Parker Hannifin Chainless Challenge 2014-15 Senior Design Project

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1. Introduction

Our team name is Bike Under Pressure and our senior project is to create a bicycle that uses fluid to transmit power from the pedals to the drive wheel. The fluid powered bicycle that we create will compete in the 2014-15 Parker, Chainless Challenge Competition. Parker hosts this event to challenge engineering students to combine two normally unassociated technologies: the bicycle and fluid power. The bicycle – due mainly to its chain and sprocket drivetrain – is widely known as the most efficient form of human powered transport. Combining this machine with a fluidic drivetrain presents a challenge in terms of efficiency, weight, and manufacturability. These engineering challenges and unconventional applications force engineering students to develop new and creative designs for hydraulic components and bicycles. At the event to occur in April 2015, around a dozen university teams will compete in races and presentations with their bicycle, including Cal Poly Bike Under Pressure. In executing such solutions Chainless Challenge teams develop a deeper understanding of hydraulics and pneumatics, the engineering design process, and project management.

1.1. Sponsor Background and Needs

The beneficiaries of this project are Parker-Hannifin Corporation, California Polytechnic State University – San Luis Obispo, Cal Poly Chainless Challenge, and – this year's team – Bike Under Pressure. The Bike Under Pressure will benefit from the Chainless Challenge competition by developing engineering skills. The program is designed to teach the value of fluid power and electronics in motion control. It also provides students with first-hand experience in working as an engineering team on a timeline to design, simulate, build, test, qualify, and compete with their concepts, as well as propose their design for commercial production. Parker will benefit from increased university visibility, expanded interest in hydraulics in higher education, student hiring opportunities, and potential innovations. Cal Poly will benefit through sustaining its relationship with Parker Hannifin and receiving specified donations from the competition.

1.2. Formal Problem Definition

Bike Under Pressure is tasked by the Parker Chainless Challenge as specified in the 2014-15 rulebook with designing, manufacturing, and racing a human powered vehicle that has no solid mechanical connection between the power input at the pedals and the power output at the tires. Furthermore, the vehicle must be powered by a single rider. The vehicle may have any number of wheels, but is hereafter referred to as a bicycle for simplicity. The bicycle must be capable of completing four events: a 200 meter sprint race, a team relay race, an efficiency challenge, and a 6.2 mile time trial race which includes a slalom. In order to complete the sprint, relay, and time trial races, the rider must be able to start the bicycle unassisted, maintain a controllable speed, maneuver through the course, and stop unassisted. In order to complete the efficiency challenge, the bicycle must charge an energy storage device and utilize the stored energy without additional rider power input to move a straight distance. In addition to successfully completing the events, the bicycle must weigh less than 225 pounds without the rider, not change between events, be safe for all operators, and use biodegradable fluid if applicable.

2. Background

2.1. Competition

The Parker-Hannifin Chainless Challenge competition has been an annual event since 2005; however the competition did not occur 2009-2011. Parker-Hannifin Corporation is an industry leader in the manufacture of hydraulic equipment; therefore the competition was originally created to spark college engineering student's interest in hydraulics. When the competition began again in 2012, it served to spark student interest in hydraulics and as a means of developing innovative hydraulic regenerative braking ideas. During the early years of the competition, it was common for many of the bikes to fail during the races and teams would resort to pushing their bike across the finish line. However, in recent competitions, the craftsmanship and design have improved to the extent that most teams make it through the competition without critical failures.

The competition has three main components: the midway design review, competition races, and design judging.

The midway review consists of a meeting with a Parker review team. The review is completed either virtually or at Cal Poly. During the review the final design will be evaluated for significant changes from the previous years' designs. If there are insufficient changes to the design, relative to previous years, 25 points will be deducted. If the new design has both a new bicycle frame and new components, 25 points will be added to the overall midway review score. Beyond deviation from the previous years' designs the midway review will evaluate the teams overall vehicle design, fluid circuit design, selection of hardware, analysis (dynamic, fluid flow, expected performance), and some partial prototyping. For each of these categories, 20, 20, 20, 30, and 10 points will be awarded respectively. The total Midway Review score will be added to the Best Paper and Best Presentation award at the competition.

The race component of the competition consists of four events that test individual components of a bike's design and craftsmanship. The below specifications are based on the 2014 competition and may change when paperwork is released for the 2015 competition.

- The first event is the Sprint race. The sprint race consists of a straight 200 meter long drag strip which each team must complete in the shortest time possible. This event tests the bicycle's acceleration and top speed, which are dependent on the weight, drivetrain, and stored energy capability of the bike.

- The team relay race is another event that awards teams for having a good "sprint" time. In this event, each team is randomly paired with another school, and the two teams' sprint times are added together. The pair of teams with the best combined sprint times wins the team relay event.

- The next race is the efficiency event. For this event, each team charges their accumulator to as much as they physically can, and then harnesses this stored energy to power the bike. No pedaling is allowed for the entire duration of the race. The efficiency of each bike is then calculated using the equation shown below, and the team with the best efficiency wins the event.

- The last event is the time trial. The time trial consists of a 6.2 mile race where each team has the option of switching riders after each two mile lap. This race often exposes the weaknesses in both the design and manufacture of the bikes. It is the race where component failure and hydraulic fluid leakage are most common. The team who finishes with the fastest time wins this event.

The distributions of both competition points and monetary awards for each race are as follows:

1st Place → 200 points
2nd Place → 100 points
3rd Place → 50 points
4th Place → 20 points

In addition to the competition points, the winning teams will win the following amounts for their engineering department:

Sprint → $2,000
Team Relay → $1,500
Time Trial → $2,000
Efficiency → $2,000

The second part of the competition is the judging of the designs, which occurs on the second day of competition. Judges evaluate each design on four different categories.

- The first category is innovation, which is judged based on how each design pushes the boundaries of the competition.

- The second category is reliability, which is generally based on how the design performed in the endurance race as well as the frequency of leaks and the occurrence of other small failures during the course of the previous day's competition.

- The third component of the design evaluation judges manufacturability and craftsmanship. The judges look for a well-built bike without excessive components that is well packaged.

- The last component to be evaluated is safety, this includes making sure the bike doesn't have dangerous or sharp protrusions and of course that the components are all running well within their ASME specifications.

Teams are awarded points in each category as follows:

1st Place → 100 points
2nd Place → 50 points
3rd Place → 25 points
4th Place → 10 points

The winner of the Innovation, Reliability and Safety, Manufacturability and Workmanship design categories, will each be awarded a portion of $2,000, 50% to 1st place 25% to 2nd place, 15% to 3rd place, 10% to 4th place, which will go to their respective engineering departments. Furthermore, the winner of Best Design will be awarded $2,000 for their engineering

department. Lastly, the teams with the best report, will be awarded $3,000. The team with the highest overall score is declared the winner of the competition and will be awarded $5,000.

2.2. Possible Drive Mechanisms/Current "State of the Art"

The Chainless Challenge competition rulebook does not specify that a hydraulic fluid system must be used to drive the bike, but most designs for the past ten years have utilized this medium to transfer mechanical power. Pneumatics is the most popular alternative to a hydraulic fluid drivetrain. Pneumatic power systems have failed in the past due to the compressibility of air. Hence the energy input to a compressible fluid such as air is converted to heat rather than work, this is apparent in the ideal gas equation. Furthermore, air is much less dense than hydraulic fluid, thus it requires either extreme speed or pressure to deliver the same amount of energy. For these reasons, pneumatics are used much less and have not been investigated by Bike Under Pressure.

Typical hydraulic bicycle designs can be broken into two main sub-systems: the hydraulic circuit and the actual vehicle.

The most common hydraulic fluid design involves using a hydraulic pump and motor to transfer energy from the pedals to the drive wheel. The energy input to the system at the pump is transferred through hydraulic tubing, to either a pump or an accumulator. If transferred to the motor the bike is propelled forward. If transferred to the accumulator the relatively high pressure hydraulic fluid is stored in order to be harnessed at a later time. One of the main challenges of this design is that the majority of purchasable hydraulic motors/pumps are designed to operate efficiently at high rotational speeds. This means that teams have to find a way to increase the rotational speed of the pump input shaft. This is usually done by incorporating a gear train that has a large gear ratio.

While most teams utilize a hydraulic system with rotational power components, there is another option that a few teams have pursued in the past. This option incorporates hydraulic linear actuators to transfer energy from the pedals to the drive wheel. These actuators are essentially piston-cylinder assemblies that force hydraulic fluid in and out of the two cylinder bore chambers that are separated by the piston head. The rod ends of the actuators are eccentrically linked to the pedals and the drive wheel and pistons are set 90 degrees out of phase. Positioning the linear actuators and pistons in this way allows the harmonic linear motion of the actuator to transmit as rotational motion in a circular drive element.

Furthermore – in response to the efficiency event – Chainless Challenge bikes need to have a means of storing hydraulic energy. This usually involves a regenerative braking system that incorporates an accumulator as the main storage element. Charging and discharging an accumulator requires the hydraulic system to be much more complicated. Valves must be incorporated to allow for an accumulator charge and discharge setting. When the bike is stationary all of the power input at the pedals can be stored in the accumulator at a relatively high pressure. This energy can then be harnessed by directing the flow through the motor, thus propelling the bike forward. Alternatively, when the bike is decelerating the energy stored in the bicycles momentum is transferred to the motor and used to pump low pressure hydraulic fluid into the accumulator. Subsequently, this energy can be used to propel the rider forward.

Vehicles for the Chainless Challenge are typically much heavier than commercial bicycles. All previous Cal Poly vehicles have weighed more than one hundred pounds. Supporting the heavy hydraulic components requires a robust frame design; either a commercial frame must be modified or a custom frame built. Mounting plates, brackets, blocks, straps, et cetera have been used to mount the hydraulic system to the bicycle. Positioning the hydraulic power components also requires modification or custom frame design. If a pump and motor is used, the connection between them and the pedals and wheel are usually achieved by spur or bevel gears. For hydraulic actuators, a crank modified with a post or shaft is used. The rod end of the actuator is attached to the crank while the other is attached to the frame.

Both bicycles and tricycles have been used, each with their own advantages and disadvantages. The primary advantage of the tricycles, and, thus the disadvantage of the bicycle, is balance. Maintaining the balanced of a 100+ pound bicycle is difficult and tricycles inherently eliminates this problem. The advantage of the bicycles is more subtle; it forces the engineers to maintain a lower weight. Also, most bicycles place the rider in an upright position in which the rider can utilize his or her weight to pedal. On the other hand, many tricycles use a recumbent position, which eliminates this pedaling advantage. Some tricycles have used an upright design, however.

In addition to using a robust frame with mounting devices, there are several other considerations in current vehicle designs. Care is taken to minimize length of hydraulic lines and number of fittings to limit fluidic power losses. The hydraulic components should not impede the pedaling motion. The center of mass should mimic that of a commercial bicycle to maintain ease of handling. Overall, current vehicle designs attempt to accommodate the hydraulic systems with minimal change to current bicycle/tricycle geometry. Despite this, this “minimum of change” results in a much more complex and heavier vehicle compared to a standard commercial bicycle.

2.3. Previous Cal Poly Designs

Cal Poly has competed in the Chainless Challenge almost every year that the competition has occurred. A detailed table summarizing previous Cal Poly designs can be found in Appendix B. This appendix also includes pictures. Below are the summaries of key points and design features.

In 2008, Cal Poly designed a bike drivetrain which incorporated a custom made gear box. The gearbox was used to increase the rotational speed of the pump input shaft which enabled the pump to operate at a higher efficiency. A Rohloff internal gear hub was used at the rear wheel to allow for gear shifting and increased system efficiency. Also, a large accumulator was hauled in a trailer and was used to store energy for the sprint and efficiency events. The team used the accumulator for these events, and then detached the trailer for the endurance race. Cal Poly was able to store much more energy in their accumulator than the other teams because they used a higher volume accumulator.

The 2009 Cal Poly team chose to pursue the same basic design as the 2008 team, but with a few improvements. First, they switched from a gear pump to a more efficient miniature 5-piston hydraulic pump. The displacement of the piston pump was .156 cc/rev which is considerably smaller than the displacement of pumps that are typically used. They built a gearbox with a 3:1 gear ratio and coupled it to Harmonic Drive LLC planetary gear set, which had an 11:1 gear ratio. Therefore they were able to achieve a 33:1 gear ratio from the pedals to the pump input

shaft. This high ratio allowed the pump input shaft to rotate at over 2300 rpm, which is typically an efficient speed. The high gear ratio at the pump was compensated for by incorporating a motor with a relatively large displacement (11cc/rev). This difference in displacement, assuming constant flowrate through the system, created a 70:1 speed reduction. A Rohloff internal gear hub was used at the rear wheel to allow for gear shifting and increased system efficiency. The top speed of the bike was 7 mph which is half the speed of the 2008 bike. Unfortunately, this bike did not race because Parker-Hannifin cancelled the competition after the team of students had already started designing the bike for their senior project.

The competition was not held again until 2012. Parker adjusted the rules by putting a limit on the accumulator volume and by requiring the accumulator to be permanently attached to the bike.

In 2012, the Cal Poly team used two chain-sprocket speed increasers to accommodate the high volumetric flow rates that the pump efficiently operates at. The hydraulic system was supported by a rear manifold which raised the center of gravity of the bicycle. The team used a 1-liter piston accumulator to store energy for the relevant races. They also used a chain-sprocket speed reducer from the motor to the drive shaft which allowed for less torque to be input at the pedals. Subsequent to the competition, Parker judges decided to penalize the use of indirect energy transfer with chains and belts.

In 2013, Cal Poly's bike won the endurance race, and finished third for overall design. This design featured a 5.5:1 gear ratio in order to increase the speed of the shaft that powered the pump. They used a F11-5 pump and motor to power their bike, and used a flexible reservoir to decrease the chance of air making its way into the hydraulic circuit. The bike's frame was also modified in order to attach a smaller rear wheel. Using a smaller rear wheel decreased the translational speed of the bike while decreasing the input torque at the pedals. Using a smaller rear wheel allowed the motor to operate more efficiently when regeneratively braking.

In 2014, Cal Poly's team was unable to finish building their bike in time for the competition. This team abandoned the classical approach that utilized pumps and motors to power the bike. Instead, they used linear actuators linked to the pedals and drive wheel. The linear actuators at the pedals were attached to a small shaft that was offset from the bottom bracket's axis of rotation. This offset forced the piston to move linearly in the bore, thus pumping hydraulic fluid through the tubing linking the linear actuator at the pedals and linear actuator at the drive wheel. The linear actuator at the drive wheel was eccentrically fastened to a gear that was meshed with a gear linked to the drive wheel.

For the hydraulic system to continuously transmit power from the pedals to the rear wheel, both flow ports must be linked to the same flow ports on the other linear actuator. For example, the extend flow port on the linear actuator linked to the pedals needs to be hydraulically connected to the Extend Flow Port on the actuator linked to the drive wheel. By hydraulically linking both volumes of the actuator bore, a hydraulic circuit is created, despite the volumes being separated by the piston.

Adjoining the proper ports of the rear and front linear actuators allows the bike to ride much like a fixed gear bicycle. Hence, rotation of the drive wheel is directly linked to rotation of the pedals. When the rider is coasting and not applying torque to the pedals fluid is still being pumped in the actuators at the pedals, thus causing the pedals to rotate. The incorporation of a

freely rotating hub will prevent the hydraulic circuit from pumping fluid while the rear wheel rotates.

2.4. Codes and Standards

The most relevant codes and standards that we will follow during the course of our project are the ASTM pressure vessel standards. All of our pressure vessels, tubing, fittings and other components must abide by these codes.

Another safety guideline that we must adhere to is a maximum of 50 drops of leaking hydraulic fluid per minute at the competition. Any vehicle leaking more than this will be eliminated. Vehicles also must have multiple, independent brakes that are fully active and will bring the vehicle to a stop in case of an emergency. It is also necessary for our rider to wear a helmet carrying a CPSC sticker, and all potentially dangerous moving components on the vehicle must be covered with guards to protect the rider. Obviously, safety is an important issue for this competition, so any vehicle that is deemed unsafe will be disqualified, as well as any team exhibiting unsportsmanlike conduct.

2.5. Funding

This competition has always been funded by Parker-Hannifin, who provides us with a catalog of options for free hydraulic components which we may have shipped to us. Parker also provides a monetary budget to purchase additional parts. In the past, Parker-Hannifin has issued each team $1000 at the kickoff event held in September. An additional $2000 is issued at the midway review, depending on whether the team completed the required deliverables. A final $1000 is awarded to each team if it brings a functioning bike to the competition.

Fortunately, Cal Poly has accumulated extra money over the years from their prize winnings, so this year, we will have an even larger budget to work with. Currently, the Cal Poly Chainless Challenge account has around $10,000 in it for future teams to utilize.

3. Objectives (Design Requirements)

Our overall goal for this project is to design and build a bike that will win the Chainless Challenge competition while adhering to both Parker's rules and ASTM standards mentioned above. Our design objectives/specifications are directly influenced by Parker's competition specifications.

3.1. Competition Analysis

In order to win the competition, the bicycle must win the most competition points. Since this is the primary objective of Bike Under Pressure, it is important that the competition scoring is analyzed in order to guide design decisions. A rough analysis of scoring situations was completed in order to simulate potential winning performances. (Table is available in Appendix H). This has been compared to previous years winning scores and these estimations are realistic. Some lessons learned from the simulation:

- An estimated score of above 900-1000 will be required to win.
- The double weighting of racing/paper categories highlights the importance of performance results over design judging. That being said, ignoring the design judging is ill-advised. We can guess that good race results will carry over into good design judging scoring.
- Failing to complete any races would be fatal for the score. Reliability has high importance.
- Inclusion of regenerative braking has high importance/weighting.
- If no categories are won and no energy recovery is used, then a stellar performance in all categories - 2nd and 3rd only with more 2nd place - would be required to win
- Use of one chain is only acceptable if performance is stellar. Use of two or more chains is likely to be damaging to winning.

3.2. Design Requirements Table and Discussion

The engineering design requirements for Bike Under Pressure are summarized in Table 1 below. These design requirements took several considerations. First, the rules of the competition create hard requirements. Second, the scoring analysis helps emphasize key design areas. Third, previous competition race results provide a performance benchmark for the vehicle.

The requirement is the numerical goal for the given parameter. The tolerance describes whether the numerical goal is a minimum or maximum. The risk is categorized as low (L), medium (M), or high (H). Failure to meet a requirement with high risk will jeopardize successful operation of the bicycle at the competition. Failure to meet low risk requirements is less likely to fail the entire project. The compliance column summarizes how to decide if the final product has met the engineering requirement. The methods are analysis (A), test (T), similarity to existing designs (S), and inspection (I).

