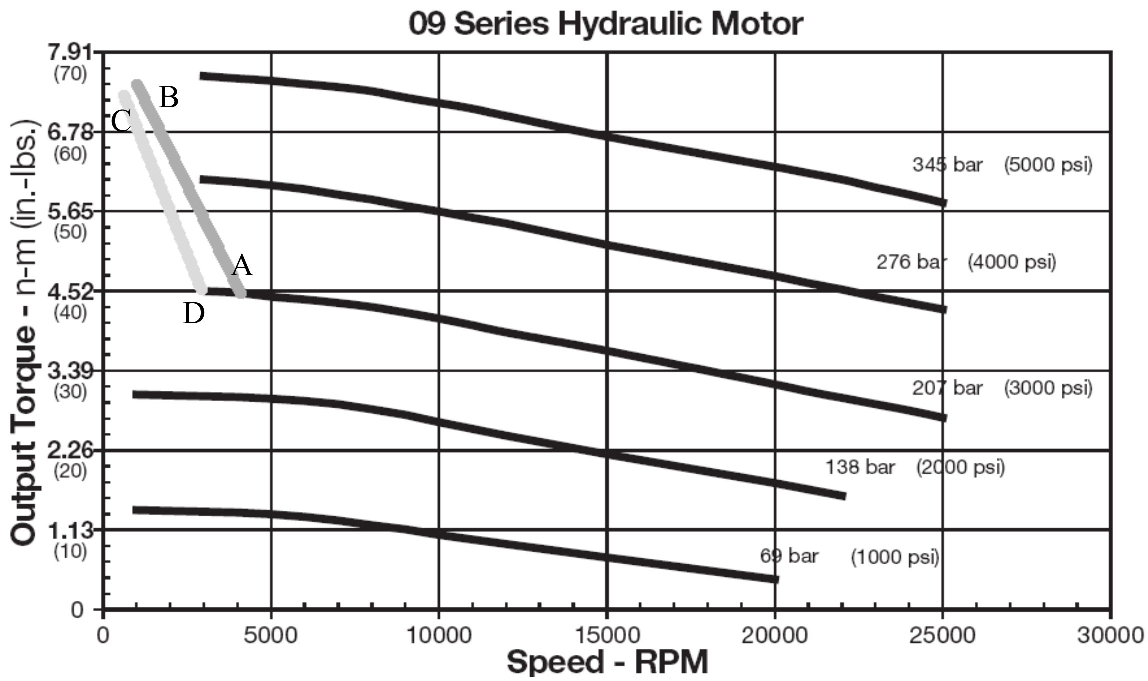
$$\Delta p = \gamma h_L \tag{14}$$

The purpose of these calculations is to determine the most suitable fitting size to be used in our application. We found that the JIC of size 6 fitting is the best fitting since it has the best compromise between pressure drop and weight. The analysis for JIC 90 deg sharp bend, JIC 90 deg smooth bend, and JIC tee fittings are summarized in Appendix E. Looking at the graphs, we predict that the other fittings that are used in our project (JIC 45 deg bend fittings, SAE-JIC adapter) will follow the same trend as these three analyzed fittings, i.e. the percentage of pressure drops in JIC 4 is much higher than those of JIC 6 and 8.

Due to piping shape constraints and resource availability under manufacturing time constraint, other sizes will inevitably be used. The specific fittings cannot be planned exactly. We chose the best fittings (in terms of size and part counts) during our assembly in the EPA workshop where a large variety of fittings are readily available.

8.3.4. Hydraulic pump/motor

As mentioned earlier in concept generation section, the hydraulic pump/motor are re-used since they meet the engineering specifications and their torque-rpm characteristics are well established. The operating characteristics of the hydraulic pump/motor are shown in Fig. 15.



The dark gray line shows the braking cycle; the light gray line shows the launch cycle. The peaks of each motor operation curve are the most efficient operating points at a given pressure. Graph provided by manufacturer.

Figure 15: Operating characteristics of hydraulic pump/motor

During braking: At about 20mph bicycle travel, a 29” diameter bicycle wheel spins the rigidly attached main gear at about 230 rpm (using v = rw). The 18:1 gear ratio of the pump gear train turns the hydraulic pump at 4140 rpm, corresponding to 4.52 N-m torque at 3000 psi (A). Although our accumulator is pre-charged at 2800 psi, there is little difference in these values. This translates to a braking torque of about 81.36 N-m applied to the main gear due to the 18:1 gear ratio (notice this does not exceed the braking torque limit of 130 N-m listed in the engineering specifications). The charging is deemed to end at around 5mph (B), whereby the cyclist typically ends braking. The dark gray line is drawn by calculating various speeds and then fitting a straight line through, with an assumption of linear deceleration (using F = ma). With an assumed total mass of 100 kg, this gives an acceptable deceleration starting from around 2.2 m/s^2 and increasing gradually up to around 4 m/s^2.

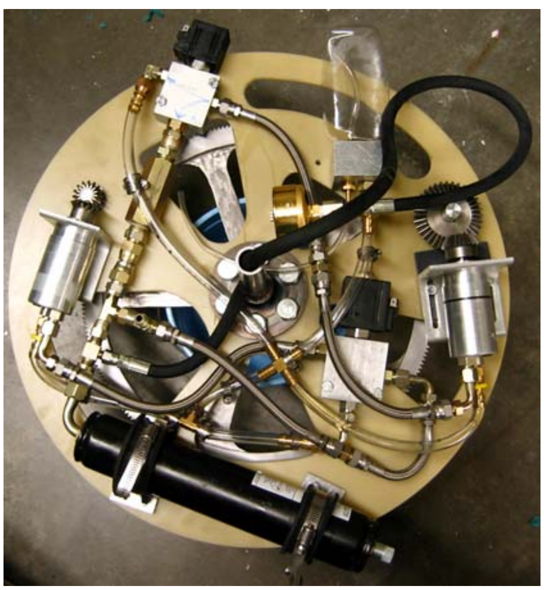
During launching: Spinning the hydraulic motor by releasing the pressure from the fully charged 5000 psi accumulator generates about 7.57 N-m of torque (C). The 14:1 gear ratio of the motor gear train applies a 105 N-m torque to the main gear. Taking into consideration friction of the tires, this roughly falls around the 90 N-m launching torque requirements listed in the engineering specifications. The motor torque of 7.57 N-m corresponds to around 800 rpm, which turns the main gear at around 57 rpm due to the 14:1 gear ratio. Using v = rw, this translates to a bicycle travel of around 5mph, which is a comfortable initial speed. The acceleration is deemed to end until the pressure falls to 3000 psi (D). The light gray line is drawn by calculating various speeds and then fitting a straight line through with an assumption of linear deceleration (using F = ma). With the smaller gear ratio, the acceleration is “gentler” than the situation in braking. With an assumed total mass of 100 kg, this gives an acceleration starting from around 2.8 m/s^2 and decreasing gradually up to less than 2 m/s^2.