Table 1. Formal Engineering Requirements

Req.#	Parameter Description	Requirement or Target	Tolerance	Risk	Compliance
1	Top Speed	17 (mph)	Min	M	A, T, I
2	# of Total Components	50 excluding fasteners	Max	L	I
3	# of Custom Components	10	Max	M	I
4	Life	50 miles	Min	H	A, T
5	Production Cost	5000 ($)	Max	M	A, S
6	Safety	ASTM Standards	Min	L	A, T, I
7	Turning Radius	7.5 ft	Max	L	A, T, S, I
8	Overall Weight of Bike	225 (lb)	Max	M	A, T, I
9	200m Sprint Time	30 (sec)	Max	M	T
10	Average Speed (Time Trial)	9 (mph)	Min	M	T
11	Energy Recapture (Regen)	> 0 (%)	Min	M	A, T, S, I

Requirement 1 is determined by average the speeds of the top 3 finishers in the 2014 sprint race, which is the only competition that we have results from. This requirement is closely related to number 10. A similar method of aggregating and averaging past competition results has been used for requirements 6, 9, 10, and 11.

Requirements 2 and 3 provide a basic insight into the manufacturability, simplicity, reliability, production cost, and overall weight of the bicycle. Generally, using fewer components, especially fewer custom made components, increases the manufacturability, simplicity, and reliability of the bicycle while decreasing the cost and weight. Furthermore, fewer components will be conducive to great workmanship. Values of 50 and 10 were chosen to represent an amount that was feasible given our skill sets, resources, and time.

Requirement 4 sets our target design life for the bicycle. This came from a desire and need for high reliability. During the course of the competition, the bike races over 6 miles plus various sprints and demonstrations. In preparation for this use, we wanted to be able to ride 50 miles with no significant breakdowns or leaks. After the 50 miles of testing, we will find it acceptable to replace easily-changeable components.

Requirement 5 is based on the budget given by Parker-Hannifin and strives to keep production cost reasonable. While keeping cost low is a design goal and is important to us, we are willing to spend a little more money in order to achieve higher performance results.

Requirement 6 outlines the commitment to uphold all relevant safety standards such as ASTM pressure vessel standards. Furthermore, we want to have zero failures of frame members.

Requirement 7 comes from the slalom portion of the competition. For this event, cones is set up in a weave pattern, alternating right and left, 13 feet apart transversely, and 15 feet apart in depth (see figure 12). With this arrangement, the tightest turn radius is roughly half of the pitch, or 7.5 feet. It can be assumed due to observation of this year's competition that the tightest corners of the time trial course are less severe than the slalom corners.

Requirement 8 deals directly with the acceleration, maneuverability, and handling of the bicycle. The weight also affects the elegance of the design. A large problem with a hydraulic bicycle, besides the inefficiency, is the weight of the hydraulic components. Past designs have typically weighed two to five times that of a standard bicycle.

Requirements 9 and 10 serve as a general metric for overall performance as these are the actual performance requirements as set by the competition.

Requirement 11 stems from the Parker-supplied rulebook for the competition. A regenerative braking system is not required, but 200 points are granted to teams that choose to implement one. These points are very valuable when computing the overall design score for each team - contributing over 20% - so it would be unwise to abstain from including such a system in our design. Subsequent to gathering information about the competition requirements, we input engineering specifications into a Quality Function Deployment (Appendix C). These specifications were compared to competition requirements stipulated by Parker Hannifin.

4. Design Development

4.1. Overview

After completing background research, Biking Under Pressure had two primary tasks: to finish manufacturing the 2013-14 Cal Poly bicycle and to generate conceptual designs, simultaneously.

Finishing the previous Cal Poly design would serve as a proof of concept for the linear actuators. This proof was important as they are a relatively new design style, as covered in section 2.2. The results of this completed manufacturing and proof of concept are covered in section 4.1.4.

Initially the greatest concern for the basis of conceptual designs was picking between pump/motors, linear actuators, or some combination for the hydraulic design. The bicycle must be built to support the hydraulic circuit and to allow its operation. Each of these categories is different from the other to the point that the frame/vehicle must be significantly different.

After the rules were familiar and background research was conducted, the team brainstormed several configurations for each type of hydraulic system. These configurations were compared and several were chosen for further design analysis. In the end, one conceptual design for each of the hydraulic systems was designed to a full part level. In Sections 4.2 and 4.3, the conceptual design for a pump and motor system and a linear actuator system is detailed. The team then had to choose between these designs.

The following sections 4.2 through 4.4 cover the main aspects of the design development in further detail.

4.2. Discussion of QFD Results

A quality function deployment (QFD) was a useful tool for design development. It allows the design requirements to be correlated into designs. It also guides the weighting of design variables. While creating the QFD, there are several considerations: the importance of each design variable, how they affect the design requirement target and how potential designs are expected to perform based on analysis and background research.

The QFD created by Bike Under Pressure is in Appendix C. The customer requirements were weighted in several ways. First, on the left, by importance to all invested parties: Biking Under Pressure, Cal Poly Advisors, and Parker Hannifin Chainless Challenge. This weighting was completed by the Biking Under Pressure team with input from the Cal Poly Advisors. The official rules for the Chainless Challenge Event were consulted in lieu of Parker Hannifin's opinion. This analysis numerically assigned values for several prevailing ideas. The Biking Under Pressure team wanted a design that was easy to manufacture. The Cal Poly Advisors stressed reliability. The Parker Chainless Challenge rules emphasized safety, weight, and the inclusion of regenerative braking. All three parties also desired a strong performance in the competition races.

In the next step, the various system variables, such as weight, wheel base, and number of components, were listed and compared to each other. The triangle at the top shows the correlations of how the system requirements interact. For example, there is a strong correlation

between overall weight and number of components, but no correlation between the center of gravity height and the gear ratio. This allows the reader to quickly see which requirements have the greatest impact on the design.

Finally, the customer requirements were weighted in comparison with the system variables. For example, overall weight has a strong weighting with acceleration and handling on hills, but low correlation to manufacturing safety.

After the QFD was completed, the previous two Cal Poly designs were scored against the system variables using actual measurements and numbers with units where applicable. For example, the wheel base of the 2012-13 bicycle was 67 inches, while the 2013-14 bicycle was 50 inches. This allowed comparison of an implementation of each major design (linear actuator and pump/motor).

Several key lessons were learned from the creation of the QFD:

- Low complexity aided manufacturability, safety, and low weight
- Weight dramatically affects acceleration and handling
- Reliability is of utmost importance in order to successfully compete
- Fewer components increases reliability *and* manufacturability
- Linear actuator design is highly complex but possibly lower weight
- Pump and motor design is more reliable, depending on execution

4.3. Preliminary Analysis

In order to properly design the entire drivetrain an excel spreadsheet was formatted. It numerically models the torque, power, and speed, at different locations in the drivetrain. The spreadsheet helped Bike Under Pressure to understand what gear ratio was needed in the front and rear drive unit. It also helped to determine the displacement volume needed for the pump and motor. Knowing the gear ratio of the drive units and displacement volumes of hydraulic components the overall gear ratio of the bicycle can be determined. The displacement volume is important because it is directly related to flowrate, which is directly related to shaft speed. Hence a fluidic gear ratio is possible. The final design has a pump and motor with the same displacement, hence the hydraulic ratio is 1:1.

The spreadsheet contains input values that were gathered during the team’s research. For example, the power input by the rider was found to be roughly .5 hp, when pedaling at a cadence. Similarly, the cadence pedal speed is approximately 80 rpm.

Since, this is a preliminary analysis there are some physical parameters that were left out. Major and minor head losses in the hydraulic circuit were neglected: notice that the pump outlet and motor inlet pressures are the same. Air resistance was also not considered; this would only have an effect on the bike’s speed.

System Properties	
Pump Displacement	0.3
Motor Displacement	0.3
Pump Efficiency	0.85
Motor Efficiency	0.85
Diameter of Rear Wheel (in.)	28
Circumference of Rear Wheel (in.)	88.0
Gear Ratio	2.7

Pump	
Speed (rpm)	1200
Torque (lbf*in)	26.3
Flowrate (gpm)	1.5584
Outlet Pressure (psi)	467.4
Output Power (hp)	0.425
Gear Ratio Rear	15

Rider Properties	
Power Input by Rider (hp.)	0.50
Crank Speed (rpm)	80
Crank Torque (lbf*in.)	393.9

Motor	
Speed (rpm)	1200
Torque (lbf*in)	19.0
Flowrate (gpm)	1.5584
Inlet Pressure (psi)	467.4
Output Power (hp)	0.3613
Gear Ratio Rear	0.18

Rear Wheel Properties	
Wheel Speed (rpm)	216
Wheel Torque (lbf*in)	105.4
Bike Speed (mph)	17.99

Figure 1. Drivetrain spreadsheet for preliminary analysis

The spreadsheet shown in its current state includes the values for Bike Under Pressures final design; of course when it was first made an used different design values could be tested. The Parker F11-5 pump and motor that were chosen have a displacement of 5 cc/rev which is equivalent to 0.3 in^3/rev. One of the important outputs of this spreadsheet is the bike's speed. Bike Under Pressure needed to ensure that this speed was competitive. The speeds of the top competitors in the 2014 event were referenced when deciding what speed Bike Under Pressure needed to aim for.

The results from this analysis suggested that a high gear ratio (multiple revolutions at wheel for every one revolution of the cranks) was necessary to achieve a high top speed. Higher gear ratios resulted in higher top speeds as well as higher efficiencies at the pump and motor. However, this analysis only looks into steady state operation. It was known by the team (and any bicycle rider) that starting from a stop at a high gear is difficult. The rider is unable to produce the necessary torque with their weight. It was clear that a transient analysis was necessary. This provided the impetus for the system simulation, as covered in section 5.2.2.2.

4.4. Proof of Concept Testing Results

The 2014 Cal Poly Chainless Challenge team was unable to complete their bike, which incorporated a drivetrain with hydraulic linear actuators. In order to test the efficacy of this type of drivetrain, Bike Under Pressure decided to finish manufacturing the bike.

Figure 2. Cal Poly 2013-14 Operable Linear Actuator Bicycle

Towards the end of ME 428 the 2014 bike was completed. However, after riding the bike for only a short distance, an under designed component yielded. Hence, the bike was unable to be ridden any further. During the completion of the 2014 bike, Bike Under Pressure did not redesign any part of the drivetrain, this was the reason that the bike failed so quickly. The small shaft between the small crank and large crank, which links the linear actuator to the cranks, was not stiff enough to handle the loads input by the rider.

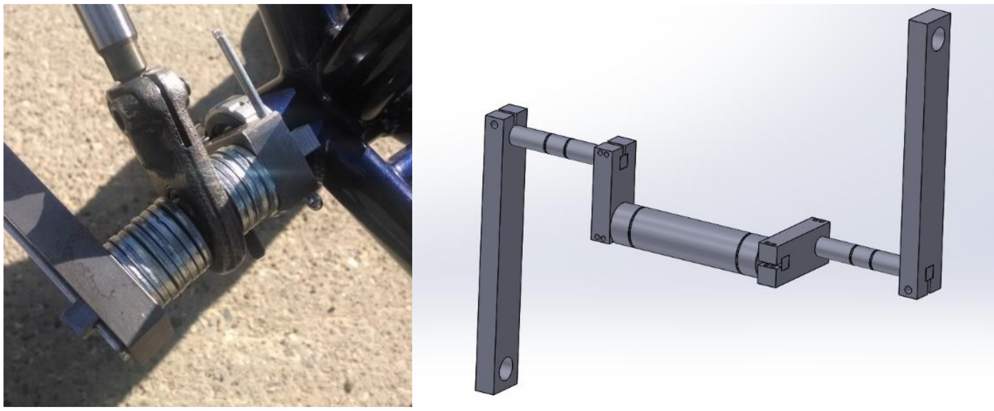


Figure 3. Deflected steel shaft that connects small crank and large crank (left), bottom bracket and crank assembly (right)

Subsequent to manufacturing the 2014 bike, Bike Under Pressure concluded a linear actuator drivetrain was feasible for riding in direct drive mode. However, incorporating regenerative braking capabilities was found to be much more challenging. Through researching and brainstorming, Bike Under Pressure discovered some ways that regenerative braking could be incorporated with a linear actuator drivetrain. One method, which was used by the Illinois Institute of Technology in the 2014 competition, is a roller which meshes with the rear wheel. This roller is coupled with a hydraulic pump. When the system is in direct drive mode, the roller is not meshed with the rear wheel. This roller and pump introduce an entirely separate hydraulic circuit that is in no way linked to the hydraulic circuit with the linear actuators. In order to use the linear actuator circuit for regenerative braking, precisely timed solenoid valves would need to be used. This is necessary because the pistons in the linear actuator, divide the system in two halves. Since, the pumping action is linear, timed valves are needed so that fluid can be pulled

from the lower pressure reservoir and pushed it into the high pressure accumulator. This is similar to how the poppet valves are used in an I.C. engine.

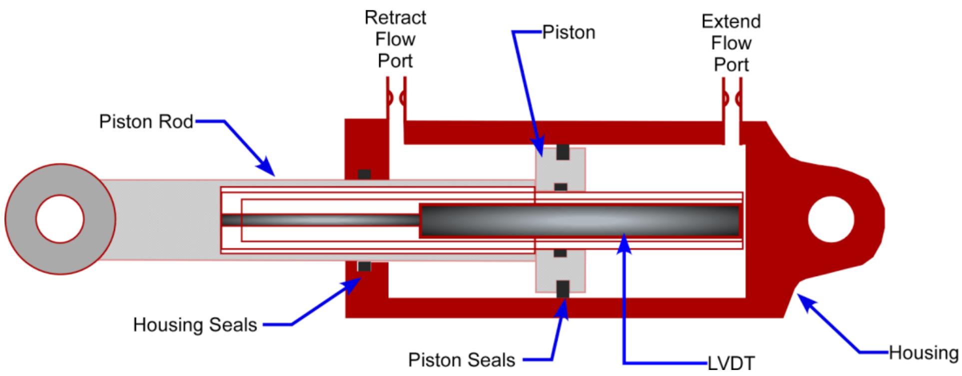


Figure 4. Schematic of a hydraulic linear actuator

The efficiency of the linear actuator drivetrain was not tested and linear actuator efficiency curves were unobtainable for high frequency, rotational, applications. However, linear efficiency curves, which compare the energy input by the fluid to the energy output by the rod end, were referenced. The linear efficiency of the Parker linear actuators used in the 2014 bike were roughly 90%.

Although Bike Under Pressure did not ultimately pursue a linear actuator drive train, completing the 2014 bike was a great learning experience for the team. Below is a list of some of the lessons that were learned.

- JIC swivel fittings make installation of the hydraulic circuit easier
- Bleeding a hydraulic circuit with linear actuators is challenging
- How to use conversational programming on a CNC Lathe
- The different hardware that can be used to for assembly, i.e. cotter pins and snap rings
- The importance of employing a good welder for fabricated components
- The difficulty of keeping geometric tolerances (concentricity, parallelism, perpendicularity)
- A linear actuator drivetrain is lighter than a pump and motor drivetrain for direct drive

4.5. Linear Actuator Conceptual Design

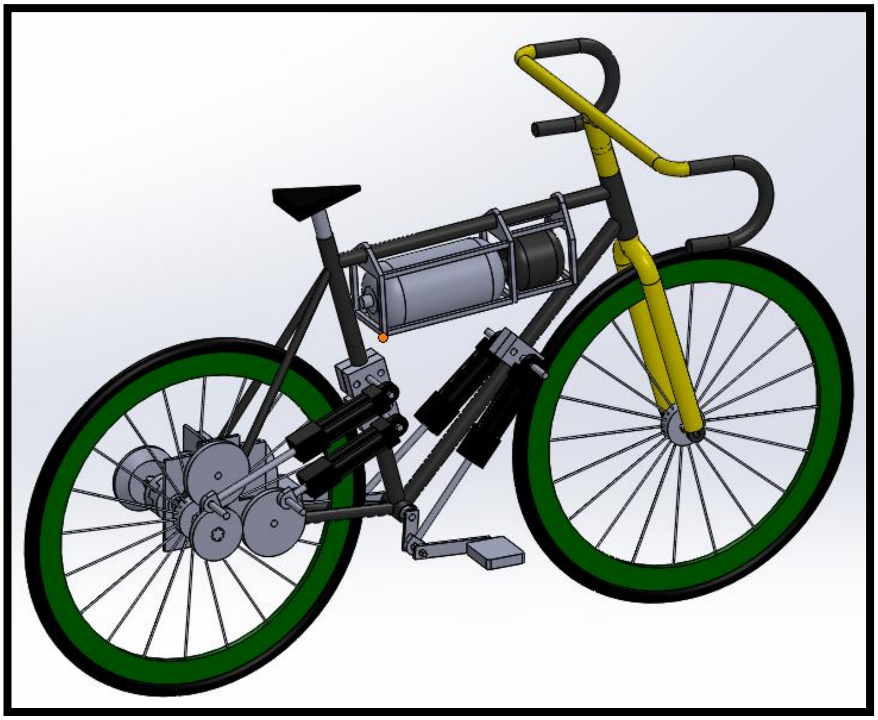


Figure 5. Linear actuator conceptual design model

The linear actuator conceptual design uses four linear actuators, two in the front and two in the back, which are linked as described in the design development to achieve rotary motion. The rear linear actuators are linked to gears which allow more torque at the rear wheel. Through use of a check valve system the accumulator could be charged; essentially the front linear actuators act as a piston pump. The accumulator discharge is then routed to the rear linear actuators. In order to achieve rotational motion, the fluid will need to be directed to the rear actuators in a controlled manner (e.g. the valves for discharge have an electronic control unit). This is a major impediment to the design feasibility as mentioned above in section 4.4.

Alternately, a hybrid system could be employed. Linked actuators would be used for direct drive, but not with the accumulator. Instead, a single hydraulic pump attached to the rear wheel would allow charging and discharging of the accumulator.

The model picture above has a rudimentary clutch included, which would allow coasting without causing fluidic losses. Co-ordination with the linear actuator design was only roughly modeled, and actual execution would take more design. This is another impediment to this design. Of course, the clutch isn’t required for successful operation, and could be left out.