It is because of the different magnitude of deceleration/acceleration necessary for comfortable (and safe) travel that two different gear ratios are used in the design.

9. FINAL DESIGN

Figure 16 shows the final design with the major components labeled. Compared to Fall 2006’s prototype shown in Fig. 2 on p. 5, it is evident that the prototype weight, width, and piping complexity is improved. Due to shipment delay for the 3-way valves, replacement valves close to the size of the intended 3-way valves were used for the design expo display (the intended valves are smaller, lighter, and more robust).

Since the future teams will incorporate the RBS in the bicycle front wheel, the superbracket and main gear are not permanently fixed to the axle to allow for easy take-out. The superbracket is connected to the axle plate by six bolts while the main gear is restrained by U-bolts to prevent it sliding along the axle during rotation.



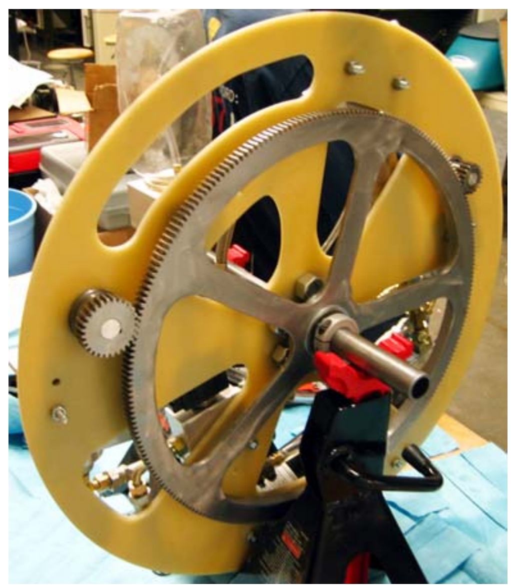


Figure 16: Photographs of either sides of the final prototype

The RBS will fit in a 29” diameter hub with a thickness of around 4”. Since the gears and the hydraulic pump/motor are re-used from the Fall 2006’s prototype, the torque and speed specifications are immediately satisfied. The smaller low pressure accumulator bottle used in our design clearly meets the < 1L volume requirement. The prototype weighs around 30 lbf excluding the display axle and fastener bolts. Significant weight reductions were made in the four major components, and are tabulated in Table 13 on p. 30.

Major components	Fall 2006 prototype	Current prototype	Reduction
Superbracket	12.1	4.2	7.9
Main gear	35.0	8.3	26.7
High pressure accumulator	6.0	4.0	2.0
3-way valves	8.5	0.8	7.7
Total	**61.6**	**17.3**	**44.3**

Table 13: Weight reduction for the four major components (units in lbf)

10. MANUFACTURING

The prototype mainly consists of hydraulic components purchased from commercial vendors and only the superbracket and main gear were specially designed and manufactured. The shafts connecting the spur and bevel gears on either side of the superbracket are cut from standard cylindrical aluminum stock and lathed to fit the inner diameter of the bearings and thus require no design. For easy reference, the manufacturing plans for the superbracket and main gear are provided along with their drawings in Appendices A and C respectively. Due to the complexity of the component variety, the manufacturing plan is neatly shown in Table 14 for easy reference.

Component	Source	Notes
Superbracket	G10 material to replace Fall 2006's aluminum plate	Mill to shape and press fit outer ring of tapered roller bearings at shaft holes
Main gear	Modify solid gear from Fall 2006 prototype	Mill pockets and reduce thickness inside the circumference. Press fit ball bearing at center hole
Axle plate, front axle, and bolts	Provided by Fall 2006 prototype	Re-using these ensures easy take-out for future teams' implementation to the bicycle front wheel
Hydraulic pump/motor		Re-used as per sponsor advice. This ensures engineering specifications are immediately met since the torque data are established by the hydraulic pump/motor
Bevel and spur gears connecting to pump/motor		Gear housings securing the bevel gear sets are removed since motion is sufficiently constrained by the main gear to spur gear connection
Low pressure accumulator		Replacing with a smaller and lighter low pressure accumulator, the <1L engineering specification is met
High pressure accumulator	Purchased a smaller and lighter 0.5L accumulator compared to Fall 2006's 0.75L accumulator	Working range is 2800 – 5000 psi, which complements the hydraulic pump/motor's desired torque-rpm characteristic
3-way valves	Purchased by sponsor	Had to substitute with slightly larger and less robust replacement valves for design expo demonstration
Plastic/reinforced tubing for piping and hydraulic fittings	Purchased from a vendor/provided by sponsor at the EPA workshop - use vinyl tubing for low pressure lines, and braided steel tubing for high pressure lines	Checked with sponsor that tubing material is suitable for standard hydraulic fluid and for the operating pressure
Bearings	Purchased from a vendor	Attached to lathed aluminum shafts; outer ring press fit to superbracket and main gear accordingly; shaft collars with set screw used to prevent motion along the shaft (appropriate since the shaft is a softer material than the set screw)
Batteries	Donated by a vendor	Connected to 3-way valves using appropriately rated wires. Since each valve is rated 14W and batteries is 12V, the rated current has to exceed 1.2A

Table 14: Manufacturing plan of prototype components

11. TESTING

The assembling and testing was done at the EPA workshop under the guidance of our sponsor. The hydraulic components were first flushed with hydraulic fluid to remove any contaminants and a pressure gauge was attached to the high pressure accumulator port to observe pressure changes. The RBS tubing is then filled with hydraulic fluid using a filtered hand pump connecting a bucket of hydraulic fluid to the high pressure accumulator port. At this stage, leakage is expected as pressure builds up. The fittings were tightened accordingly using hand wrenches, and pipe tape was applied to the threads of the checkvalve which had a tendency to leak. As a rule of thumb, further leakage should not occur once leaks observed up to 2000 psi are corrected.

Next, the 3-way valves were activated in turn using the batteries to check that the flow works as designed (described in selected concept section). Unfortunately during our initial testing, valve B failed to electrically switch the port connection. Nevertheless, we have observed that the prototype worked as intended. Without the intended 3-way valves the designed range of 2000 psi to 5000 psi could not be tested, because the replacement valves were intended for usage up to 3000 psi. Valve B was subsequently replaced with a working solenoid in time for the design expo demonstration.

Due to time constraints, comprehensive testing was unable to be accomplished. However an outline of the test has been determined. Testing would involve using a laser tachometer with the front main gear, such that the tachometer can register the rpm of the main gear. Using a preset precharge, we could determine an approximate efficiency of the launch system by measuring the output spin rate of the main gear. In addition, another test was planned using an input on the main gear using a high torque motor to charge the accumulator up to various pressures. The rate of spinning of the main gear could be measured to calculate the deceleration until the accumulator is pressurized to the desired level. Then, by actuating the launch cycle, the acceleration of the main gear could be measured and the top rpm could be measured. Using the maximum initial rpm before charging the accumulator and the output rpm after launch, an approximate efficiency could be quantified for the prototype.

Another test included measuring the temperatures of the motor, pump, and accumulator during the three different flow cycles. This would allow us to determine approximate energy losses due to heat transfer and figure out where are the areas of concern. In addition, acoustic measurements of the main gear were to be taken in order to quiet the gear train either through acoustical engineering or other means.

Further testing can be done when the prototype is placed into the front wheel of a bicycle. Several different tests could be performed: pressure storage for several braking rates, braking within a set distance, launching, and also an efficiency measurement that compares the distance recovered during launch compared to the distance used to pressurize the high pressure accumulator, or the speed recovered during launch compared to the speed used to pressurize the high pressure accumulator.

12. FUTURE IMPROVEMENTS

Further weight reduction can be achieved in the gearing system by combining the role of the main gear and the bicycle hub through the use of a molded plastic wheel hub that incorporates a plastic main gear with inner gear teeth, as described in the concept selection section of this report. An alternative approach would be to design a spider gear, whereby the spokes and central core are replaced by carbon fiber composites to lighten the main gear. In addition, half displacement hydraulic motor/pump, with fittings on the sides as opposed to the back, could be used to eliminate the use of intermediate bevel gears. This would allow the pump and motor to be attached perpendicularly to the superbracket. These two ideas were proposed by our sponsor but were ruled out this semester due to the size of the current motor and pump. In addition, plastic gears could be used to significantly reduce the weight of the gear train and still provide significant strength. This may be achieved by making special custom manufacture requests through UFE (a gear manufacturer).

In terms of hydraulic fittings, we noted that the fittings account for 2.6 lbf while the tubing accounts for only 1.9 lbf. As such, it would be more effective to reduce the weight of the fittings. This can be accomplished by using customized fittings that would eliminate the use of connectors in joining fittings of different sizes. In addition, the brass barbs on the low pressure lines could be replaced with plastic barbs.

Also, to improve safety, as few fasteners as possible should be used. Even though we have determined that the bolts are strong for supporting the components on the superbracket, prolonged ride over bumpy road surfaces could loosen the bolts. Therefore, the accumulators could be directly fastened to the superbracket. This would also decrease the width of the system, since the accumulator protrudes further than any other component with its current mounting brackets. Also, a carbon fiber accumulator could be looked into in more detail to lighten up the high pressure accumulator significantly.

Also, we would recommend implementing slots instead of mounting holes in the superbracket so allow for some assembling flexibility. Theoretical plans for layouts, etc., would need to be changed in order to accomplish this. This idea also makes the system more adaptable to varying realistic conditions.

Much attention should be given to precise alignment of the gears. Any error will lead to a large increase in noise, drag, and tooth life. For these reasons, we recommend reconsidering the mounting style of the pump and motor of our current design for increased robustness. In addition, future teams should note that the clutch bearing on the current prototype was attached to the motor spur gear using epoxy instead of being press fitted. This was done to compensate for a machining error.

The current steel axle and plate weigh at least 10 lbf, and it is definitely overdesigned. Obtaining a different bicycle axle could be a quick way to decrease some weight.

If our layout design needs to be changed for the following semesters, we recommend making these changes the first priority when working on this project. These recommendations take longer than one may estimate, custom fittings/hoses take significant time to order and receive, and these would put a hold on other concurrent progress on the project.

13. CONCLUSIONS

The objective was to design, build and assemble a compact RBS that meets the torque, weight, and size specifications, and can be mounted inside the front wheel of a bicycle. To accomplish this, analysis on the performance of individual commercial components together with that on the overall system are made to determine suitable components. In addition, the main gear and superbracket were designed and fabricated. Significant weight reductions in the major components of the RBS (high pressure accumulator, main gear, superbracket, and 3-way valves) were accomplished in our project, and overall design specifications have essentially been met (see Table 15 below). Our assembled prototype and accomplished work will be carried on by future teams for final implementation in the front wheel of a bicycle.

The RBS eventually hopes to encourage the use of bicycles as the cleaner alternative for city travel by making cycling easier. This is accomplished by re-using energy from frictional braking that would otherwise be wasted. The idea is practical because there is significant stop and go in city travel that would facilitate energy capture. Furthermore, electrical energy consumption in switching the valves are low since electricity is used only during the short braking or launching process.

Engineering Specifications	Target Value	Achieved	Reasons
Wheel width and diameter	4" and 29" respectively	Yes	Components are closely mounted to 22" superbracket
Maximum braking torque	130 N-m	Yes	Same gearing ratio and gears were used as per sponsor request
Top operating speed of bicycle	20 mph	Yes	
Approximate efficiency	> 70%	Yes	
Maximum launch torque	90 N-m	Yes	
Maximum system working pressure	5000 psi	Yes	High pressure accumulator is rated to 5000 psi
Total weight of hydraulic system	< 22 lbf	No	Prototype weighs around 30 lbf
Motor/pump displacement	1.5 c.c.	Yes	Same hydraulic motor/pump were used as per sponsor request
Maximum volume of fluid	1 L	Yes	Low pressure accumulator bottle is clearly < 1L
Learning curve	~ 1 hour	Yes	Simple electric circuit is used to activate valves

Table 15: Design specifications

14. ACKNOWLEDGMENTS

We wish to thank the EPA and our sponsor David Swain for his supervision and guidance on the hydraulic operation of the RBS. EPA's grant to our university has given us a special opportunity to learn about the use of hydraulics in RBS design and has enhanced our undergraduate educational experience. We are also thankful to Bob Coury, Marv Cressey, and Steve Emanuel for their machining advice and in our manufacturing process. In addition, we are especially grateful to Jason Moore for sharing his expertise, tools, and equipment.