4.6. Pump and Motor Conceptual Design

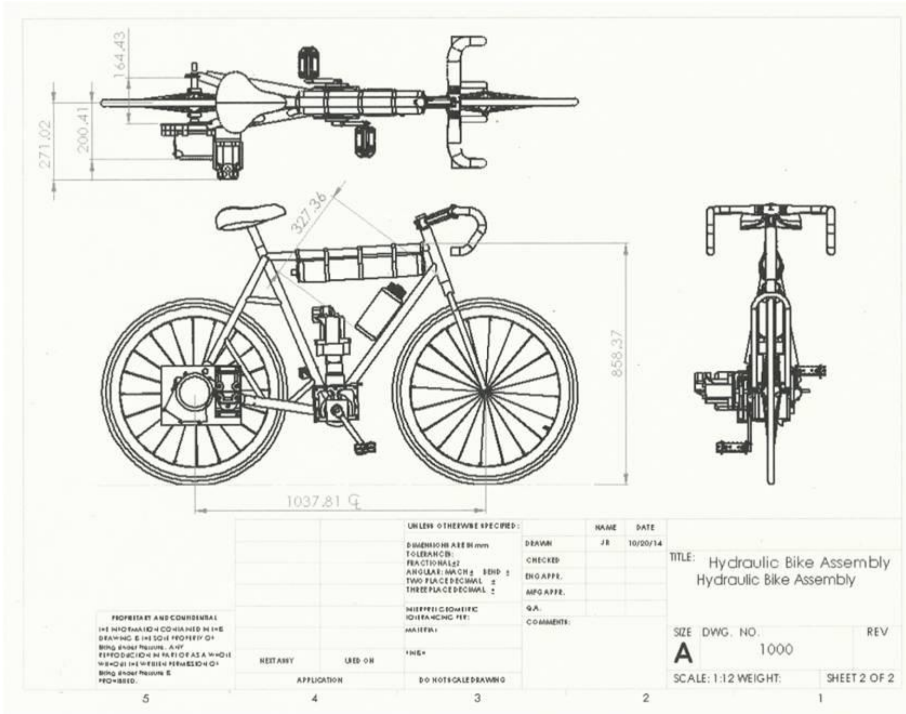


Figure 6. Pump and motor conceptual design assembly drawing

The pump and motor conceptual design features a mechanical gearing unit at both the pedals and at the back wheel. The Parker F11-5 hydraulic pump is used for both the pump and the motor. The efficiency curve for the F11-5 pump is shown below. The front drive unit acts a speed increaser for the pump in order to achieve efficiency greater than 75%. If maximum human pedaling speed is around 50 to 70 rpm, then a speed increase of 10:1 or more will be required. This is achieved with a beveled gearbox and a planetary gearset mounted on top. The pump couples to the planetary gearset with a mounting block and shaft coupler.

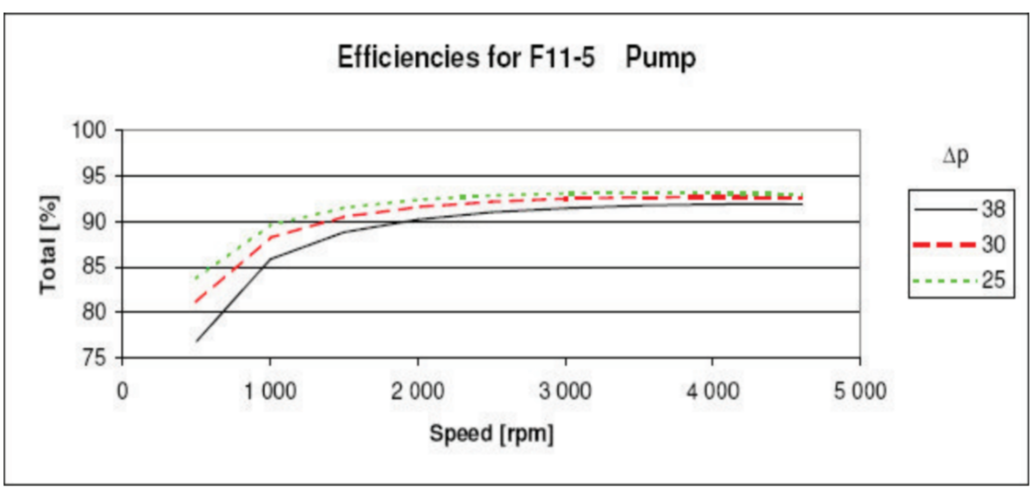


Figure 7. Efficiency curves for F11-5 bent axis piston pump (provided by Parker)

The hydraulic system has several operation modes. Direct drive mode links the pump and motor together directly and the displacements and rotational speeds are the same. Use of valves allows an accumulator to be charged by the pump and discharge to the motor. Regenerative braking is also possible through directing fluid moved by the motor to the accumulator.

The rear drive unit is a torque increaser. This allows the high speed at the motor to create useful torque at the wheel. This is achieved with a pinion attached to the output shaft of the motor and a gear attached to the rear wheel shaft and hub. The rear drive unit also includes a clutch – one side connects to the shaft and one side to the gear. Disengaging the clutch allows the bicycle to coast without moving fluid in the circuit. While pedaling, the clutch will remain engaged. A clutch, rather than a freewheel, is necessary due to the need to incorporate regenerative braking.

The frame uses standard bicycle geometry with several changes to accommodate the front and rear drive units as well as the hydraulic circuit. An L-shaped bracket allows the front drive unit to be bolted securely in place. A large plate and mounting block at the rear allows the clutch, motor, and gear to be mounted. The hydraulic circuit mostly holds itself in place due to the rigidity of the fitting.

4.7. Concept Selection

Once the two conceptual designs were elaborated, the team had to pick one option to pursue. The designs were compared in terms of the engineering requirements. The top speed and acceleration achievable was difficult to estimate, and it mostly depends on the mechanical gearing in the system. Since the gearing wasn't set in stone for either design, this was considered even. That being said, it was known that working gearing into the linear actuator geometry was more of a challenge.

Originally it was assumed that the linear actuator designs main advantage would be ridability. This assumption was based on two observations. First, linear actuators had the potential to line up human cycling cadence with the resistance of the system. The power stroke from the rider could be used during the extension of the actuator rod. This combats the tendency of the pump/motor system to be difficult to pedal because of human cycling cadence. Second, the linear actuator system was thought to be lighter, mostly because of the lower weight of the actuators (compared to the pumps). Lower weight increases ridability by handling easier and more like a regular chained bicycle. It would also improve acceleration. However, once the linear actuator conceptual design was complete, it became clear that it would not weigh less. In order to incorporate an accumulator a hybrid system including a pump (or electronic control) was needed. Proof of concept testing showed that coordinating bicycle geometry and rod actuation with human cycling cadence was more difficult than originally assumed. Further complicating the problem was the difficulty in getting the bicycle into motion at low speeds. Due to these problems, the assumption the ridability would be increased by use of linear actuators was inconclusive at best.

Lastly, the efficiency of the linear actuators was unknown. The actuation efficiency for rod extension – energy from actuation force ending up in fluid – is in the high 90% range according to various catalog information, but there was no data for fast repetitive, rotational motion. It was known that the actuation efficiency drops with higher linear actuation speeds, so it was assumed

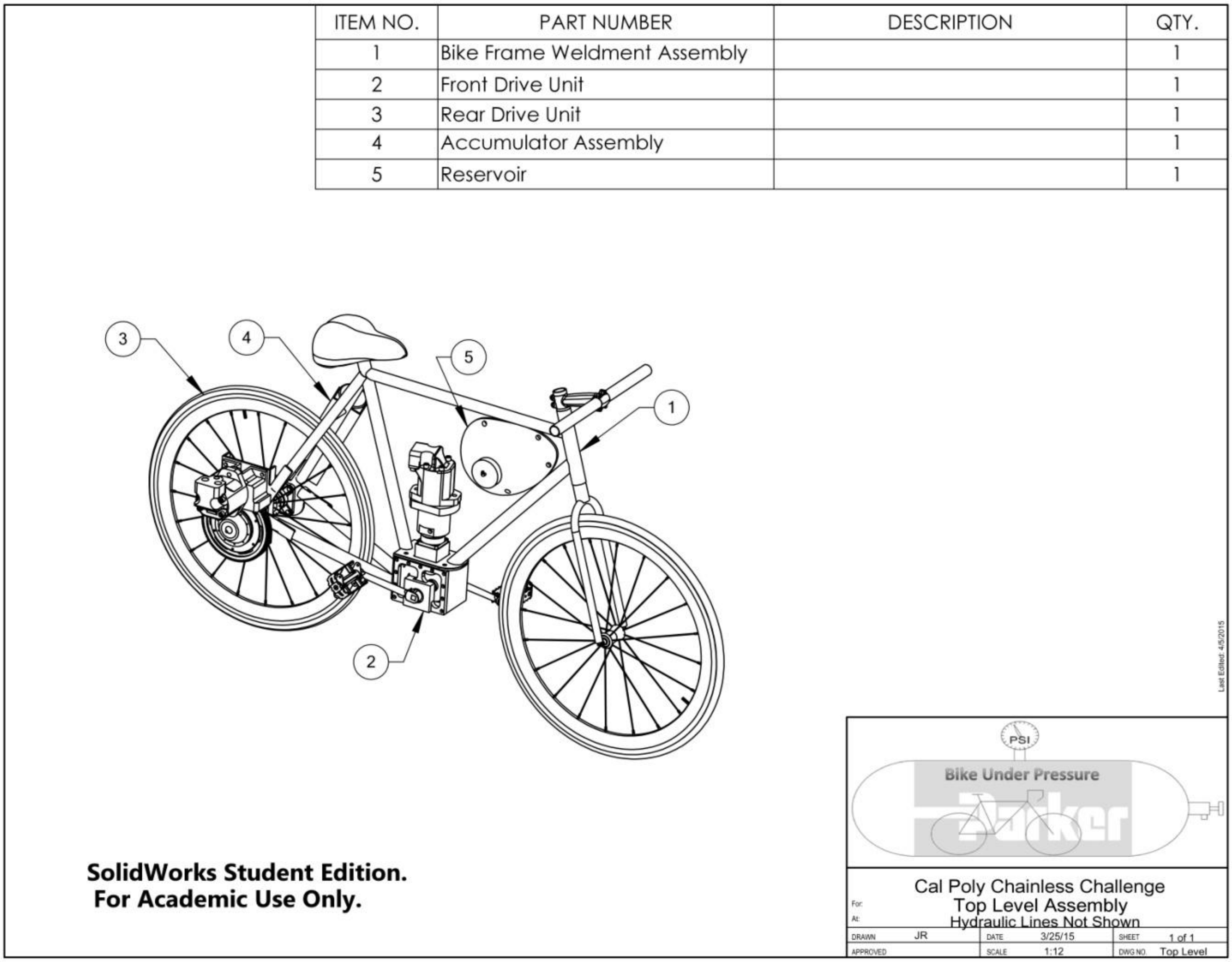
that it would also decrease with increased rotational speed. This was opposite of the hydraulic piston pumps, such as the F11-5. Thus, it was reasonably assumed that the pump/motor system would yield higher maximum riding speed efficiency.

In summary, the Bike Under Pressure team chose to pursue their pump/motor design due to its relatively lower complexity and higher potential efficiency without significant disadvantage in weight and ridability.

5. Description of the Final Design

5.1. Overall Description with Labeled Solid Model

The final design is a refined, iterated version of the pump and motor conceptual design. The main sub-systems are: the frame, the front drive unit, and rear drive unit.



ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	Bike Frame Weldment Assembly		1
2	Front Drive Unit		1
3	Rear Drive Unit		1
4	Accumulator Assembly		1
5	Reservoir		1

Figure 8. Top Level Assembly of Final Design

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	Parker F11-5 Hydraulic Motor		1
2	Planetary Coupler Housing		1
3	Pump to Gearset Coupler Inset		1
4	Pump to Gearset Coupler		1
5	5-1 Planetary Gearset		1
6	Pinion Gear Shaft		1
7	PInion		1
8	Gearbox to Planetary		1
9	Gear Box		1
10	bearing side		2
11	Gear		1
12	shaft		1
13	Crank		2
14	Pedal		2
15	Box Plate		1

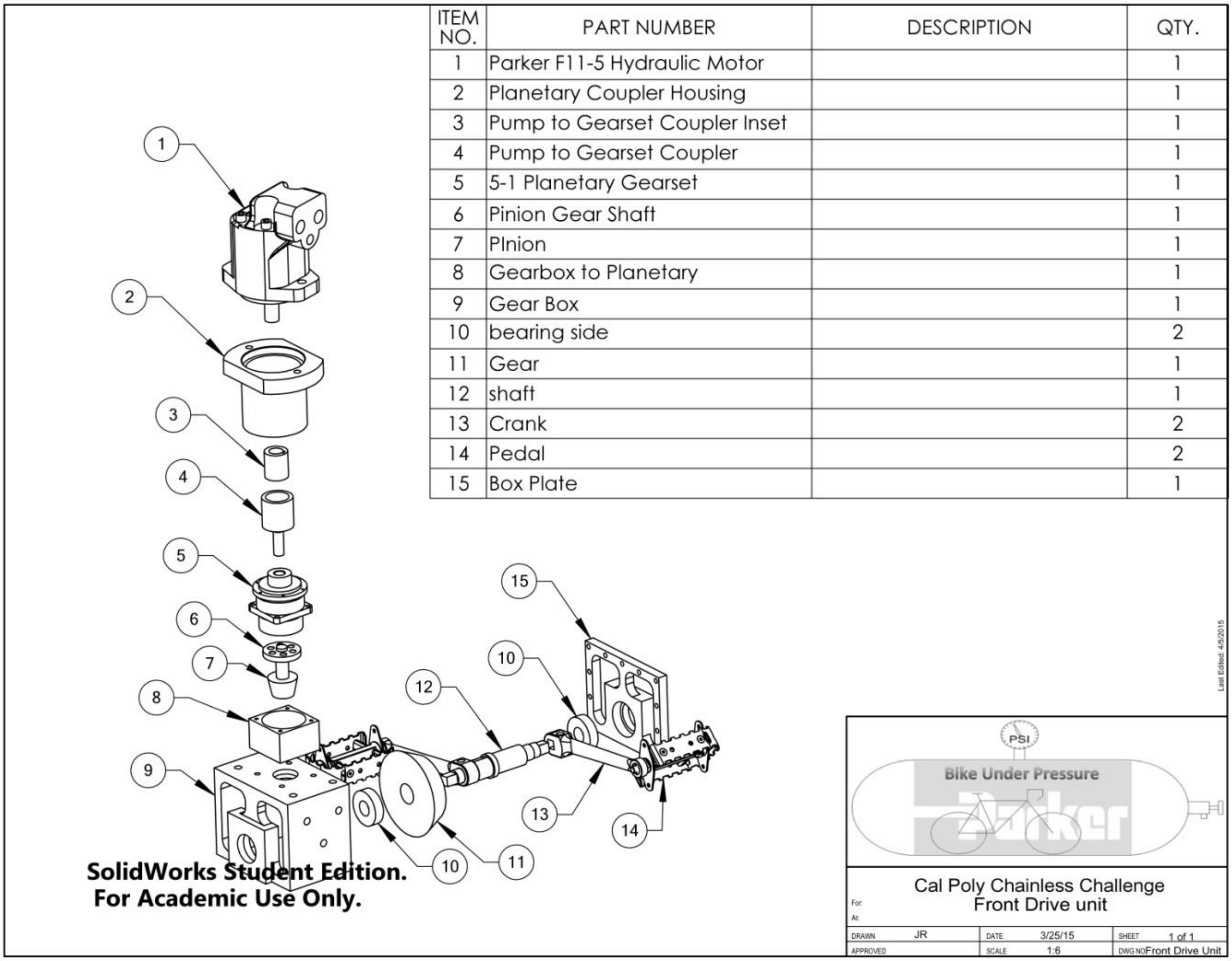


Figure 9. Exploded View of Front Drive Unit Assembly

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	DBR Clutch Assembly		1
2	Motor Mount Block 4		1
3	Parker F11-5 Hydraulic Motor		1
4	16 Tooth Pinion		1
5	Keyed Rear Shaft		1
6	Flanged Ball Bearing		2
7	Left Dropout		1
8	Right Dropout		1
9	Wheel		1
10	Rear Hub		1

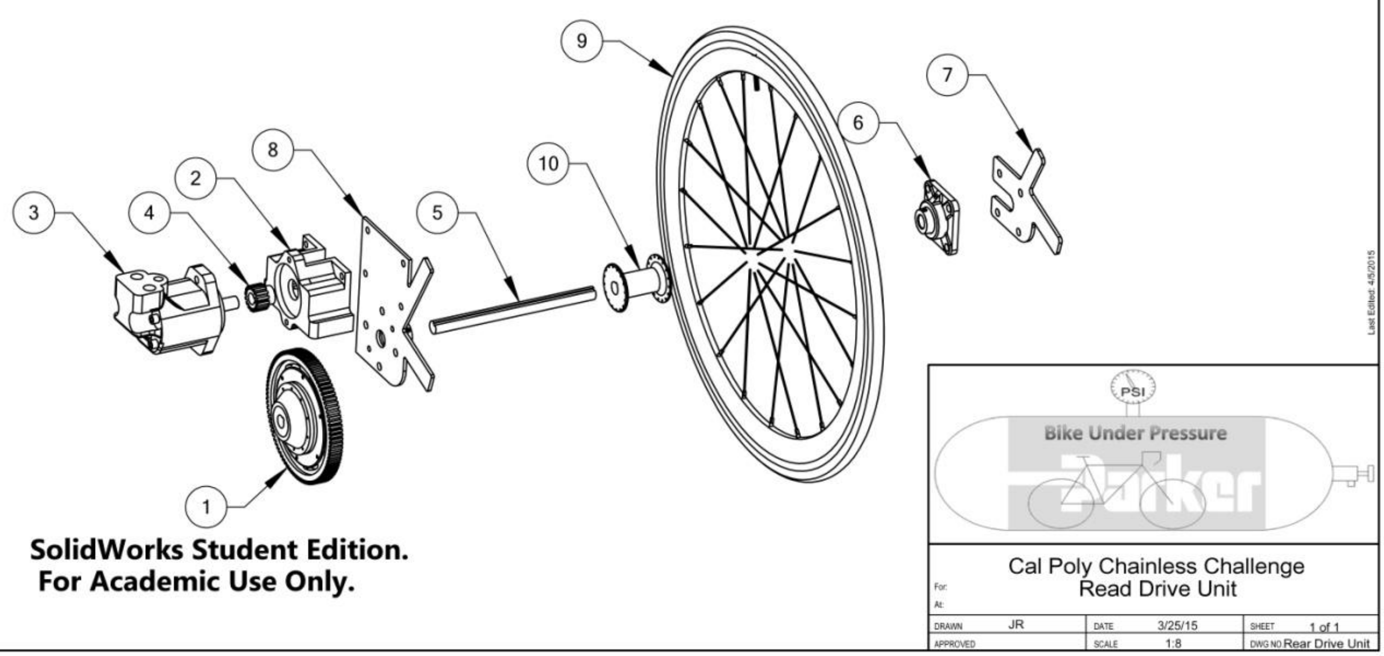


Figure 10. Exploded View of Rear Drive Unit Assembly

5.2. Detailed Design Description

5.2.1. Hydraulic Circuit Design

5.2.1.1. Circuit Operation Modes

The hydraulic circuit was designed for four primary modes or functions, with each tailored to a specific component of the competition. The first mode is direct drive as can be seen in Figure 4 below. This mode is to be primarily used in the time trail race as well to supplement accumulator assist in the sprint and relay events. The second mode is the regenerative braking and charging mode this can be seen in Figure 5 below. This mode can be used to transfer the bikes kinetic energy into captured potential energy through pressurizing our accumulator with fluid from our reservoir. This will be used to charge the accumulator in preparation for all of the races. The third mode is Accumulator Assist as seen Figure 6 below. This will be used to provide power to Bike Under Pressure during the Efficiency, Sprint and Relay Events as well as supplemental power during the Time Trial. The last mode is coasting as seen in Figure 7. This mode is unique to Bike Under Pressure due to the inclusion of a clutch in our design. This clutch allows for full disengagement of the Motor from the rear drive shaft ensuring maximum efficiency when coasting. Coasting will be used primarily in the efficiency challenge by increasing the length component of the efficiency challenge equation.

The fluidic power losses in the hoses and fittings were estimated to be 2 to 3% of the total input power. Details on this calculation are available in Appendix D.

The hydraulic fluid selected is Mobil EAL 224H fluid, which is biodegradable as per the rules. More information is available in Appendix E.

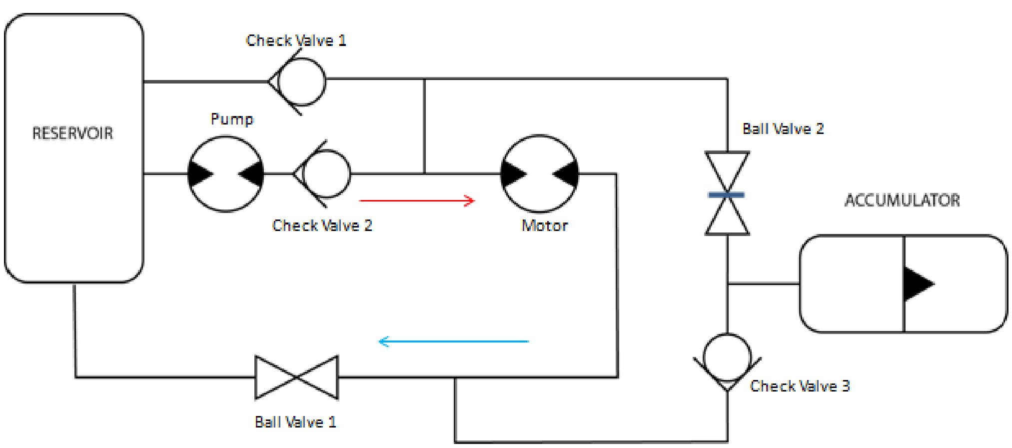


Figure 9. Direct drive configuration of the hydraulic circuit

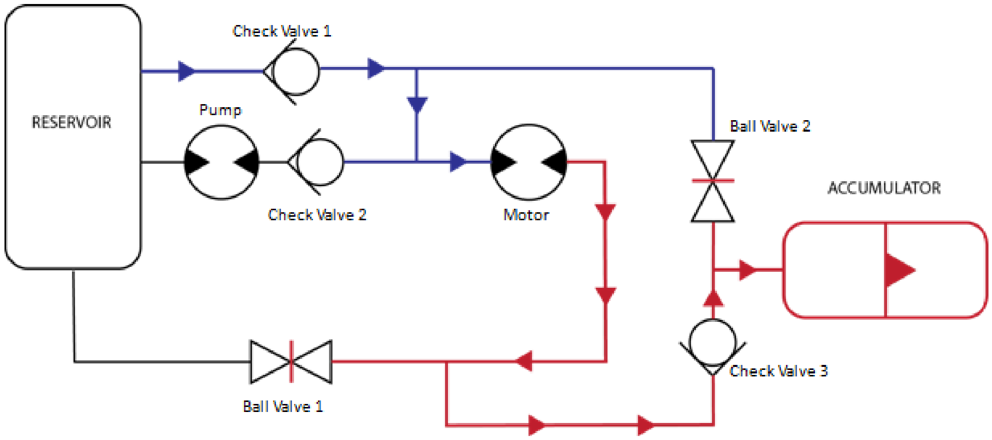


Figure 10. Charging and regenerative braking configuration of the hydraulic circuit

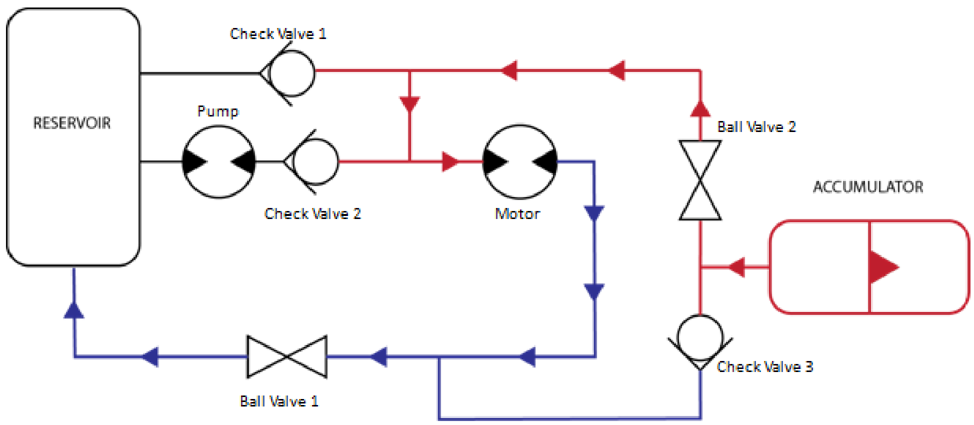


Figure 11. Accumulator assist configuration of the hydraulic circuit

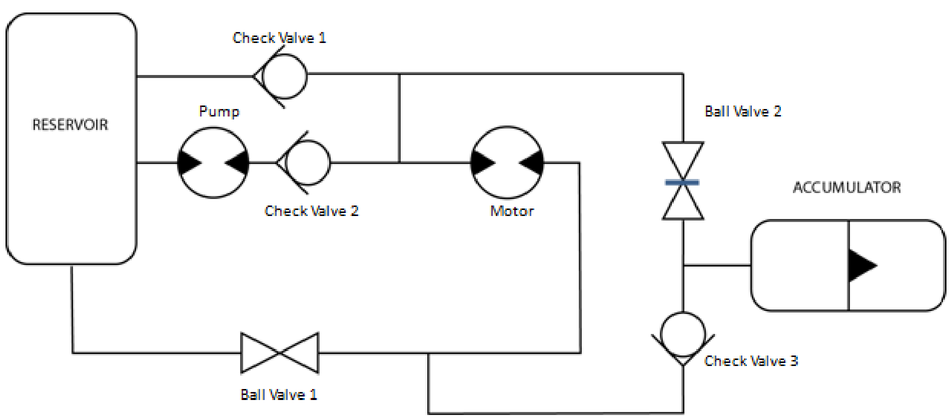


Figure 12. Coasting configuration of the hydraulic circuit

5.2.1.2. Accumulator

The accumulator used in the design is a 63 cubic inch Parker Hannifin piston accumulator. A piston accumulator style was chosen over a Bladder style accumulator due to its ease of mounting and generally smaller sizes. Furthermore, piston accumulators can handle much high gas compression ratios as well as higher flow rates than bladder accumulators. Both of these qualities are useful in the sprint event.

5.2.1.3. Reservoir

The reservoir used in our design is a 4 liter flexible MSR bladder. The benefits of using such a bladder are the ability to remove most of the air from the bladder which in turn lowers foaming and overall air in our hydraulic system. Furthermore the various mounting points on the bladder provided for a variety of installation options.

5.2.1.4. Valves, Hoses, and Fittings

The valves used in our hydraulic circuit consisted of three check valves, to separate high and low pressure components of the system and two manual ball valves to change which operation mode the system was in at a given time. The fittings used varied between the high pressure and low pressure components of the circuit. The system uses high pressure and low pressure hoses in their respective sides of the valves. The high pressure side consisted of ½” stainless steel JIC fittings. While the low pressure side of the circuit has ½” brass fittings.

5.2.1.5. Pump and Motor Selection

The pump and motor used are the Parker F11-5 CC/rev. The reason why this particular pump was chosen is because they are bent axis piston pumps. Bent axis piston pumps are preferred over gerotor, orbital, or vane pumps and motors because they are positive displacement with minimal leakage at starting rotational speeds. In order to achieve high efficiency at low pedaling speeds, a speed increaser was needed. This is described below in the front drive unit section.

5.2.2. Vehicle Design

The vehicle was designed around the hydraulic system and its requirement; the hydraulic design dictated the geometry and components of the vehicle. Of primary concern was determining the best mechanical gear ratios, choosing which equipment to use to achieve those ratios, and then to mount them to the frame and connect them to the hydraulics.

5.2.2.1. Note on Calculations

Many sizing calculations were performed during the design of the bicycle. Many were “back of the envelope” calculations for bolt sizing, clearance, fluid static pressure, volumetric analysis, or other common calculations. These calculations assumed a peak design load of a 200 pounds at the end of a 6 inch crank, that is, 1200 in-lbf torque input. For structural members of the vehicle, static loading of 1 G was assumed. Due to this low loading case a factor of safety generally design to be greater than 5. Several, such as sizing the various shafts and were calculated with the Engineering Equation Solver (EES) or MATLAB software. Commented EES code is

available in Appendix D for several important calculations.

5.2.2.2. System Simulation Results

In the conceptual design, the front and rear drive units had mechanical gear ratios in order to operate the pump efficiently and also to get useful power at the back wheel. The speed increase in the front drive unit and the speed decrease in the rear drive unit do not necessarily need to be the same. If they were, this would result in a system gear ratio of 1:1 in direct drive mode. For every full rotation of the cranks, the back wheel would turn once. A ratio of 2:1 means for every full rotation of the cranks, the back wheel would turn two full rotations. Many bicycles take advantage of a chain, derailleur, and sprocket cassette in order to change this system gear ratio. However, due design restraint on flexible mechanical elements gear changes require a full gearbox with a clutch. While the design incorporates a clutch, this is only to allow coasting, that is, a neutral gear. Another option was to use a Rohloff internal gear hub; however, this would not work with a clutch to coast system due to the internal freewheel.

The system as designed thus had to use a fixed gear. Choosing this gear was of utmost importance: too low of a ratio and the bicycle would have a low top speed, too high of a ratio and the bicycle would be difficult to pedal at low speeds. In order to choose the system gear ratio, a system simulation analysis was conducted. Details and assumptions are available in Appendix D. This simulation used a simple torque versus speed (power) profile for a cyclist in order to move the bicycle forward at the fixed gear ratio. The torque profile starts constant at low speed, and then decreases linearly after 70 rpm to zero around 130 rpm. This simulation was run at several gear ratios and the velocity and position output was used to compare them.

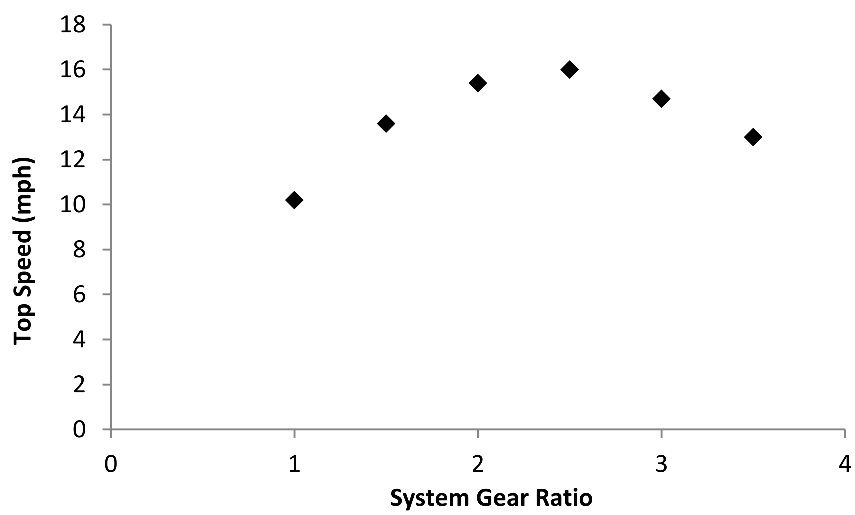


Figure 13. Simulated top speed at varied system gear ratio

The top speed occurs when the cyclist is at maximum leg rotation speed while still getting useful energy into the cranks. A higher system gear ratio allows a higher maximum power at that maximum speed. There are diminishing returns, however, because of the assumed torque profile with lower power output at higher rotational speeds. The top speeds at system gear ratios of 3 and 3.5 are inaccurate due to the simplistic model for aerodynamic drag. With better modeling, these results would probably be similar to the speed for 2.5. Maximum speed before diminishing

returns likely comes around 2.7 or 2.8 system gearing.

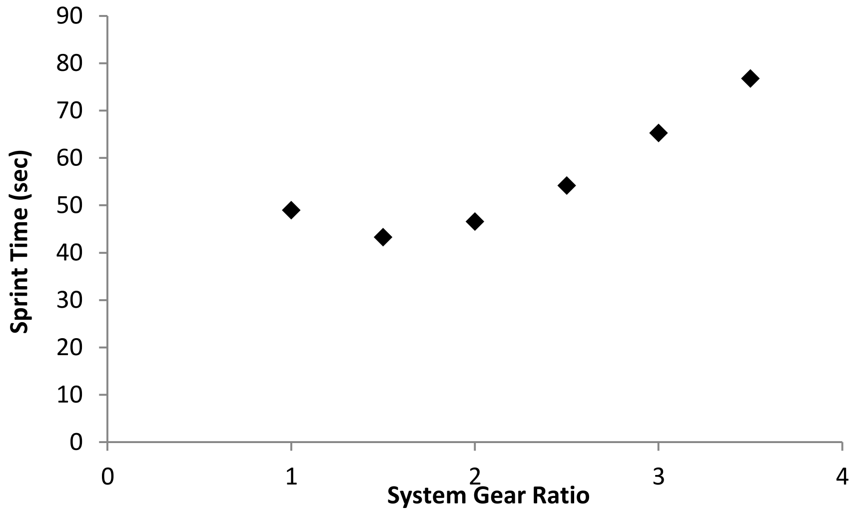


Figure 14. Sprint time at varied system gear ratios

The sprint time is calculated simply from time where the bicycle passes 200 meters, the distance of the sprint challenge. Low gears have a low top speed and thus perform poorly. High gears, while able to achieve a higher top speed, take a long time to accelerate and are slower in the race. The poor performance of higher gears is exaggerated in this model due to extremely poor predicted pump performance at low speeds. The increase in sprint time at high system gear ratios is likely less steep.

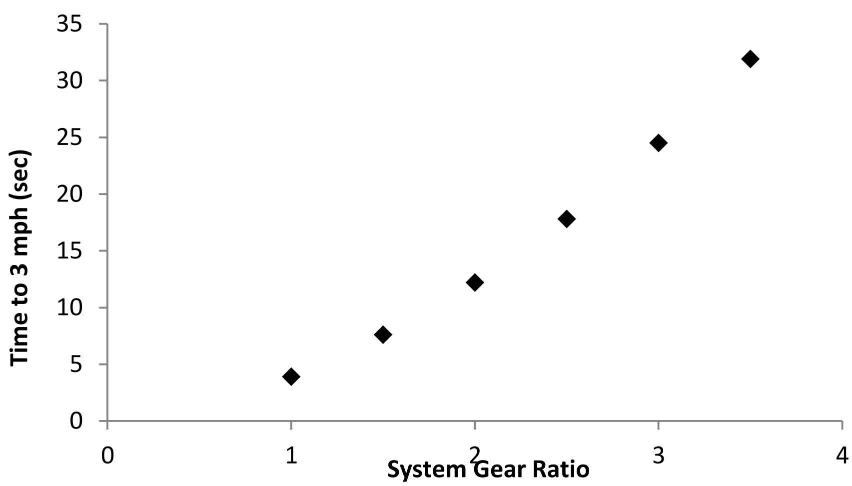


Figure 15. Time to 5 ft/s at varied system gear ratios

The time to 3 mph is a measure of how difficult the bicycle is to get started. A longer time to 3 mph means the rider has to put out more balancing effort while gaining speed. While this handling performance is difficult to compare numerically, this measure gives some idea of the relationship. The correlation, while taking into account the variable efficiency of the pump and motor, remains surprisingly linear. A higher system gear ratio will be more difficult to balance.

As with the sprint time predictions, these times are too long due to the poor predicted pump performance at low speeds.

While each of these correlations is intuitive for the average cyclist, it helps to have numerical values for comparison. In this case, we can see that a ratio of 2.5:1 to 3:1 will yield the highest top speed. This means time trial performance will be best. The sprint does not take accumulator assistance into account. While a system gear ratio of 1.5:1 would be best for the sprint in direct drive only mode, it would severely hurt top speed. The balancing time is gets quite long after a 2:1 system gear ratio. However, as mentioned, the model is weakest at low speeds. What the model can tell us is that the balancing difficulty linearly increases with increased system gear ratio.

Background research showed that fixed gear bicycles, such as beach cruisers, typically employ a 2.7:1 gear ratio, which is supported by these conclusions. Bike Under Pressure chose a system gear ratio of 2.7:1 after analyzing the results. This will result in a high top speed, a moderate sprint time *even without an accumulator boost*, and a moderately difficult start up balancing. However, it is hoped that the rider can push off with their foot enough to bring the bicycle up to a balanceable speed.

5.2.2.3. Frame Design

Figure 16. Custom frame with mounting L-bracket and rear dropouts

The frame of the bicycle is customized to accommodate the front and rear drive units as well as provide mounting points for the hydraulic lines and components. The team first had to choose between a bicycle and a tricycle. A bicycle has fewer components and likely a lower weight. These facets are consistent with the lessons learned from the quality function deployment, so a bicycle was chosen. Once this was decided, the next step was to determine the size and geometry of the frame. The manufacturing of the frame was outsourced to the Cal Poly Frame Builders, but Bike Under Pressure designed it.