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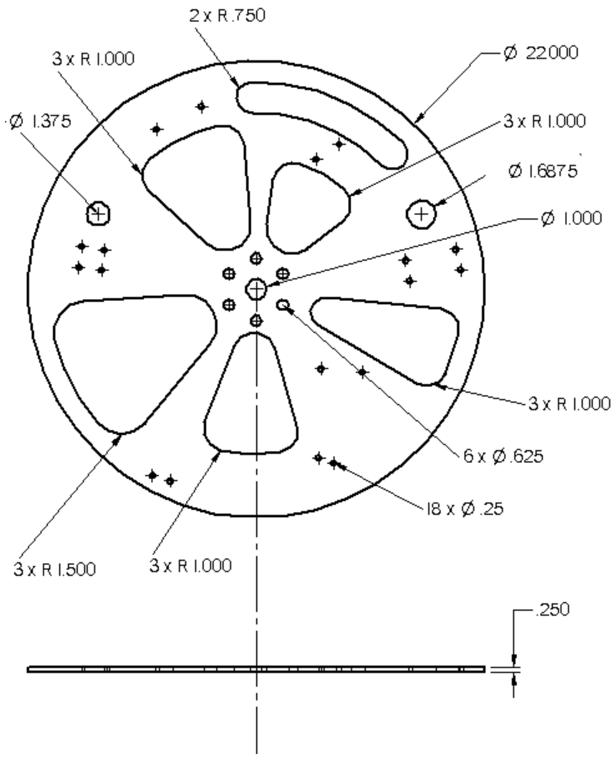
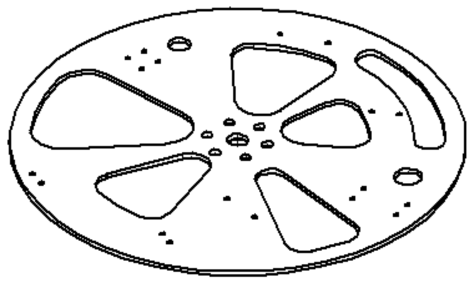
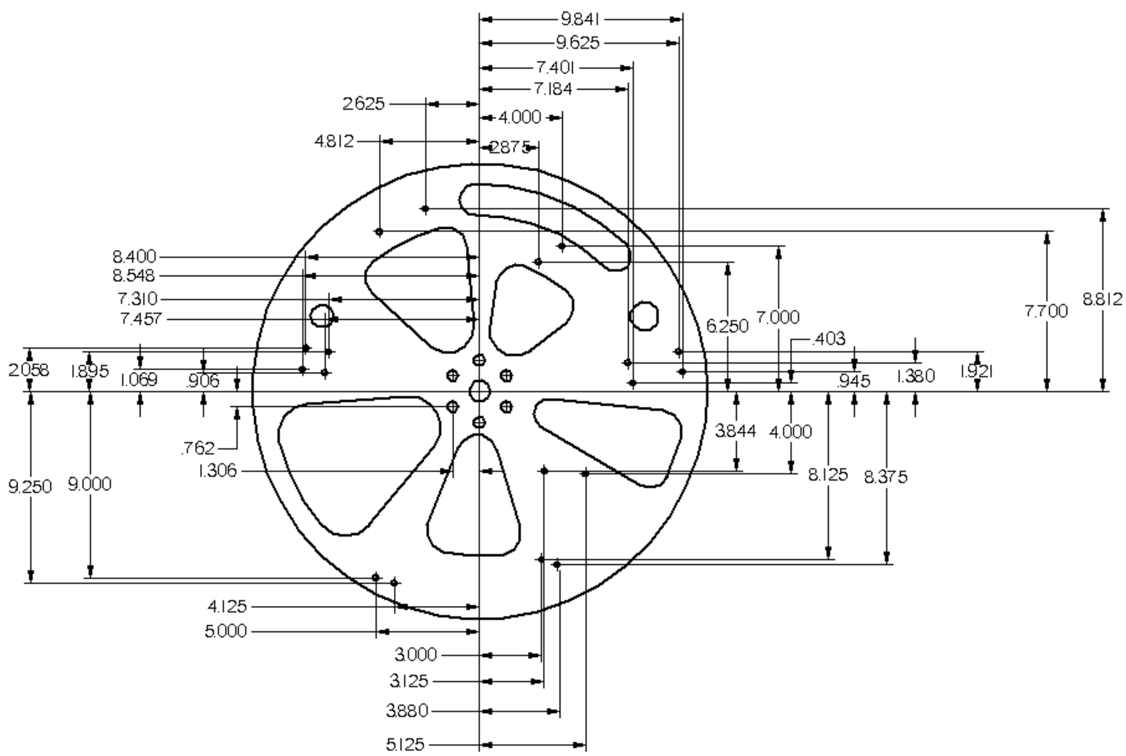
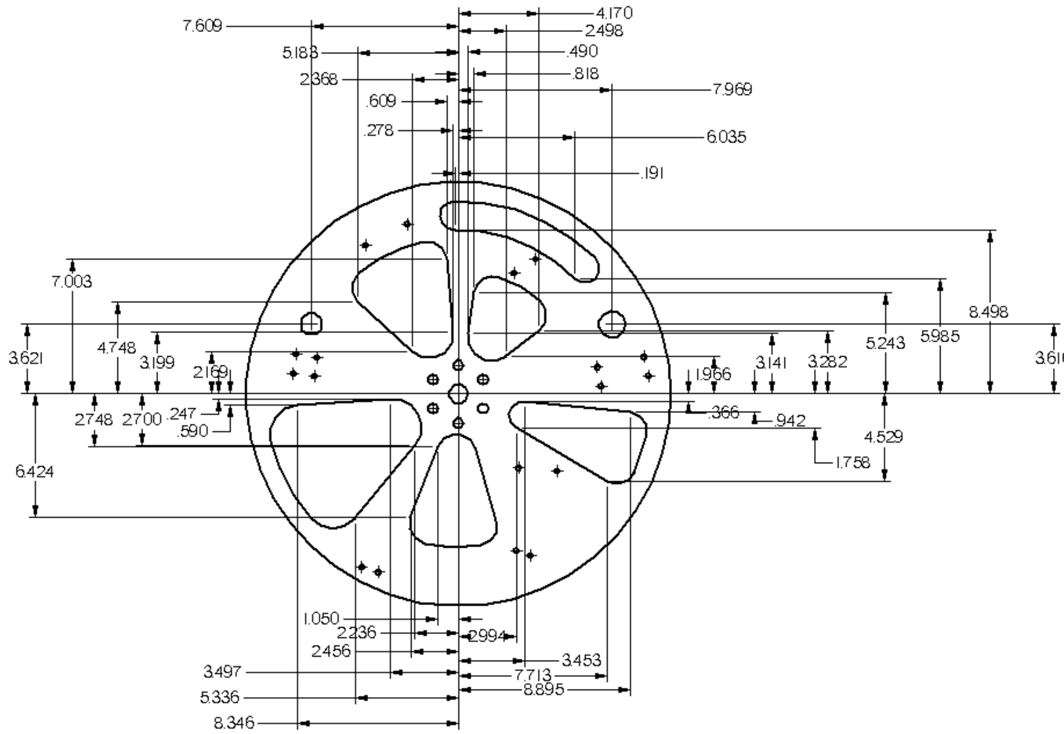
APPENDIX A: DRAWINGS & MANUFACTURING PLAN FOR SUPERBRACKET