Considering the predicted size and weight of all the mechanical and hydraulic drive train, Bike Under Pressure decided to use an oversized tube set, and also to switch the seat stays, chain stays and seat tube with thick-walled (0.049”) tubing. Furthermore, the material chosen is 4130 Chromoly steel, which has a minimum yield strength of 85 ksi, much stronger than plain low carbon steel. With this strength, the frame can support then extra weight of hydraulics. Both the dropouts and the front bracket are also made from 4130 steel for ease of welding.

Next, decisions needed to be made regarding the angles of the seat and head tubes. Most racing frames use more vertical seat and head tubes, which accommodates higher top speeds and acceleration. On the other hand, seat and head tubes with a more acute angle help the riders maintain control and balance. Bike Under pressure decided to set these tubes at angles in between the common racing and cruising frame designs.

The most apparent difference between a standard frame and Bike Under Pressure’s frame is the asymmetry of the rear dropouts and the front gear box mounting bracket. These features allowed for easy assembly and proper shaft alignment. Stiffening members were later added to one of the dropouts to prevent bending (details in Appendix D).

5.2.2.4. Front Drive Unit

The front drive unit is an assembly that converts mechanical power, input by the rider at the cranks, into fluid power. The fluid power flows from the Parker F-11 pump to the rear drive unit.

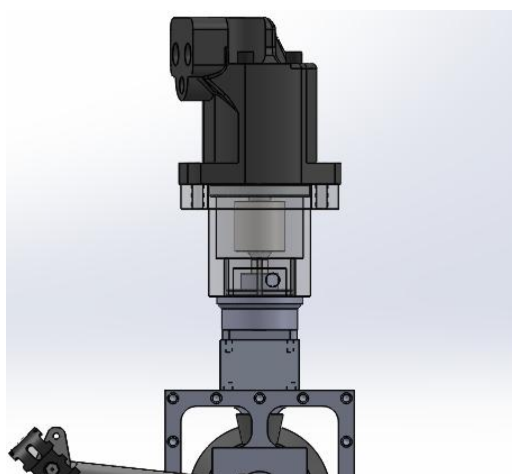


Figure 17. Front drive unit with transparent coupling housing to show shaft coupler

The hydraulic pumps that were considered during the design of the bicycle are most efficient at high speeds, relative to the cadence speed of a typical bicycle rider. In order to increase the speed of the pump’s input shaft a speed increasing gear ratio was used.

Towards the bottom of the front drive unit is the gear box. This component was reused from Cal Poly’s 2009 Chainless Challenge bike. The gearbox houses a 90 degree pinion and bevel gear. This helical bevel gear set is a 3:1 speed increaser. Bike Under Pressure chose to incorporate

this gearbox after researching the drive trains of previous Chainless Challenge vehicles. Using 90 degree bevel gears, allows for lateral power input at the cranks to be directed vertically. Directing the power vertically allows mechanical and hydraulic components to be placed inside the triangle of the frame without interfering with the rider's legs. This aluminum gearbox with steel bevel gears was salvaged because it was in good working condition and the bicycle's final design required us to manufacture several other custom parts. Thus, reusing the gearbox allowed us to focus on the manufacture and design of these other components. The gearbox was designed for 1400 in-lbf, greater than the Bike Under Pressure chosen design load of 1200 in-lbf. Sizing, stress, and bearing calculations for the gearbox are available in the 2009 Cal Poly Chainless Challenge Senior Project Report (Bedegi et al).

Above the gearbox is a planetary gear set. This planetary has a gear ratio of 5:1. Thus, the front drive unit's overall gear ratio is 15:1. This large gear ratio allows the hydraulic pump to operate at a more efficient speed, likely over 75% at top pedaling speed. See Figure 15 below (repeat of Figure 7), which shows the efficiency curves for different operating speeds of the F11-5 pump.

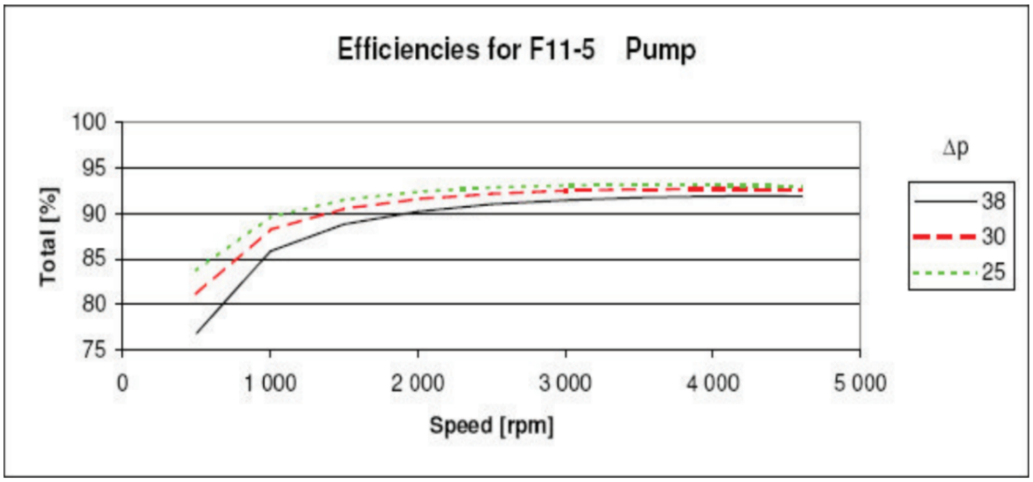


Figure 18. Efficiency curves for F11-5 bent axis piston pump (provided by Parker)

The planetary and gearbox pinion are mechanically linked with flanged coupler. The planetary is held in place by fastening to the square aluminum mount at the top of the gearbox. The square aluminum mount and flanged coupler were also a part of the 2009 front drive unit. Bike Under Pressure designed the rest of the unit around these components. The 2009 team used a planetary gear set with an 11:1 gear ratio. This gear ratio was too large for Bike Under Pressure's design, so a planetary with a 5:1 ratio, was purchased from the same manufacturer, Harmonic Drive LLC. The bolt pattern on the planetary that fastens to the aluminum mount and the bolt heads that mate to the flanged coupler, have the same pattern as the planetary used in 2009.

It was calculated that the bolts which hold the square aluminum mount to the gearbox will be strong enough to support the front drive unit should the bicycle tip over violently.

The output of the planetary is a shaft collar, which clamps to the pump/planetary coupler. The coupler links the planetary to the input shaft of the Parker F-11 pump. It is made from 1018 steel (for strength) and contains a press fit insert on the side that mates with the pump. The incorporation of a press fit insert was necessary, so that a keyway could be broached. The coupler has a factor of safety against yielding of 1.4. The press fit of .001" has a factor of safety

of 3.9 against slipping.

The pump is supported by a flanged aluminum coupler housing. This housing is made from 7075 high strength aluminum and has a flange for the pump and planetary gear set. This part was designed by Harmonic Drive LLC; however, Bike Under Pressure manufactured it to save on cost. The drawing from Harmonic Drive LLC is included

A Parker F11-5 cc/rev hydraulic bent axis piston pump is mounted to the top of the coupler housing. The pump mates with the coupler and is linked to the hydraulic circuit.

The entire front drive unit is fastens to the L-Bracket which is welded to the tubes of the frame.

5.2.2.5. Rear Drive Unit

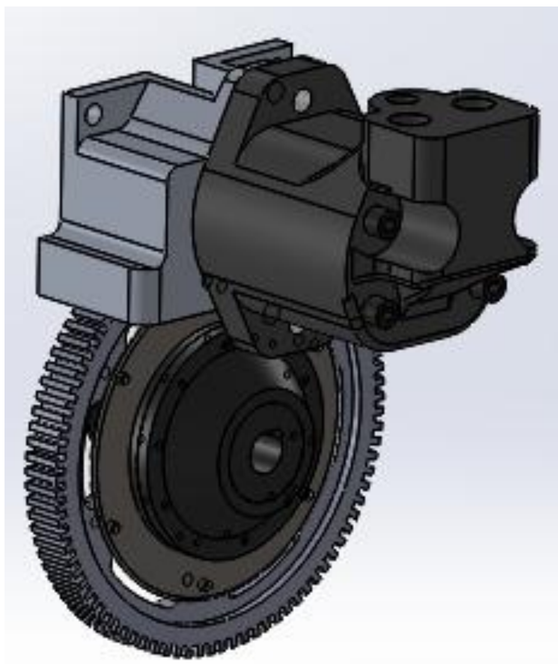


Figure 19. Rear drive unit

The rear drive unit accomplishes two main functions: have a speed reduction (torque increase) between the motor and rear wheel to obtain the desired overall gear ratio, and to include a means in which to disengage the rear wheel from the motor. In order to obtain an overall system gear ratio of 2.7:1 and with a front drive train gear ratio of 15:1 the rear drive ratio needed to be about 5.5:1. This ratio was obtained through a 90 tooth spur gear mated to a 16 tooth pinion. With this arrangement, the actual system gear ratio is 2.67:1. The gear and pinion will be steel for resistance to contact stress and tooth bending. Both gear and pinion are heavily modified so they could be attached to the clutch and pump, respectively.

Details on the gear size selection are available in Appendix E.

The second function of the rear drive assembly was accomplished by incorporating a clutch. The clutch is a DBR Conical Friction Clutch. More information about the clutch is available in Appendix B. This clutch disengages motion between the rear drive axle and the 90 tooth spur

gear while being mounted to the right rear dropout. In order to ensure high charging pressures the clutch springs were upgraded to stiffer Belleville Washers to increase the conical friction force. The rear drive assembly also included an aluminum motor mount which provided a rigid platform for the motor while also offsetting the pinion from the frame to ensure efficient tooth meshing between the 16 tooth pinion and 90 tooth spur gear.

The motor is supported by a 6061 aluminum mounting block. This mounting block, in turn, is bolted to the right rear dropout. Bearings on both rear dropouts support a shaft. This shaft has the rear wheel hub and the dynamic part of the clutch connected to it with keys and set screws. The shaft is 1045 medium tensile carbon steel that came with a pre-cut keyway. An early, rough analysis suggested that the minimum diameter for the shaft, given our geometry was 0.5 inches. The final shaft was chosen to be 0.75 inches. The static part of the clutch is also bolted to the right rear dropout.

5.2.2.6. Standard Bicycle Components

Although the design contains many custom components designed and modified to allow the efficient incorporation of hydraulics onto a bicycle some components are standard. These components include the front fork assembly, front and rear break assemblies, front tire, handle bar, hand grips, stem, seat, seat post, crank arms and pedals. The rear wheel although containing a standard rim and tire includes a modified hub to allow for mating with the rear drive axle. The bike frame was made from an oversized tube set with modified rear dropouts, upgraded seat and chain stays and lastly an L bracket replacing the standard bottom bracket.

5.3. Cost Analysis

A cost analysis was conducted in order to determine a rough estimate for the prototype and production costs of the bicycle. Detailed tables showing the estimated costs are in Appendix G.

The prototype costs were determined from the actual cost of components which Bike Under Pressure purchased, the value of the donated Parker components, and the labor costs. The labor cost was $60/hr. The labor was estimated from the actual amount of time the team spend manufacturing and assembling. It also includes the time spent iterating the design when the manufacturing did not go as planned. The labor component contributes the most to the total cost of the prototype. The total prototype cost is $15,071 (although the actual amount spent by Bike Under Pressure is closer to $4,600.)

In order to determine the cost of a high production version of the Bike under Pressure the overall cost of the bike was divided into two categories purchased components and custom components. The purchased components where split into four sub groups including frame, front and rear drive trains as well as the hydraulic system. These four groups were then broken up into their individual components whose prices were scaled to high volume costs. Furthermore assembly time for the four individual subgroups was determined and scaled once again based on high volume projections.

The custom components were broken into three groups these included the front and rear drive trains and the frame. All three groups were analyzed by comparing material cost, labor cost and equipment cost. The material costs included stock metals and standard components to be

modified. The labor cost was broken into assembly, manufacturing, setup and machining time. The equipment cost included tooling, and fixtures. The overall labor cost was calculated using a rate of 60 dollars per hour.

All these costs were summed for each custom part then added to the purchased components cost which resulted in an overall per unit cost for a 500 unit run of $3,798.

5.4. Special Safety Considerations

- When riding the bike a helmet should always be worn by the rider.
- If clips are worn while riding the bike, the clips should be loose enough so that the rider can easily detach and put his/her feet on the ground.
- The bike does not have a kick stand and the center of gravity is not in the center of the bike. Therefore, when the bike is stationary it must be well supported.
- Brake pad engagement should always be checked prior to riding.
- When the accumulator is discharged the ball valve should be gradually opened. This will prevent the rider from accelerating too abruptly. Fluid hammer will also be avoided.
- When gears are rotating do not put fingers near them.
- The bike should not be ridden alone
- There right rear dropout stiffening members have moderately sharp edges. Be careful when working around them to avoid any scrapes

5.5. Repair and Maintenance Considerations

Clutch: If the clutch is assembled and disassembled regularly it should be done in a clean environment. The mechanical components, i.e. thrust bearings, washers, and shims should be well greased before operation.

Rear Tire: The weight of the bike is mostly towards the rear. If the bike is mounted on the stand such that the weight of the bike is supported by the rear wheel, the tire will slowly deflate.

Set Screws: The set screws located on the rear drive shaft should be occasionally inspected to ensure that they haven't become loose.

Tire Pressure: The front tire should be pressurized to approximately 85 psi while the back tire should be pressurized to 110 psi. The tires should be checked regularly to ensure that the pressure is maintained close to these values.

Leaks: The hydraulic circuit should be regularly checked for leaks. This can be done by inspecting the ground where the vehicle has been parked or by checking the fittings for dampness.

Brakes: As the brake pads wear the barrel adjusters can be altered to change the distance that the lever needs to be pulled.

6. Manufacturing and Assembly

6.1. Part Manufacturing and Modifications

Gearbox: This assembly was salvaged from the 2009 bike; however, Bike under Pressure had to drill a new bolt pattern on the side that mounts to the L-Backet. This needed to be done because the preexisting bolt pattern did not work with the design of our chainstays.

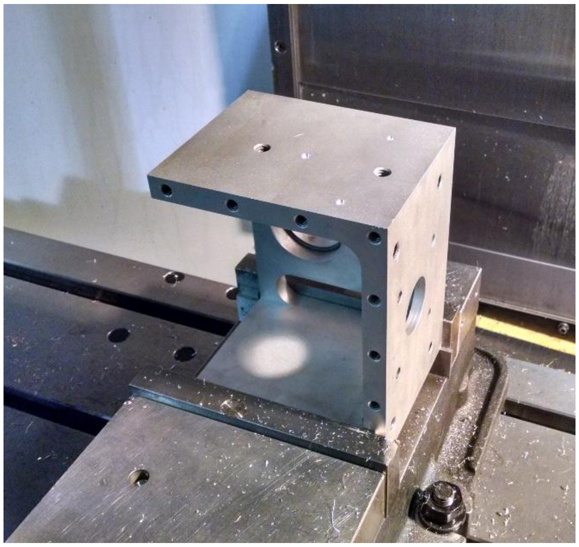


Figure 20. Drilling modified bolt pattern in reused gearbox

Pump/Planetary Coupler: The coupler was made to transfer power from the shaft collar in the planetary gearset to the keyed input shaft of the Parker F-11 pump. The coupler consists of two parts. The shaft mates with the output of the planetary gear set and the keyed blind hole mates with the input shaft of the pump. The part that mates with the pump is a press fit insert with a keyway in it. The pump/planetary coupler was designed this way because a through hole was necessary in order to broach the keyway. Therefore the coupler consists of two parts, one part has a shaft and a bored hole. The other part is the press fit insert with a keyway.

Both parts of the coupler were made on a Hass TL1 CNC lathe. The stock was 2” diameter, 1018 steel. A CAM software was not used for this part because the controller of the CNC lathe has conversational programming capabilities. Hence, these parts were made by entering the desired values of the part’s features, into the controller.

Two setups were required to manufacture the part of the coupler with the shaft. In order to maintain concentricity, while changing the setup, a dial indicator was used. The keyway in the press fit insert was broached using an arbor press.

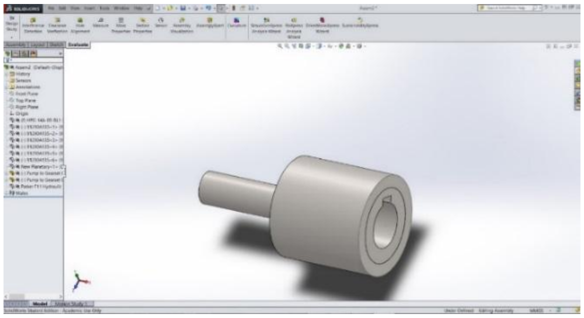


Figure 21. SolidWorks model of coupler

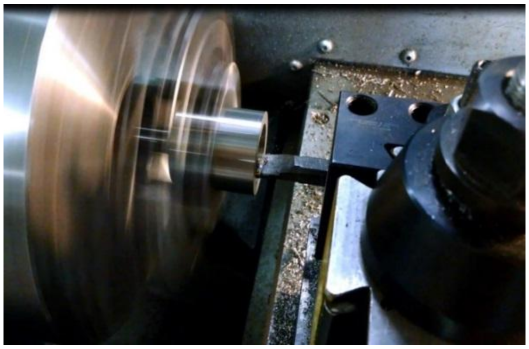


Figure 22. Boring the hole for the insert to be pressed

Figure 23. Coupler before being broached and press fit (right) and press fit insert (left)

Coupler Housing: The housing was designed by Harmonic Drive LLC, the company that the planetary gear set was purchased from. It has a flange for the pump and a flange for the planetary gear set. Harmonic Drive wanted to charge $800 for them to manufacture the coupler housing. Bike under Pressure decided to manufacture this part to save on cost. Fortunately, Harmonic sent us the SolidWorks model.

The adapter was machined on a Haas VF2 3-axis CNC mill. It would have been best to machine the part on a lathe, however, Bike under Pressure did not have access to a lathe with a chuck large enough to accommodate the work piece. The work piece was a 5.5” diameter, 6” long, billet of 7075 aluminum. The toolpaths were programmed using PTC Creo.