Superbracket volume: 63.31 in^3

Superbracket Manufacturing Plan

Part Name: Superbracket

Material: 24” by 24” by 0.25” G10 composite

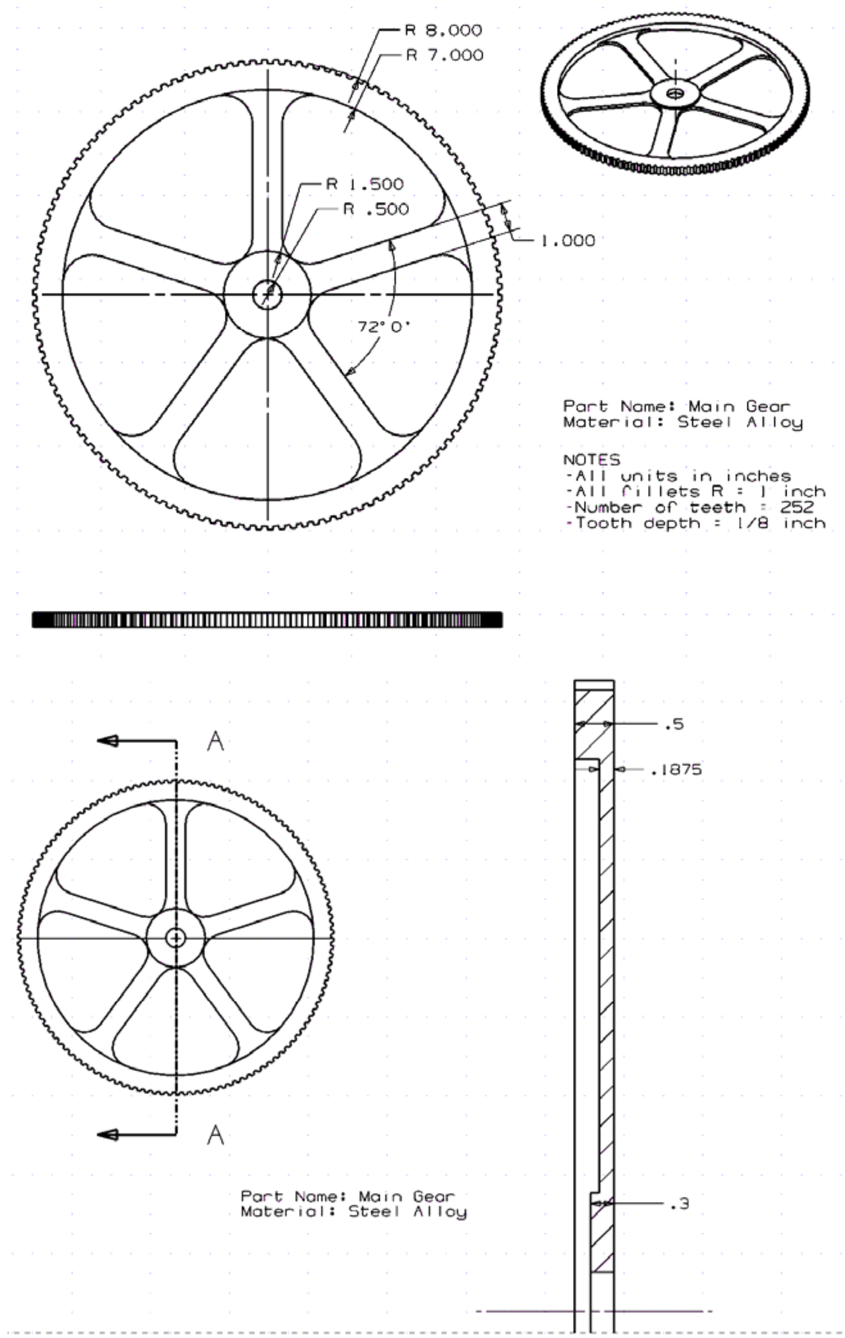
No.	Process Description	Machine	Speed (rpm)	Tool	Fixtures
1	Mill center hole, 6 axle holes, and bearing shaft holes	CNC mill	600	1/2” end mill	Indexing fixture, vise
2	Drill the 18 x 0.25” holes	-	-	Hand drill	Grip clamp
3	Cut the lightning holes out	-	-	Rotary hand tool	Grip clamp
4	Sand the cut surfaces	-	-	Rotary hand tool	Grip clamp

APPENDIX B: SPUR/BEVEL GEARS CALCULATIONS

	Spur gears to main gear		**Bevel gears to pump**		**Bevel gears to motor**
	To motor	**To pump**			
Part #	NB18B-1/2	NB28B-1/2	L152BY-P (1:2 bevel)	L152BY-G (1:2 bevel)	HLK101Y (1:1 bevel)
Material	steel	steel	steel unhardened	steel unhardened	hardened steel (tooth only)
Dia. Pitch, P	16	16	12	12	12
Pitch dia., D(in)	1.125	1.750	1.5	3.0	1.25
Bore (in)	0.5	0.625	0.5	0.625	0.5
Hub dia. (in)	0.94	1.45	1.31	2.12	1.00
Hub proj. (in)	0.44	0.50	0.81	0.88	0.50
Face width, F (in)	0.5	0.5	0.54	0.54	0.29
# teeth	18	28	18	36	15
Pressure angle	14.5°	14.5°	20°	20°	20°
Circular pitch (in/tooth)	0.1963	0.1963	0.2618	0.2618	0.2618
Design RPM	800	2088	4176	2088	800
Safe static stress, S (lbf/in^2)	20000	20000	20000	20000	25000
Tooth form factor, Y	0.270	0.314	0.402	0.415	0.333
Rated torque (substitute bold-faced information in eqn. (2)) (N-m)	7.7	7.48	6.16	12.7	7.42
Required torque with service factor of 0.8 applied (N-m)	6.06	7.23	3.62	7.23	6.06
Rated satisfies Required ?	Yes	Yes	Yes	Yes	Yes

Note: a service factor of 1 represents either moderate shock of 15 minutes per 2hrs or a uniform load of 10 hours a day according to AGMA class of service [9].