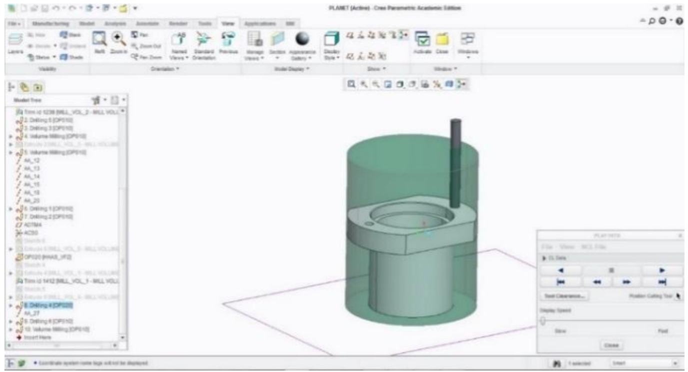


Figure 24. Toolpath simulation in PTC Creo

In order to fixture the cylindrical stock, aluminum soft jaws were manufactured. A guide hole was drilled through the stock during the first operation. In the second setup, the axis of this hole was found using a co-axial dial indicator. This was done so the features machined during the first operation would be concentric with the features machined during the second operation

The part was designed to have two pilot holes one for the planetary gear set on the bottom and one for the pump on the top. When the entire front drive unit was assembled it was found that the top pilot hole was not concentric with the lower pilot hole. This eccentricity was likely caused by the inaccuracies of the fixture (soft jaws). Material was removed from the top pilot hole in order to allow the pump's pilot boss to be inserted into the housing. This modification was done using a CNC lathe. The bolt pattern was also drilled again on a CNC mill. To measure that the top and bottom flange faces were parallel a dial indicator and a MICROFLAT were used. Parallelism of the flange faces was achieved by facing off material on the CNC lathe.

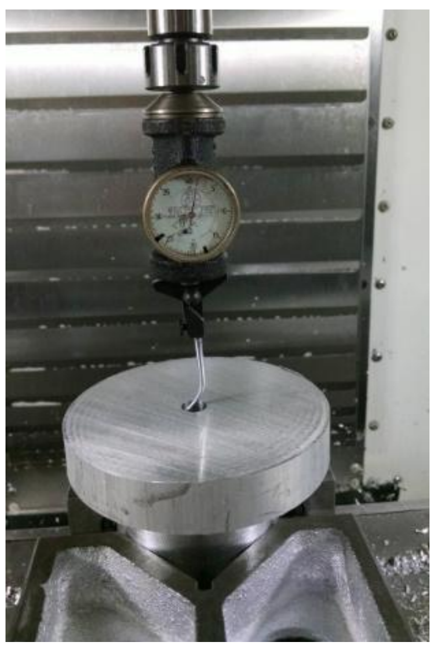


Figure 25. Origin being located for second setup by using a coaxial dial indicator

Figure 26. Coupler housing in between first and second operations

Motor Mount: The motor mount is an aluminum part that fastens to the right rear dropout. It supports the motor and houses the rear pinion. It was machined from a 3”x3”x6” billet of 6061 aluminum. The material removal was done using a Haas VF2 3-axis CNC mill. Three different setups were required in order to machine this part. In all three setups the work piece was fixed in a vise. The tool paths were programmed using PTC Creo.

Manufacturing of this part went surprisingly well considering the complexity of the part’s features. A guide hole was drilled during the first operation. The center of this hole was found using a co-axial dial indicator during the second setup. This center was used as the X and Y work offsets for the second operation.

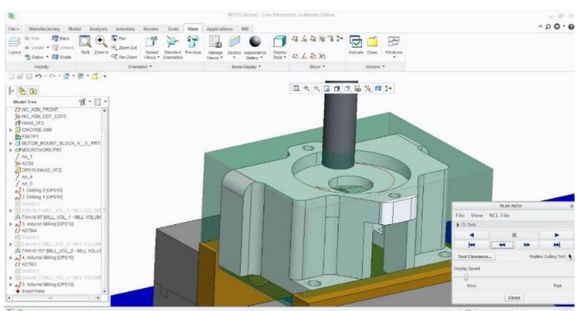


Figure 27. Side of motor mount toolpath simulation in PTC Creo

Figure 28. Motor mount after first operation. Notice the guide hole in the center of the part.

Figure 29. Motor mount during machining

Clutch/Gear Adapter Plate: The gear was originally designed to be a solid steel gear. Since there were none available a spoked, cast iron gear was used instead. In order to mount this to the clutch, an adapter plate had to be used. The gear calculations were redone for cast iron, and it should be safe against tooth bending and contact stress. Because the steel pinion is harder than the cast iron, wear will be increased. It should not be significant for the design life of the bicycle.

The clutch/gear adapter plate connects the rear gear to the clutch. The clutch's flange bolt pattern, which consists of ten 10-24 botls, is used to fasten the adapter plate to the clutch. The bolt pattern in the plate is countersunk in order to accurately locate the plate relative to the clutch. The outer bolt pattern consists of six ¼-20 bolts. The bolt pattern mates the gear and the plate.

The gear originaly had a hub, which was removed by turning it on a CNC lathe. Both sides of the gear were faced. The two bolt patterns were drilled using a Hass VF2 3-axis CNC mill. The plate and gear were fixtured to an aluminum plate which fastened to the T-slots in the table. Two dowel pins were incorporated into the fixture, to locate the spoke pattern relative to the machine. The gear and plate were fastened to the fixture using 10-24 bolts in the outer bolt pattern. By fastening to the fixture it was possible to bore out the center of the gear and plate. This material needed to be removed so that the plate and gear could fit over the sprocket of the clutch.

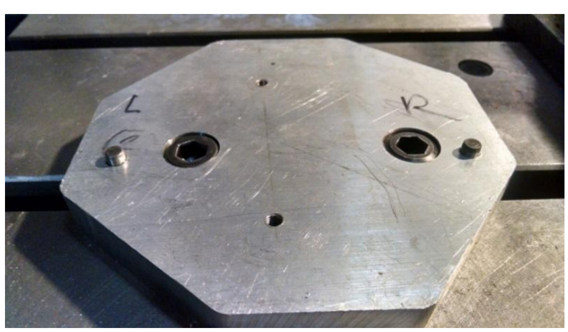


Figure 30. Aluminum sacrificial fixture plate with locating dowel pins

Figure 31. Tapping holes in aluminum fixture plate to fasten plate and gear to it

Figure 32. Finished adapter plate and gear

Figure 33. Clutch/Gear assembly being tested for concentricity with a CMM. The center of the drive hole in the clutch and the center of the gear, were concentric within .0045"

Rear Dropouts: The rear dropouts of the bicycle are plates that weld to the frame and support our drive components. The dropouts were manufactured on a Haas VF3 3-axis CNC mill. Each dropout was machined from .190”x12”x12” 4130 steel plate.

The bolt patterns in the dropouts were drilled while they were toe clamped to a sacrifical aluminum fixture plate. These same bolt batterns were drilled and tapped in a flat steel fixture plate. In order to mill the profiles of the dropouts, the workpiece was fastened to the steel fixture

plate. The outer countor was machined in one pass with a 3/8” roughing end mill, at a very low feed rate.

After the dropouts were welded to the frame and the components were bolted to the plate, it was decided that stiffening members should be welded to the right dropout. This was done to reduce possible deflection. The stiffening members were manufactured by shearing strips of .190” thick, 4130 steel plate.

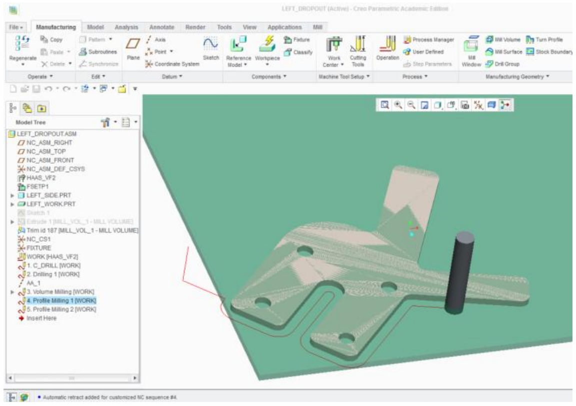


Figure 34. Toolpath simulation for left rear dropout

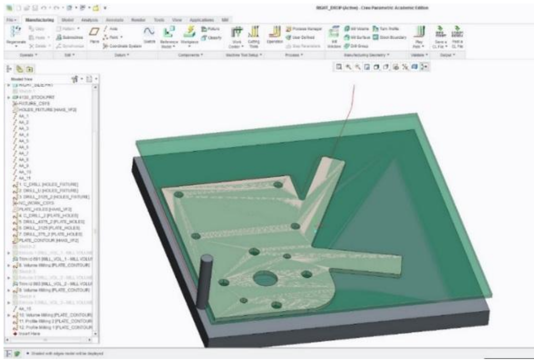


Figure 35. Toolpath simulation of right rear dropout

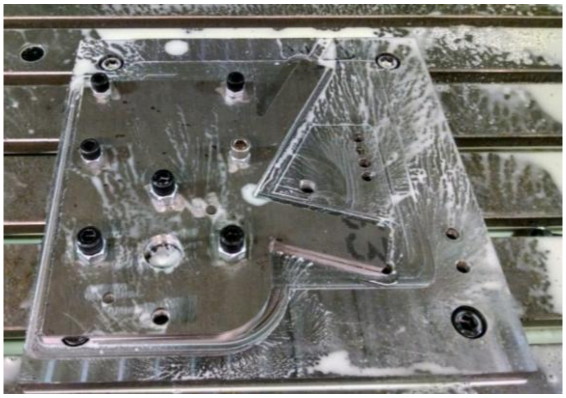


Figure 36. Finished right rear dropout, while still fixed to the steel fixture plate

Figure 37. Stiffening members on both sides of the right rear dropout

Front L-Bracket: The two parts that make up the L-bracket were machined on a Haas VF3 3-axis CNC mill. They were manufactured from a 3” x 6” x .190” 4130 steel plate. To fixture the part a machinist’s vice was used. The two parts were welded together and then welded to the tubes of the frame.

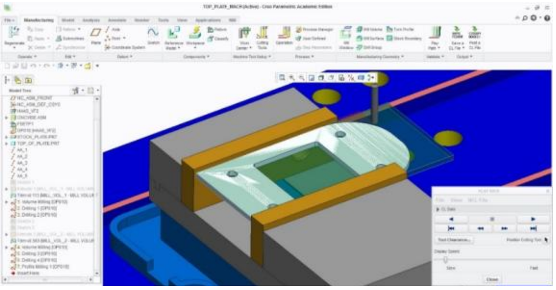


Figure 38. Toolpath simulation top of L-Bracket PTC Creo

Figure 39. Top part of the L-Bracket

Hub: The hub on the rear wheel neeeded to be customed manufactured in order to accommodate the 3/4”drive shaft. A keyed drive shaft was used in order to accommodate the 3/16” keyway in the clutch. The hub’s body was manufactured using a CNC lathe and the spoke hole pattern was drilled using a CNC mill. The tool paths for machining this part were created using conversational programming.

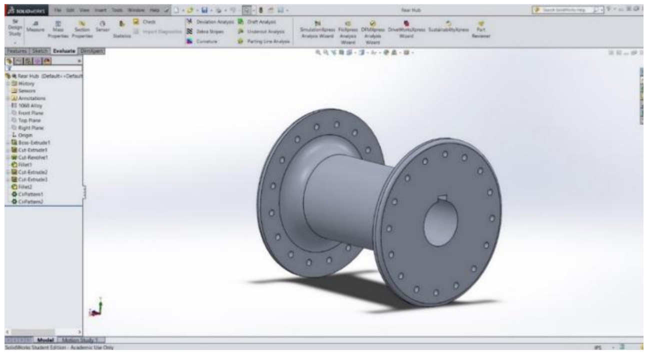


Figure 40. Rear wheel hub in SolidWorks

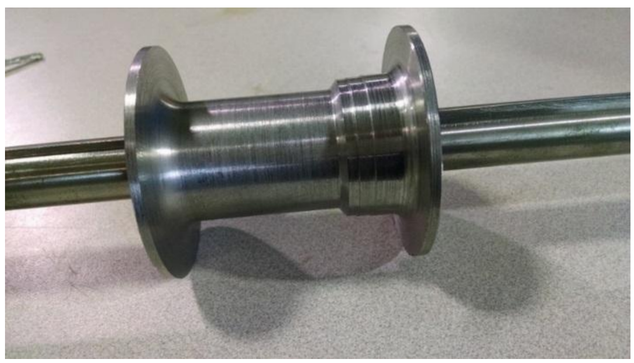


Figure 41. Partially complete rear wheel hub

Figure 42. Hub during manufacturing

Rear Pinion: The pinion that is attached to the output shaft of our hydraulic motor was modified from its original condition. The original hole in the pinion was bored from 5/8” to 18 mm (.71”). A 6mm x 6mm keyway was also borached in the pinion. A setscrew hole was not drilled and tapped in the hub, because after boring and broaching there was not enough material left on the hub. The pinion is axially fixed by a bolt and washer, which are fastened into the tapped hole at the end of the motor ouput shaft.

Brake Bridge: The brake bridge was manufactured from 4130 ½” diameter, .049” thick tubing.

The tubing was rougly cut to the desired length and then the ends were mitered in order for them to flushly weld to the seat stays. A 5/16” diameter hole was drilled through in the center of the bridge in order to support the brake calipers.

Rear Shaft Spacers: Spacers were manufactured in order to ensure that hub was axially fixed on the rear shaft. Spacers were made from .75” ID .125” thick aluminum tubing.

6.2. Assembly and Modifications

Front Drive Unit: The gearbox was found in the Chainless Challenge cage in its fully assembled state. However, it was disassembled in order for Bike under Pressure to gain familiarity with the internal parts. While the gearbox was disassembled the components were checked for damages and the tapered roller bearings in the side of the gear box were regreased.

The 4 bolt flange in the middle of the planetary is bolted to the aluminum adpater at the top of he gearbox. The six bolt flange at the top of the planetary gear set is fastened to the coupler housing. The orientation of the coupler housing’s upper bolt pattern is critical to ensure that the pump’s hydrualic ports are located where they can link to the hydraulic circuit. The pump/planetary coupler’s shaft is clamped to the collar of the planetary gear set. Some of the bolts that fasten the front drive unit together have UNF threads. Thus, assembly and disassembly were kept to a minimum in order to hinder wearing of the threads.

Figure 43. Assembled front drive unit

Rear Drive Unit: The rear drive unit is more difficult to put together than the front. The flange mounted bearings are fastened to the inside of the rear dropouts. The motor mount – with the motor fastened to it – is fastened to the top of the right rear dropout.

Figure 44. Assembled rear drive unit

A dimensioned drawing was unable to be obtained for the clutch, therefore calipers were used in order to dimension the cluch's bolt flange. This resulted in minor misalignment between the clutch bolt flange and the corresponding bolt pattern in the rear dropout. This makes assembly slightly more difficult, however, it is still possible if the shaft and the clutch/gear assembly, are installed simultaneously. In order to tighten the bolt that holds the clutch's aluminum cable guide in place, a ball ended allen wrench needs to be fed through the hole in the plate and a ½"socket wrench, ratcheted on the otherside of the dropout.

Figure 45. The hole needed to access the aluminum cable guide adjacent to the clutch housing

Lockwashers were used on all of the rear drive unit bolts to prevent untightening. All of the bolts on the rear drive unit are UNC. The aluminum threads in the motor mount were tightened down with care to prevent stripping of the threads. Spacers were put on the drive shaft to axially locate the hub. There are setscrews on the bearing mounted flanges and the rear wheel hub. These should be tightend prior to riding. A bolt and washer were fastened to the end of the drive

shaft to prevent the key in the clutch from falling out. The bolt should be only finger tight. If it is tightened too much, the preload can be taken out of the clutch and it will slip.

Figure 46. Set screws located on hub and flange bearing

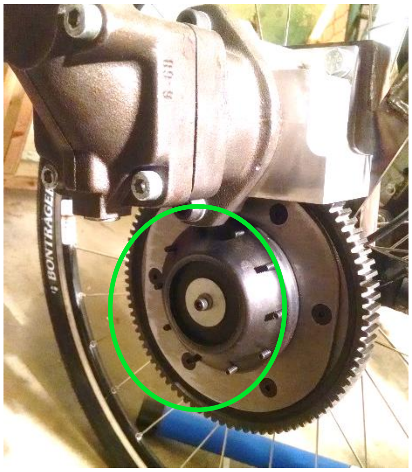


Figure 47. Washer that hold clutch key in place

Front Fork: The front fork from the 2014 Cal Poly Chainless Challenge bike was reused. The head tube on this year’s bike was faced and reamed to accommodate this fork. The bearing seats for this fork were pressed into the head tube.

Brake Components: The front brakes were a part of the reused front fork. The rear caliper brake was purchased and is mounted to the custom brake bridge. The rear brake cable is housed and held to the frame by zipties. Cables tensions were adjusted using the barrel adjusters.

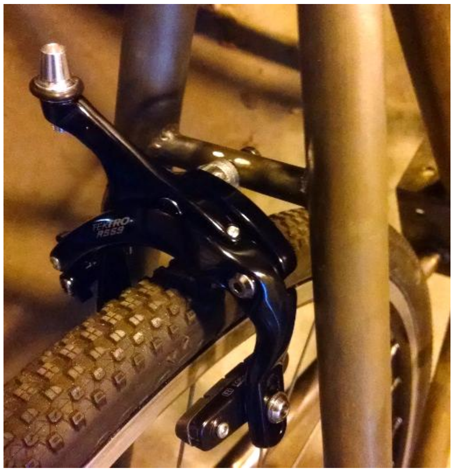


Figure 48. Brake caliper mounted to custom brake bridge

Clutch/Gear: The clutch was disassembled and resassembled multiple times in order to test the bike with Belleville washers of different thicknesses. Care was taken to not expose the greased internals of the clutch to a dirty environment. The tapered clutch driver was lightly greased to prevent galling of the friction surfaces. This being said, the driver should not be greased so much that the surfaces don't engage.

The countersunk bolts that hold the clutch, plate, and gear together will make the clutch assembly concentric if the bolts are visibly seated in the coutersink, subsequent to fastening.

Standard bike cable and houing was originally used to actuate the clutch. However, as thicker springs were tested the actuation force required caused the cable housing to compress and the clutch would not actuate.

Hydraulic Cirucuit: The hydraulic circuit has a very specfic setup, therefore all hoses and fittings must be properly installed in order for them to tie in with the hydraulic components, i.e. pump, motor, and accumulator.

All pipe fittings in the circuit were sealed using Loctite 545. The JIC fittings are self sealing due to their 37 degree chamfer. The JIC fittings were carefully tightened down to ensure that the chamfered end was not damaged.

Hydraulic fluid was added to the system at the reservoir. The reservoir was positioned at the highest elevation relative to the other hydraulic compenents. This was done to force air to rise to the reservoir and to force the fluid to flow downward. The reservoir was then mounted with zipties toward the front of of the bikes trigangle. The reservoir was carefully opened in order to get rid of the initial air in the system.

The accumulator is fixed with hose clamps to an angle iron fixture. We found this fixture in the IME machine shop and modified it to fit our application. The angle iron fixture is clamped to the left seat stay.

Figure 49. Accumulator clamped to the angle bracket fixture

7. Design Verification

7.1. Test Descriptions and Results

Bike Under Pressure set out to test three main characteristics of the bike, it's endurance, efficiency, and speed. The team also considered the bike's weight, turning radius, clutch efficacy, and ergonomics.

7.1.1. Race Testing (Endurance, Speed, and Efficiency)

The bike's endurance was tested by simply riding the bike for long distances. A ½ mile course was mapped out and teammates took turns riding the course. During endurance testing the bike was ridden a total of 35 miles. Times were recorded for the first 12 miles in order to determine if the average speed was competitive relative to the average speeds of bikes in the 2014 Chainless Challenge endurance race. Bike Under Pressure determined the average speed using an odometer and recorded mile times. The average speed was roughly 11 mph. This speed is very competitive considering the average speed for the winning bike in the 2014 event was 9 mph. The average speed from competition also includes exchange time, and Bike Under Pressure must take this into account to which figuring their best times.

Table 2. Endurance Testing Results

Mile	Time (min:sec)
1	5:40
2	5:38
3	5:31
4	5:47
5	5:40
6	5:38
7*	5:19
8	5:25
9	5:24
10	5:21
11	5:25
12	5:19
*Resolved pinion key	

Additionally, subjecting the bike to this much riding allowed Bike Under Pressure to gain familiarity with operating the vehicle and to troubleshoot any issues. A major issue that was resolved was the unsupported front gearbox pinion. During the first six miles of riding the bike made a loud creaking noise. The team was clueless as to what was causing this noise until the key that locks the pinion fell out of the keyway. Originally, there was a small aluminum key stop that was fastened to the threaded hole in the shaft of the flanged coupler. This key stop fell out prior to testing; however the key itself did not break free until after the first six miles of testing. The problem was resolved by fastening a washer to the end of the shaft. A lock washer was used in order to prevent the bolt from unfastening. After fixing this problem the bike rode

much smoother and more efficiently, this is evident in the decreased mile time after the sixth mile of testing.

The bike's top speed was tested in order to determine how competitive the bike would be in the sprint event. Testing for this was done on a 200 meter straight away, which is the same distance as the actual event. Prior to riding the bike on the straight away, the accumulator was charged to 3000 psi. This was the maximum pressure reading of the original pressure gauge in the hydraulic circuit. Two techniques were used for dumping the accumulator. The accumulator was dumped at the beginning of the 200 meters and then the rider maintained the bike's speed while in direct drive mode. Alternatively, the rider pedaled the bike up to a cadence and then dumped the accumulator.

Table 3. Sprint Testing Results

Trial	Time (sec)
1^	29.2
2^	28.4
3^	27.8
4^	28.8
5^	29.3
6^	?
7^	28.8
8^	29.0
9*	27.4
10*	26.5
11*	27.3
12*	27.7
13*	26.5
^Accumulator dumped last * Accumulator dumped first Wind was approx. same for all tests	

Before performing 13 time trials on the sprint course it was evident that dumping the accumulator at the start of the course produced faster times. The average time for the sprint test was 28.1 secs. This time is competitive considering the winning time in the 2014 event was 22 seconds. The times recorded during testing may be slightly faster than those achieved during the actual competition. This is because the rider was assisted by the wind and the course had a very slight down grade. The lightest weight team member completed the course in the shortest amount of time, because the boost from the accumulator was most effective for this rider. This rider will represent the team for this event in the actual competition. The actual top speed (not just the sprint time) achieved during this testing was 23.1 mph. This occurred with accumulator assist.

The bike's efficiency was tested by using a straight strip of road as the course. The accumulator was charged to various different pressures prior to testing. The rider then sat on the bike without pedaling and dumped the energy stored in the accumulator. Bike Under Pressure did this test 10

times using various techniques for dumping the accumulator. The accumulator can be dumped all at once or it can be dumped by intermittently opening and closing the ball valve. Parker uses a dimensionless scoring ratio which is:

$$\text{Score} = \frac{\text{W} * \text{L}}{\text{P} * \text{V}}$$

where W is the combined weight of the bicycle and rider, L is the distance travelled in inches, P is the gas pre-charge pressure in psi, and V is accumulator volume in cubic inches. The best score achieved during testing was 30.5, which is roughly what the best results were from last year's competition.

7.1.2. Other Tests

The weight was tested by placing it on two scales – one for each wheel – and the total recorded. The bicycle weighs 126 pounds.

The turning radius was tested by mimicking the slalom set up from competition. This set up requires that the bicycle maintain a 7.5 ft testing radius. They bicycle made several passes through two full turns of the slalom course with no problem.

The ergonomics were tested simply by riding it. General opinion of the Bike Under Pressure team members was positive. Pedaling is difficult at low speeds (and in general), as expected, but smooth. The increased weight makes turning more difficult than a normal bicycle; leaning into corners must be much less compared to usual bicycles. The Q-factor – the distance between the pedal attachment points at the cranks – of 6 inches is slightly wider than a typical mountain bicycle (5.7 in), but not to the point of disrupted pedaling.

Regenerative braking was tested in several ways: first by rolling the bike while in charging mode, and second by actually stopping the bicycle by switching from direct drive to charging mode. Both methods worked: the bicycle stops moving and accumulator pressure rises. Regenerative braking can stop the bicycle and rider moving at 11 mph within 30 ft. During this stopping the accumulator fluid pressure rises from zero to 1000 psi.

7.1.3. Clutch Performance

Clutch performance was testing by pulling the clutch lever, then manually turning the gear to see if disengagement was achieved. With the original springs (two 0.05 in Belleville washers in parallel), clutch actuation was easily achieved. The next test was to ride the bicycle in direct drive mode. Unfortunately, the clutch slipped excessively. In was unable to hold the necessary torque. Different spring sizes and series combinations were tested. While Bike Under Pressure thought other series and parallel combinations were possible, there is simply not enough space or geometry to allow any other than two springs in series. Springs that were too stiff would cause the housing to fail; springs that were too weak would cause the clutch to slip.

Table 4 below shows the manufacturer data for the springs; data is for a single Belleville washer. The working load is the recommended operating point (likely in the middle of the deflection). Flat load is the load required to make the washers flat i.e. the solid length. The following table summarizes the attempts to work the clutch. The effective spring rate assumes that these

Belleville washers are linear springs which have a spring rate equal to their working load divided by some deflection which was kept the same for all tests. The spring rate was used for comparison purposes when decided which spring combination to test next.

Table 4. Summary of Belleville Washers

Thickness (in)	Working Load (lbf)	Flat Load (lbf)
0.050	360	??
0.065	590	860
0.084	855	1488
0.097	1180	2140

Table 5. Summary of Clutch Testing

Spring Series Combination	Effective Spring Rate (lbf/deflection)	Result
0.050/0.050	180	Actuates easily; slips badly in direct drive
0.065/0.065	295	Actuates moderately; slips in direct drive
0.084/0.084	428	Does not actuate; housing ferrules badly damaged; does not slip in direct drive or accumulator assist
0.084/0.050	253	Never tested; will likely slip
0.084/0.065	349	Does not actuate; does not slip in direct drive or accumulator assist; housing split

Other effective intermediate spring rates could be achieved with thin Belleville washers in parallel-series combination or with intermediate thicknesses. In addition stiffer, stronger housing could be used. Bike Under Pressure had already made modifications to the housing with some success, but was running out of time to make the necessary modifications to increase available actuation force with a long lever (or otherwise). In the end, the clutch was abandoned and made solid by bolting both sides together. While this hurts performance in the efficiency event and (potentially) the innovation judging, it will help in the sprint event. Better sprint performance is possible due because the accumulator can be pressurized higher; there is no risk of slipping the clutch.

7.2. Specification Verification Checklist

A modified requirements table shows the results from this testing. Color coding green means the requirement was met; color coding red means the requirement was not met. If un-colored, the result is inconclusive.

Table 6. Engineering Requirement Verification

Req.#	Parameter Description	Requirement	Tolerance	Risk	Result
1	Top Speed	17 (mph)	Min	L	23
2	# of Total Components	50 excluding fasteners	Max	L	~50 not including bolts, keys etc.
3	# of Custom Components	10	Max	M	11+frame
4	Life	50 (miles)	Min	H	35 so far w/ no major problems
5	Production Cost	5000 ($)	Max	M	4600
6	Safety	Rules Guidelines	Min	L	No dangerous failures
7	Turning Radius	7.5 ft	Max	L	7
8	Overall Weight of Bike	225 (lb)	Max	M	126
9	200m Sprint Time	30 (sec)	Max	M	~30
10	Average Speed (Time Trial)	9 (mph)	Min	M	11
11	Energy Recapture (Regen)	> 0 (%)	Min	M	√

The number of custom components requirement was not met; however, Bike Under Pressure managed to make all the necessary parts in time. The number of total components requirement was never fully set. It was meant to keep the design simple. Bike Under Pressure did finish the bicycle in time for competition, so this requirement is essentially met. The life testing was not completed for 50 full miles; however, due to the limited number of problems during testing, the bicycle should easily last well past the required miles, and thus competition.

8. Project Management Plan

Participation in the Chainless Challenge Completion will fulfill the senior project requirement for the Bike Under Pressure team. As such, the team received support from the Cal Poly Mechanical Engineering department through the senior project class series and professor advising. The senior project class provided structure to the planning, design, and execution of the project.

Additional assignments in the classroom – outside of the requirements of the Parker competition – focused the team on the design process. One such assignment was to choose team roles and create a list of expectations. This document is summarized in the following paragraphs below because it defines the project plan.

- Matt Pallotta is the point of contact and lead manufacturer. As the point of contact, he is responsible for communicating with our advisor, professor, and Parker Hannifin. He is also responsible for fostering communication within the team. This is done by making sure all members remain current with all aspects of the project. As the lead manufacturer, he is responsible for ensuring all custom designed parts are properly manufactured.

- Jack Rechtin is the lead designer of the bike. As the lead designer, he is responsible for modeling the bike in SolidWorks. In addition to modeling the bike, he will make drawings for custom parts so that they can be properly manufactured and fabricated.

- Nathan Klammer is the head of engineering analysis, meeting secretary, and document controller. As the head of engineering analysis he is responsible for performing hand calculations on critical aspects of the bike. As the meeting secretary he will be responsible for formatting the meeting agenda and recording the meeting minutes. As the document controller he is responsible for assigning various parts of the final report to different team members and organizing the report in the required format.

- Kemper Whaley is the treasurer and leader of the hydraulic system. As the treasurer he is responsible for keeping track of all expenses and formatting the cost analysis. He will also handle reimbursement paperwork for purchases made by all team members. As the leader of the hydraulic system he will be in charge of designing, purchasing, and installing the hydraulic circuit.

- The aforementioned roles of each team member are flexible. If another team member needs help with completing a task he can ask for the assistance of his team. As the project progresses, if it is apparent that the workload is unbalanced then the other team members will be expected to work to ensure fairness.

- All members will be expected to attend meetings unless they have a reasonable excuse for being unable to attend. If a team member absent to a meeting, and does not give a reasonable excuse, the rest of the team will bring this to that team member's attention. If unexcused absences continue with this team member, the rest of the team members will inform the professor.

- Weekly meetings will be held with Professor Fabijanic. A meeting agenda will be sent prior to 5 p.m. on the day before the meeting. Minutes for the meeting will be sent to all attendees before 5 p.m. of the day following the meeting. Both of these documents will be sent out by the team secretary.

Another important assignment and task – important not just for the class work – was creating a schedule. The team created a Gantt chart (available in Appendix A) of the various tasks it takes

to move the project from problem definition to competition. This chart identified critical tasks, those that if delayed, will slow the entire project. Of primary concern was the time necessary for machining of custom components. Also of importance was the lead time on ordered parts. During the course of the year, the schedule wasn't followed rigidly; rather it was used as a guide to check against. The senior project class also kept the project on schedule.

9. Competition Results, Recommendations, and Conclusions

9.1. Competition Results

The Chainless Challenge competition was help on April 8 – 10th in Irvine, California. Entries from nine different schools including Cal Poly competed. Bike Under Pressure's bicycle finished each of the races without any leaks or mechanical failures. The 200 meter sprint time was 29.9 seconds, good enough to place second. This also received points for second place in the sprint partnership. Cal Poly was paired with the 5th place team for the sprint partnership. For the efficiency race, the 125 pound bicycle (with a 170 pound rider) cruised 181 meters (7120 inches) using a 63 cubic inch accumulator with a gas pre-charge of 800 psi. This results in an efficiency score of 42, easily placing first. Finally in the endurance, time-trail race, three riders completed the 6 mile course in 30 minutes 48 seconds, winning the race by a few minutes.

Figure 50. Cal Poly Bike Under Pressure at the 2015 Parker Hannifin Chainless Challenge

In addition to the strong racing performance, Cal Poly Bike Under Pressure placed well in the design judging categories. These were: third in innovation, first in reliability/safety, fourth in manufacturability/workmanship, third in best design chosen by peers, third in cost analysis, and second in best paper/presentation. This performance gave Cal Poly enough points to win first in best overall with 1125 points.

9.2. Recommendations and Lessons Learned

Here is a list (in no particular order of importance) of recommendations and lessons learned for future teams and continuing, improving work on this hydraulic bicycle design:

- Variable displacement hydraulic pumps and motors is generally ill suited for human power. Significant design work at controlling the swash plate is recommended should the feat be attempted.
- A vacuum pump with a 3-way valve and air filter is useful for filling the hydraulic circuit and avoiding air in the lines. This method was recommended by a Parker employee and was not attempted by Bike Under Pressure.
- Parker manufactures gearheads which are compatible with many of their products. Bike Under Pressue did not use these gearheads, but they are recommended for investigation in the future because they could possibly be provided free of cost.
- Up front work in fixturing is worth a fortune. Maintaining concentricity in manufacturing and assembly is difficult but necessary for smooth operation of rotary components.
- Ask the department technicians for help, especially with maintaining tolerances in manufacturing.
- Solenoid valves, while not necessary, would aid the rider in controlling valve operation (wires can run to a control panel mounted in an easy to access place).
- Instrument previous Cal Poly bicycles (Bike Under Pressure's is in working order at the time of this report's publication) to gain data for insight into loads and hydraulic performance. Do not take them apart for parts unless the function of the parts and assemblies are absolutely known. Bike Under Pressure's final prototype should serve as a starting point for future designs and a learning tool for new teams.
- A lightweight, efficient linear actuator design is possible. Study into the efficiency (theoretical modeling the rotary motion with known linear efficiency curves AND testing to compare to the model) is highly recommended.
- Maintaining an accurate model helps visualize problems before they arise, but many problems will occur even if the model is "perfect".

This is only a small portion of the lessons learned. It is recommended for future teams to do their reading and research. They can also contact the Bike Under Pressure team to ask about this project (as long as the calpoly.edu addresses are still valid).

9.3. Conclusion and Thanks

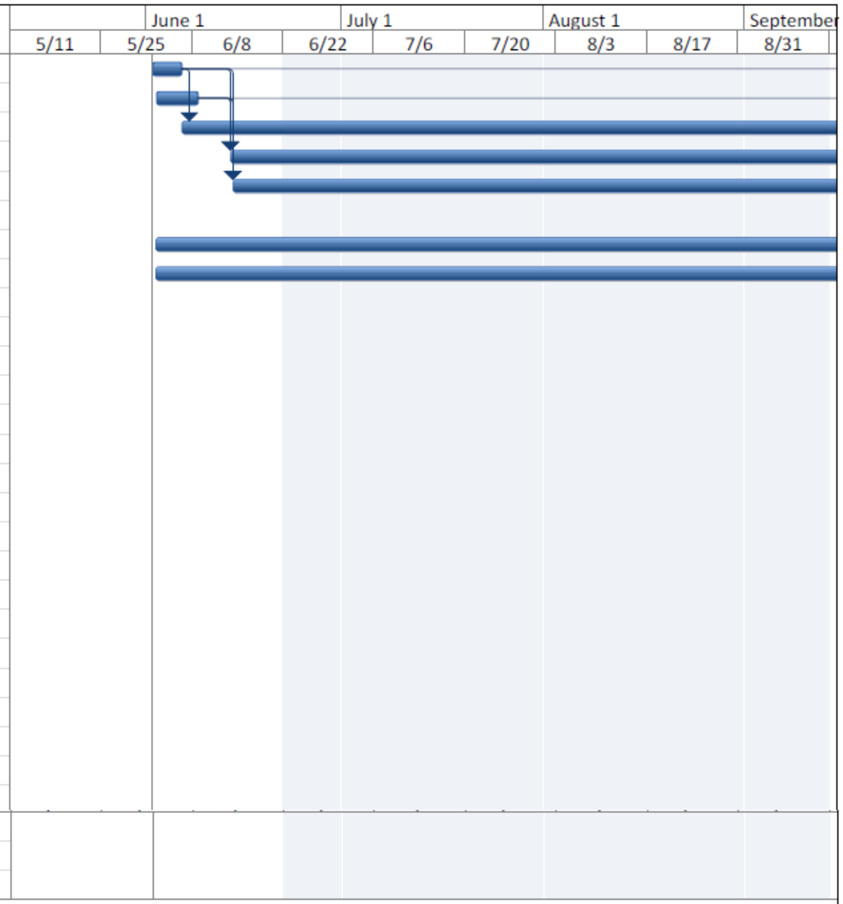
Hydraulic bicycles are not expected, nor recommended to replace conventional chained bicycles due to high production cost, low efficiency, high weight, and decreased ridability. While hydraulics can be incredibly efficient in their current accepted applications, the low speed nature