APPENDIX C: DRAWINGS, MANUFACTURING PLAN & CALCULATIONS FOR MAIN GEAR



Main gear volume: 29.774 in^3

Main Gear Manufacturing Plan

Part Name: Main Gear

Material: 0.5” thick, 16” diameter mild steel

No.	Process Description	Machine	Speed (rpm)	Tool	Fixtures
1	Hold part in indexing fixture and mount on mill table	CNC Mill	-	-	Indexing fixture, vise
2	Locate center of part	CNC Mill	800	Edge finder	Indexing fixture, vise
3	Enter start/end points into CNC program	CNC Mill	800	End mill (D=.75")	Indexing fixture, vise
4	Copy and rotate pockets 4x	CNC Mill	800	End mill (D=.75")	Indexing fixture, vise
5	Cut circles outside the axle support and inside the gear ring	CNC Mill	800	End mill (D=.75")	Indexing fixture, vise
6	Mount part on mill table	Mill			Vise
7	Manually cut out the material left by the frame of step 4	Mill	800	Face mill (D=1")	Vise
8	Manually shave axle support down	Mill	800	Face mill (D=1")	Vise

Main gear calculations (material properties from efunda.com)

Material	Elastic modulus (GPa)	Yield stress (MPa)	Density (kg/m^3)
Steel	190	280	7700

	Units	Quantity
Number of spokes, Ns	#	5
Spoke width, h	in	1
	m	0.0254
Spoke Thickness, b	in	0.1875
	m	0.0047625
Gear radius, r_gear	in	7
	m	0.1778
Center hub radius, r_hub	in	1.5
	m	0.0381
Spoke length, r_spoke = r_gear - r_hub	m	0.1397
Braking torque, T (Max. value, for conservative design)	N-m	130
Force at tip of spoke, F	N	186.1130995
Moment of inertia (assume rectangular cross section), I	m^4	6.50362E-09
Max. normal stress at edge of hub due to bending, σ	MPa	**50.771755**
Max. shear stress at neutral axis due to transverse loading,τ	MPa	**2.307807**
Estimated mass (theoretical volume by CAD times density)	kg	3.75683

Note: the maximum normal stress or shear stress is well below the yield stress. Either case is assumed as a principal stress due to the configuration considered (as described in report).

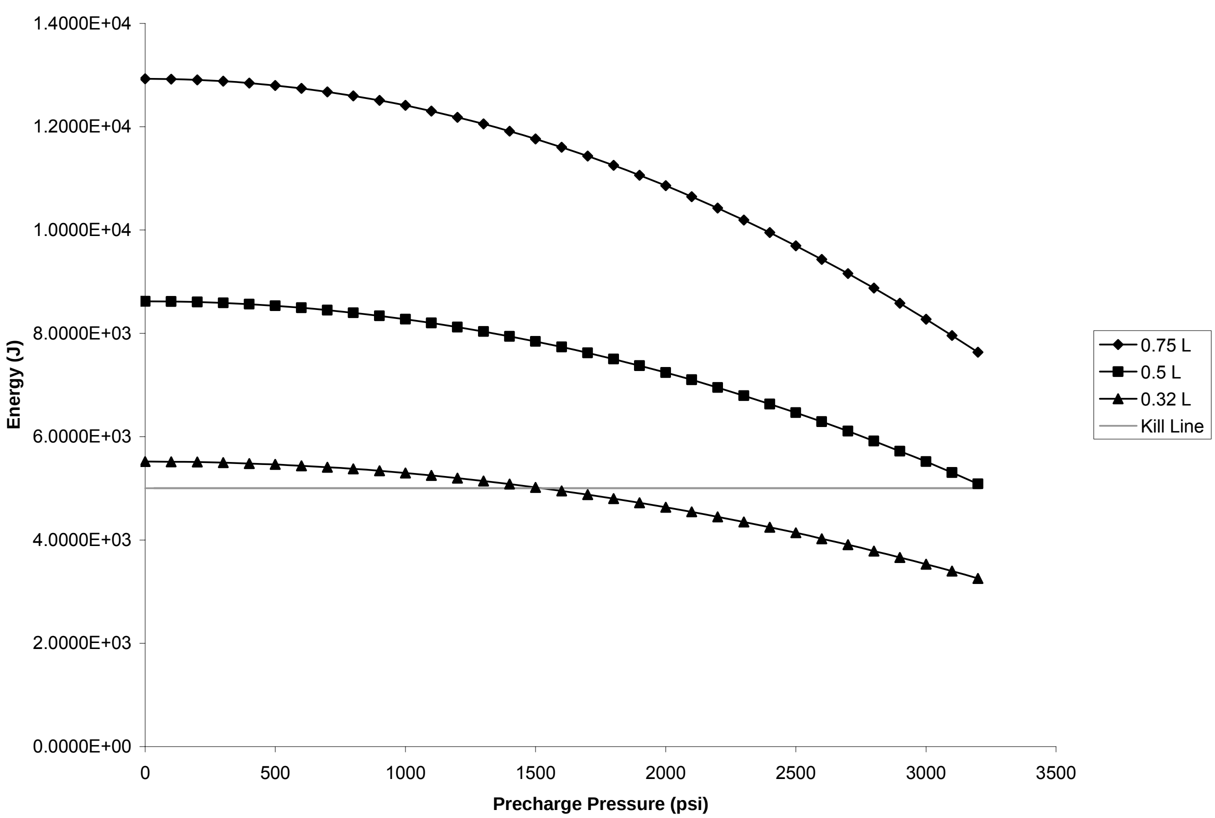
APPENDIX D: ACCUMULATOR ANALYSIS

It is assumed that the overall system is around 80% efficient as determined by earlier teams. Using $E = 1/2mv^2$, releasing 5000 J of stored energy could provide sufficient kinetic energy to move a bicycle with passenger mass of 100 kg at 20 mph.

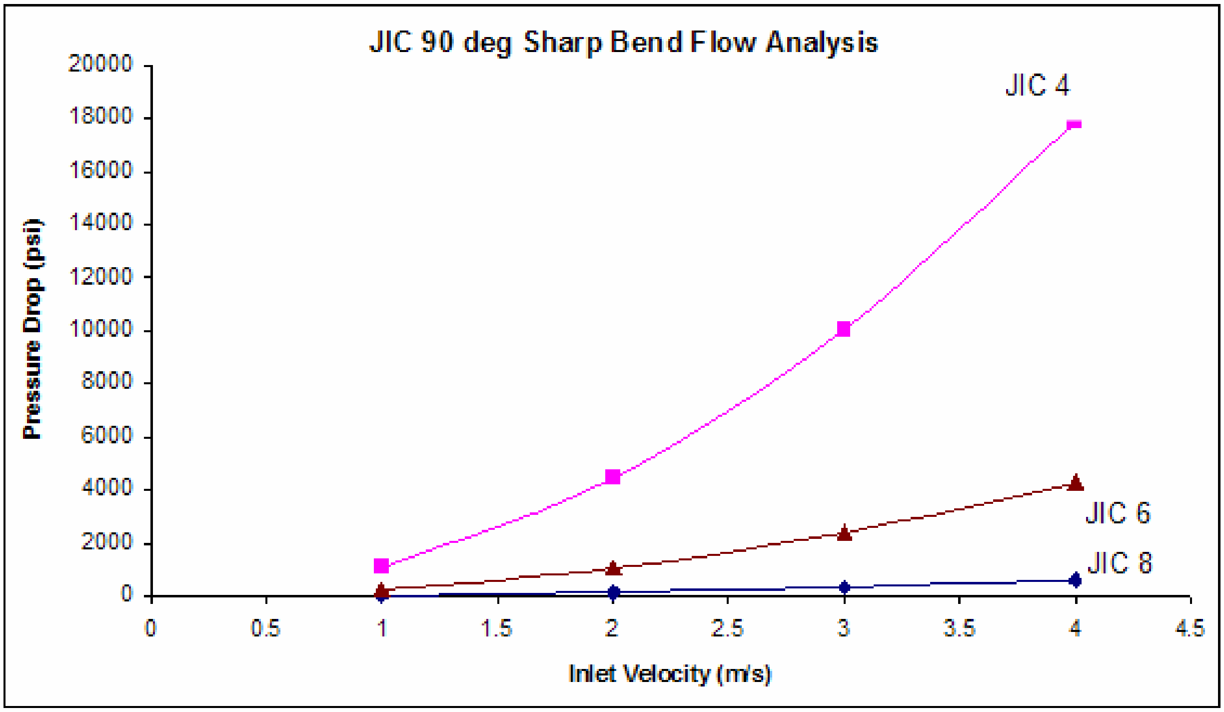
In choosing the high pressure accumulator, we have used the commercial approach of assuming $PV = nRT$ and T constant during the accumulation of pressure. Thus, $P_1V_1 = P_2V_2$, where P_1 is the precharge pressure ranging from 0 to 3200 psi and P_2 is the maximum rated pressure of 5000psi. The precharge improves energy density and allows a reduction in the volume of operating fluid. We have sourced the accumulator from Parker as advised by Mr. Swain. While Parker rates the ACP Series Accumulators with a 50mm bore for a maximum pressure of 4000 psi, after talking with Parker, we will be able to go to 5000 psi without any trouble. Calculating V_2 for a V_1 of $0.32l$, $0.5l$, and $0.75l$ where V_1 is the volume of the cylinder using the equation below.

$$V_2 = V_1 \frac{P_1}{P_2}$$

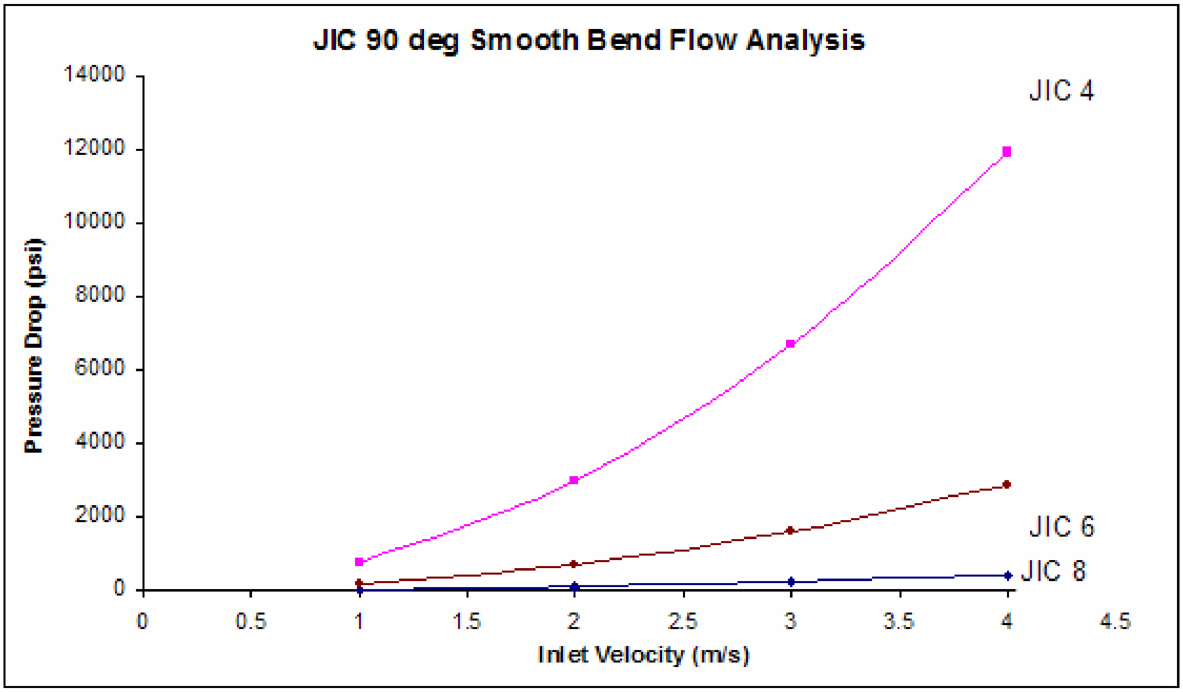
And converting all units to (m^3) and Pa and using $\Delta P = {(P_1+P_2)}/{2}$ and $\Delta V = V_1 - V_2$ we can calculate the energy stored as $E = \Delta P * \Delta V$. The plot below shows that the $0.5l$ accumulator precharged to 3200 psi stores more than the required 5000J indicated by the horizontal line. The $0.75l$ accumulator also fulfills this requirement, but is not chosen because it is heavier.



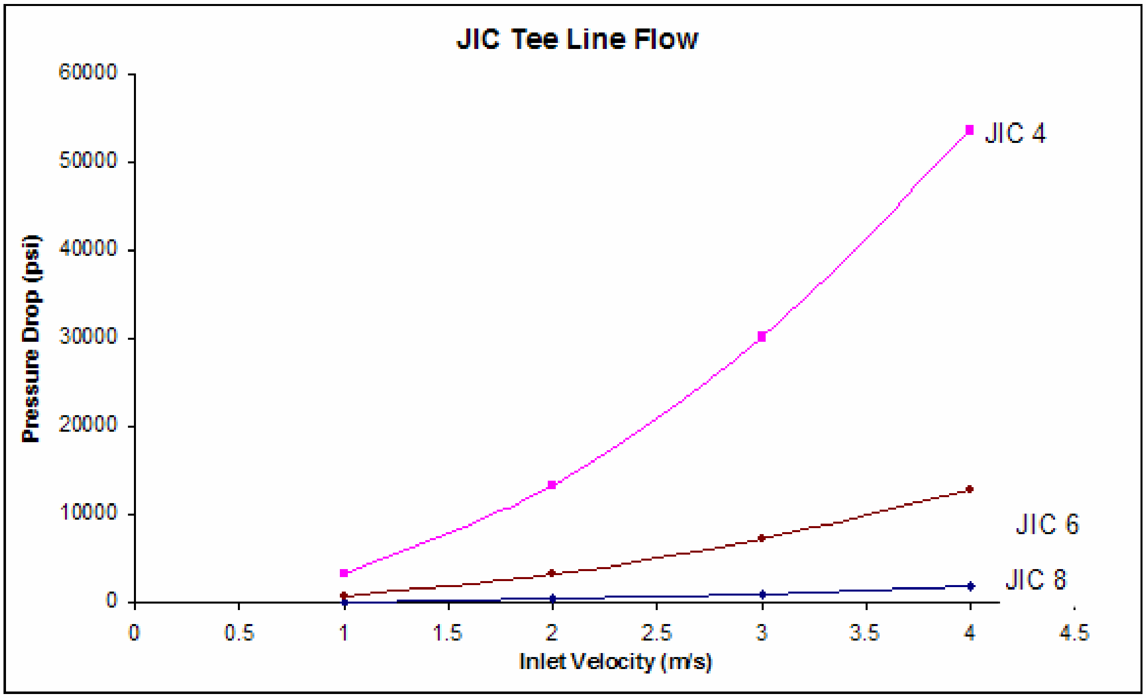
APPENDIX E: COMPARISON OF FITTINGS



Pressure Drop in JIC 4 is much larger than JIC 6 and 8 in 90 deg Sharp Bend Fitting



Pressure Drop in JIC 4 is much larger than JIC 6 and 8 in 90 deg Smooth Bend Fitting



Pressure Drop in JIC 4 is much larger than JIC 6 and 8 in Tee Line Flow

APPENDIX F: BILL OF MATERIALS

Quantity	Part Description	Part Number	Usage	Purchased From	Price (each)	Price (w tax & ship)
1	Parker Hydraulic High Pressure Accumulator	ACP05AA050E1KTC	High Pressure Accumulator	Exotic Rubber and Plastic	138.19	
	(piston accumulator with gas valve)			Jackson Branch, 517-787-1540		
1	Parker Hydraulic High Pressure Accumulator	ACP05AA032E1KTC	High Pressure Accumulator	Exotic Rubber and Plastic	125.81	318.65
	(piston accumulator with gas valve)			Jackson Branch, 517-787-1540		
20	1.2 volts AAA Rechargeable NiMH battery	N/A	Power for Hydraulic Valve	BatteriesPlus	donated	donated
2	10 Pack AAA Battery Holder and Electrical Wiring			Packard Rd		
2	Parker 3 Way Spool Type Valve	DSH083-N-Omit-Omit-D012-L-P-R-A-6T	Hydraulic Valve	Parker through EPA	donated	donated
2	One Way Locking Steel Needle Roller Bearing	2489K4	Clutch Bearing for Gears	McMASTER-CARR	7.82	19.64
	Acetal Cage, 1/2" Shaft Diameter, 3/4" OD, 1/2" Width					
1	Steel Ball Bearing Flanged	634K373	Ball Bearing for Gear	McMASTER-CARR	15.78	15.78
	Double Sealed for 1" Shaft Diameter, 2"OD					
1	Steel Tapered-Roller Bearing Roller Assembly	5709K11	Roller Bearing for Gear	McMASTER-CARR	21.98	21.98
	for 5/8" Shaft Diameter					
1	Steel Tapered-Roller Bearing	5709K51	Roller Bearing for Gear	McMASTER-CARR	8.98	8.98
	Outer Ring for 5/8" Shaft Diameter					
1	Tapered-Roller Bearing Roller Assembly	23915T11	Roller Bearing for Gear	McMASTER-CARR	24.10	24.10
	for 1/2" Shaft Diameter					
2	0.5" shaft diameter C-shape shaft collars		Secure motor spur gear	donated by team member	0.00	0.00
2	1" shaft diameter shaft collars		Secure main gear	donated by team member	0.00	0.00
1	Tapered-Roller Bearing	23915T71	Roller Bearing for Gear	McMASTER-CARR	9.91	17.91
	Outer Ring for 1/2" Shaft Diameter					
1	3" x 3" Angle Iron (1 ft)	N/A	Accumulator/Engine Mounts	ASAP Source, Ann Arbor	12.62	12.62
1	6" x 6" x 1.25" Steel Block	AAA00300	Axle Plate	ASAP Source, Ann Arbor	9.75	9.75
	Fittings (estimated)	N/A	Fittings	EPA	donated	donated
2	Parker 09 Series Hydraulic Gear Motors	09SGCKP	Regen/Pump Motors	EPA	donated	donated
1	Boston Spur Gear (0.5" Bore)	NB18B-1/2	Motor Spur Gear	Motion Industries	19.10	19.10
1	Boston Spur Gear (5/8" Bore)	NB28B-5/8	Pump Spur Gear	Motion Industries	28.20	28.20
1	Boston Bevel Gear (5/8" Bore)	L152BY-G	Pump Bevel Gear	Motion Industries	52.20	52.20
1	Boston Bevel Gear (0.5" Bore)	L152BY-P	Motor Bevel Gear	Motion Industries	32.40	32.40
2	Boston Miter Gear	HLK101Y	Pump/Motor Gear	Motion Industries	28.20	56.40
1	Steel Spur Gear 16" Pitch / 252 Teeth	N/A	Main Gear	Ann Arbor Gearing Technologies	donated	donated
1	Pair of Jack Stands	70030567058	Prototype Stands	Walmart	17.89	17.89
1	29" Bicycle Rim	N/A	Bicycle Rim	Wheels in Motion, Ann Arbor	57.24	57.24
3	Hose Barb Tee (3/8")	4029732	Low Pressure Hosing	Ace Hardware, Ann Arbor	5.99	17.97
11	Hose Clamp	41915	Low Pressure Hosing	Ace Hardware, Ann Arbor	1.29	14.19
26	1/4-20 Bolts	*56	Assembly	Ace Hardware, Ann Arbor	0.08	2.08
26	1/4-20 Nuts	*56	Assembly	Ace Hardware, Ann Arbor	0.20	5.20
6	5/8" Bolts	*56	Assembly	Ace Hardware, Ann Arbor	0.35	2.10
1	8" Tube 3/8" ID	4027504	Low Pressure Tubing	Ace Hardware, Ann Arbor	3.12	3.12
1	Magnet Ceramic Block	2108553	Low Pressure Accu Filter	Ace Hardware, Ann Arbor	3.49	3.49
1	Low Pressure Accumulator	Honey Bottle	Low Pressure Accumulator	Team Member	donated	donated
1	Hydraulic Oil	N/A	Hydraulic Oil	EPA	donated	donated
1	G10/FR4 Sheet 24x24x0.25"	GEEX.2502424N	Superbracket	Accurate Plastics	100.89	113.68
					TOTAL	874.67