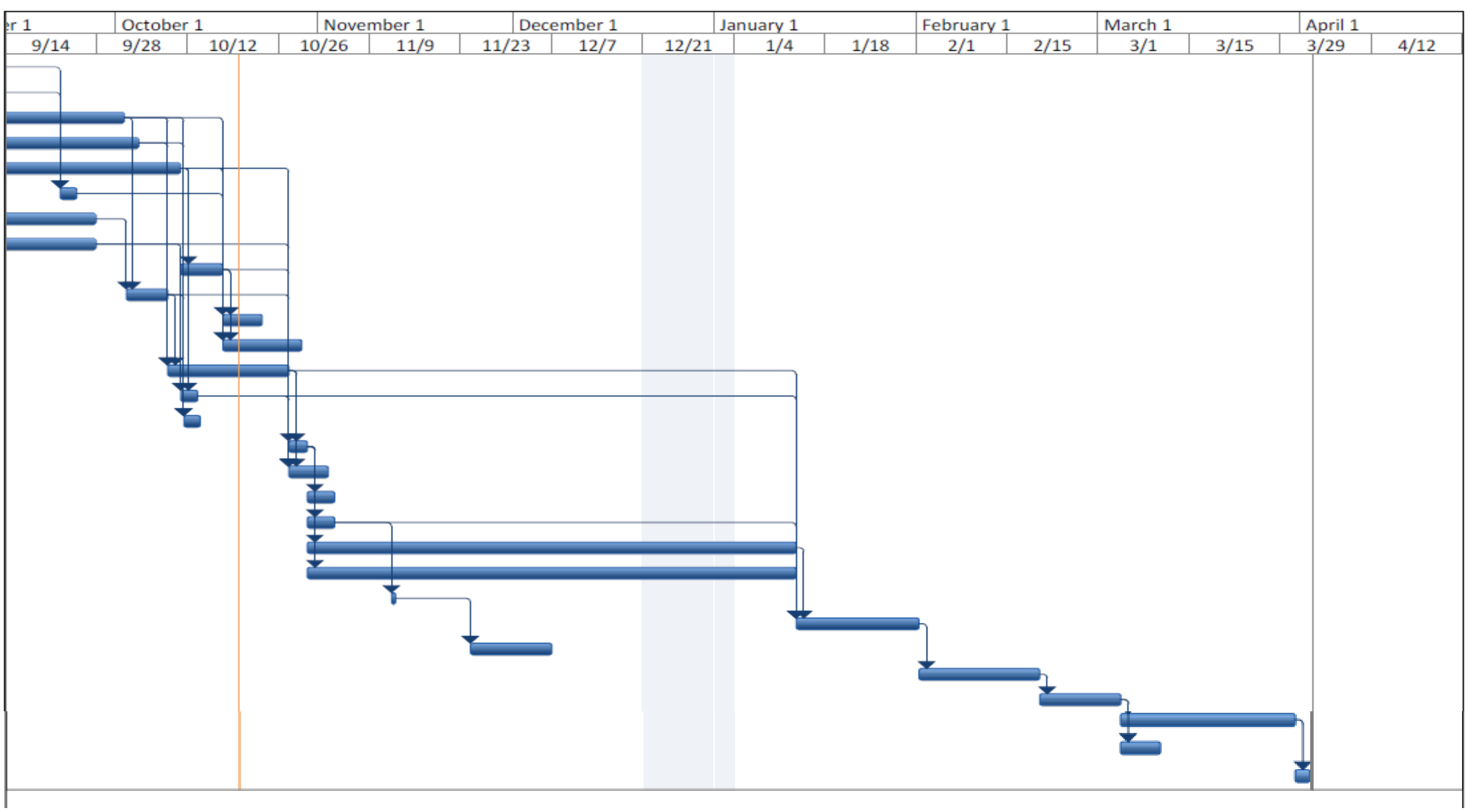
of human power is a difficult relative roadblock. Research and investigation into hydraulic bicycles does, however, reveal promising technologies. Regenerative braking with hydraulics offers an efficient means of capturing braking energy. Energy storage in pressurized fluid accumulators is an alternative to chemical battery or mechanical flywheel storage.

The Cal Poly 2014-15 Chainless Challenge Team “Bike Under Pressure” would like to thank Parker Hannifin for hosting this competition. It provided the backbone for a great senior project. We’d like to thank Sandy Harper, the Parker Meeting and Events Services, all other Parker employees involved, and our advisors for helping us make this project a reality. We would also like to thank our project advisors: John Fabijanic, Dr. James Widmann. We also extend special thanks to the technicians Ladd Caine and George Leone for all their help and advice, and without whom the prototype never would have been completed.

10. Appendix A: Gantt Chart

ID		Task Name
1		Conduct Concept Design review
2		Test proof of concept bike
3		Develop hydraulic circuit
4		Design pedal, input, and output geometry
5		Design frame and interfaces/mounting
6		Select hydraulic components
7		Obtain efficiency data from Parker/test for efficiency
8		Obtain competition results from Parker
9		Select hydraulic hose size
10		Determine desired gear ratio
11		Select gears
12		Analyze theoretical efficiency and power
13		Perform stress/deflection analysis
14		Order hydraulic components
15		Order mechanical components
16		Perform initial cost analysis
17		Compile Final design
18		Conduct design review
19		Write final design report
20		Machine custom parts
21		Frame Builders Custom Frame
22		Conduct midway review
23		Assemble prototype
24		Test prototype
25		Redesign critical components
26		Update final design
27		Final reliability testing
28		Write final project report
29		Ready to compete





11. Appendix B: Table of Previous Designs

Previous Cal Poly Design Summaries		
2005	**2006**	**2007**
		
Pump		
Cam follower with linear piston pump	90 degree bevel gear with welded bracket to frame holding Piston Pump	90 degree bevel gear mounted with cast bracket holding Piston Pump and gears
Good		
● Light Weight ● Compact	● Compact ● Efficient	● Compact ● Efficient
Bad		
● Not Efficient ● Uncomfortable to ride ● Hard to manufacture	● Not Rigid (skipped gears) ● Misaligned gears (damage)	● Not rigid (skipped gears) ● Misaligned gears (damage)
Motor		
Gear pump bracketed to the rear drop out, a chain transferring torque to hub	Variable displacement piston pump coupled with a shaft drive to spiral bevel gear attached to rear hub	Variable displacement piston pump coupled with a shaft drive to spiral bevel gear attached to rear hub
Good		
● Efficient power transfer ● Rigid (didn't skip gears) ● Light Weight	● Efficient power transfer ● Infinitely variable speed ratio	● Efficient power transfer ● Infinitely variable speed ratio
Bad		
● Used Chain ● Non Parker motor	● Heavy ● Skipped gears ● High Center of gravity	● Heavy + High CG ● Skipped gears (broke gears) ● Broke drive shaft during sprint race
Accumulator		
Bike trailer holding high pressure reservoir and low pressure tank attached behind bicycle	Bike trailer holding high pressure reservoir and low pressure tank connect behind bicycle	Bike trailer holding high pressure reservoir and low pressure tank connect behind bicycle
Good		
● Reduced bike weight for endurance race	● Reduced bike weight for endurance race ● **Won Sprint Race**	● Reduced bike weight for endurance race
Bad		
● Adds additional rolling resistance	● Adds additional rolling resistance	● Broke rear drive due to high pre-charge pressure

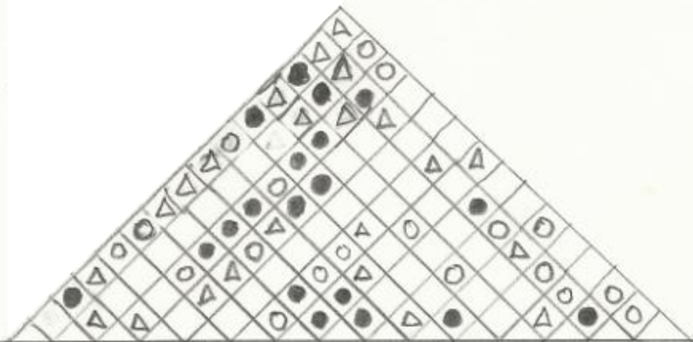
Previous Cal Poly Design Summaries		
2008	**2009**	**2012**
		
Pump		
90 degree bevel gear mounted within gear box. Pump supported by shaft and bearing. 11 cc/rev fixed displacement gear pump (high volumetric flow rate).	90 degree bevel gear mounted within gear box. Pump supported by Aluminum gear box. Oildyne miniature 5-piston pump.	Parker F-11 pump is powered by double-chain speed increaser. This allows the desired ratio of 1:2:8. Supported by hitch that supports all hydraulic components.
Good		
● Contained ● Light Weight ●Compact ● G-Box does not rely on bike frame to support Bevel gear loads ● Better gear meshing	● Contained ● Light Weight ●Compact ● G-Box does not rely on bike frame to support Bevel gear loads ● Better gear meshing	● Tested pump rpm can be accommodated by large gear ratio. ● Design has larger alignment tolerance because gear meshing is not a problem ● Rear weight balanced by the accumulator toward the front of the bike.
Bad		
● Low gear ratio (2:1) ● Pump requires high rpm	● Miniature piston pump makes for high resistance pedaling ● Produced lower than expected flowrates ● Low gear ratio (3:1) ● Gear train efficiency ratios were not conservative enough	● High center of gravity.
Motor		
11 cc/rev fixed displacement gear motor. Motor turns 14-speed Rohloff internal gear hub.	5 cc/rev hydraulic motor receives high-pressure fluid from pump. Motor turns 14-speed Rohloff internal gear hub. Gear hub allows for varying speeds without heavy and inefficient variable displacement motor.	Parker F-11 motor output is connected to chain and sprocket speed reducers. Supported by hitch that supports all hydraulic components.
Good		

● Gear hub allows for varying speeds without heavy and inefficient variable displacement motor ● Less resistant fed back to pump ● Direct drive from motor to drive shaft, transfers torque more efficiently ● Designed to be removable for future Chainless Challenge competitions	● Gear hub allows for varying speeds without heavy and inefficient variable displacement ● Piston pumps are efficient at high pressures	● Because of high gear ratio from pedal to pump, the motor functions at high efficiency.
Bad		
● Cannot adjust gear ratio because motor is fixed and because drive mechanism is direct drive.	● Motor is inefficient at low volumetric flowrates ● Motor displacement volume is fixed so if gear ratio is wrong need new motor ● Gear train efficiency ratios were not conservative enough	● Design was so efficient that Parker judges decided to revamp the rules.
Accumulator		
3.5 gallon pre-charged parker accumulator	3.5 gallon pre-charged parker accumulator Subsequent to this competition the large accumulator attached pulled in a trailer was banned	1-liter piston accumulator position in the middle of the bike. Enabled the design to have regenerative braking.
Good		
● Reduced bike weight for endurance race	● Reduced bike weight for endurance race	● Position offsets rear weight of hydraulic components ● Small accumulator keeps weight lower
Bad		
● Adds additional rolling resistance ● Need for hydraulic system control requires long tubing segments which increases losses	● Adds additional rolling resistance ● Need for hydraulic system control requires long tubing segments which increases losses ● Provided too much power. Bike reached speeds of 40 mph.	● Raised the center of gravity of the bike. ● Was relatively small in volume, thus limiting regenerative abilities.

Previous Cal Poly Design Summaries	
2013	2014
	
Pump	
Parker F11-005, 5 cc/rev fixed gear pump. Motor supported by frame and sheet metal spacers. Linked to pedals by two gears with high gear ratio between them.	Linear actuators (hydraulic cylinders) are attached to an offset arm at the pedal axle which allows for rotational motion to be converted to linear pumping motion.
Good	
● Gear ratio allows for pump to reach high/efficient RPM ● Low center of gravity ● Close to motor so less losses from long tubing	● Linear actuators are more light weight than hydraulic pump. ● Can handle very high forces.
Bad	
● Fixed displacement pump hence design cannot be adjusted if design has incorrect gear ratio ● Supported by sheet metal spacers rather than by gear boxes, which was done in previous years	● Difficult to transfer rotational to linear motion. ● Difficult to properly time front and back linear actuators. ● Linear hydraulic actuators are not designed for high speeds like pumps are.
Motor	
Parker F11-005, 5 cc/rev fixed gear motor, directly drives a 16" wheel. Using a smaller diameter wheel makes desired shaft speed achievable.	Motor is just another linear actuator slightly less than 90 degrees out of phase with front linear actuator. Attached to an offset arm on a gear. This gear is meshed with a gear that is directly linked to the drive wheel.
Good	
● Directly driven wheel is more efficient than adding a gear reducer.	● Linear actuators are more light weight than hydraulic motor. ● Can handle very high forces.
Bad	
● Motor is located on one side of the bike, thereforeoffsetting the bikes weight.● Fixed gear motor and no gear reducer makes for difficult drivetrain modifications.	● Difficult to transfer rotational to linear motion.● Difficult to properly time front and back linear actuators.● Linear hydraulic actuators are not designed for highspeeds like motors are.

Accumulator	
1 gallon bladder accumulator supported by welded manifold at the rear of the bike.	2, 1.5 liter piston accumulators. We never mounted to bike.
Good	
● Large accumulator volume allows for more regenerative capacity ● Low rear wheel an putting accumulator at rear of bike shifts down the entire center of gravity	● Using two accumulators allows for rider to have more control of hydraulic fluid and usage.
Bad	
● Large, 1 gallon, accumulator makes for a very heavy bike (304 lbs.).	● Accumulators are heavy

12. Appendix C: QFD



Customer Requirements (Step #2)

Customer (Step #1) Requirements (Whats)	Weighting (Total 100)	Advisors	Parker Hanniffin Judges	Rider	Chainless Challenge Team	Accumulator Size	MEOP of Accumulator	Overall Weight (lbf)	Wheel Base (in)	Color	# of Welds	# of Components	# of Custom Components	# of Stock Components	# of Wheels	Center of Gravity Height (in)	# of Fittings	Hose Diameter (in)	Drive Train	Gear Ratio	2012-2013 Bike	2013-2014 Bike
Safety																						
Safe to ride		5	5	6	4	1	3	1	0	0	1	0	0	0	1	3	0	0	0	0	5	6
Safe to transport/ship		3	3	3	4	1	0	1	1	0	0	0	0	0	0	0	0	0	0	0	5	5
Safe to manufacture		5	3	3	5	0	0	0	0	0	1	0	1	0	0	0	0	0	0	0	7	5
Comfortable for rider		3	3	5	3	0	0	1	0	0	0	0	0	0	1	1	0	0	0	0	6	7
Performance																						
Acceleration		5	5	7	7	3	9	9	0	0	0	0	0	0	1	0	0	1	3	3	5	?
Max Speed		6	6	7	7	9	3	9	0	0	0	0	0	0	1	0	0	1	3	3	7	?
Turning Radius		4	5	6	6	0	0	1	9	0	0	0	0	0	3	3	0	0	0	0	4	8
Light Weight		7	7	7	7	9	0	9	0	0	0	3	1	0	3	0	1	0	1	0	5	7
High cruising speed		6	6	7	7	3	0	3	0	0	0	3	1	0	3	0	1	3	1	9	7	?
Human Powered *																						
Can handle slight hills		5	5	7	6	3	0	9	0	0	0	1	1	0	1	0	1	0	1	9	6	?
Easy to maintain speed		5	5	7	6	3	0	3	0	0	0	3	3	1	1	0	1	3	1	9	7	?
Efficient		8	9	6	8	3	9	9	0	0	0	1	1	1	1	0	3	3	3	3	6	?
Regenerative Braking		7	9	4	8	1	1	3	0	0	0	1	1	1	0	0	3	1	3	1	9	?
Reliable		9	7	8	9	0	0	0	0	0	9	9	9	9	1	0	9	1	3	0	7	0
Other																						
Aesthetics		5	7	3	3	1	0	1	1	9	3	3	3	3	3	1	1	0	1	0	6	4
Innovative Design		5	9	3	8	0	0	3	0	0	0	0	1	0	0	0	0	0	3	0	3	6
Competativeness		6	6	7	9	3	3	1	1	0	1	1	3	1	1	1	3	1	9	3	9	0
Units						gal	psi	lbf	in	-	-	-	-	-	-	in	-	in	-	-		
2012-2013 Bike						1	1500	304	66.9	CP	5	80	20	60	2	13.5	10	0.75	Pump & Motor	~~9:1~~ 5.5:1		
2013-2014 Bike						0.8	4000	60	50	Blue	10	60	15	45	2	16	2	0.13	Linear Actuator	3:1		
Target						0.5	2000	100	42	CP	10	30	10	20	2	16	10	0.63	Hybrid	10:1		

● = 9	Strong Correlation
○ = 3	Medium Correlation
Δ = 1	Small Correlation
Blank	No Correlation

13. Appendix D: Calculations

Planetary Gearset to Pump Coupling Shaft Stress

File:C:\Users\Nathan\Documents\EES\ME429 Coupling Shaft Stress.EES 10/19/2014 3:36:55 PM Page 1
EES Ver. 9.699: #552: For use by Mech. Engin. Students and Faculty at Cal Poly

Planetary Gearset to Pump Coupling Shaft Stress

The coupling shaft is the smallest component in the mechanical drivetrain in the front. It must withstand the peak input torque from the rider. The very large (r/d > 1) fillet avoids a stress concentration; any concentration is assumed to be negligible.

Peak torque input from rider in lbf-in, assuming 200 lbf rider putting full weight on one 6in crank

$$Torque_{rider} = 1200 \quad [lbf \cdot in]$$

Gearing from torque input to coupling shaft

$$gearing = 15$$

Shaft Diameter in inches

$$d = 0.3 \quad [in]$$

Torque in the shaft

$$Torque = \frac{Torque_{rider}}{gearing}$$

Shear stress in shaft

$$\tau = \frac{Torque \cdot \frac{d}{2}}{\frac{\pi \cdot d^4}{32}}$$

Von Mises Stress in shaft

$$\sigma_{VM} = \sqrt{3 \cdot \tau^2}$$

Yield Strength of Steel

$$S_{Y,steel} = 36000$$

Factor of Safety against yielding

$$n_{FOS} = \frac{S_{Y,steel}}{\sigma_{VM}}$$

This result suggests that the shaft can withstand the peak torque it will transmit.

SOLUTION

Unit Settings: SI C kPa kJ mass deg

$d = 0.3$ [in]	$gearing = 15$	$n_{FOS} = 1.377$
$\sigma_{VM} = 26137$	$S_{Y,steel} = 36000$	$\tau = 15090$
$Torque = 80$	$Torque_{rider} = 1200$ [lbf·in]	

2 potential unit problems were detected.

Front Gearbox Bolt Stress

File:C:\Users\Nathan\Documents\EES\ME429 Front Gearbox Bolt Stress.EES 10/19/2014 3:16:17 PM Page 1
EES Ver. 9.699: #552: For use by Mech. Engin. Students and Faculty at Cal Poly

Stress in 4x M6 bolts holding front assembly components, including the pump and planetary gearset, to the front gearbox

These calculations find the stress in the bolts if the bicylcle is laying horizontal. In this position the assembly components act like a cantilevered beam. The load is the static weight which has a moment arm of the distance from the center of gravity of the top components to the bolts. This weight and dimension were taken from the Solidworks model. There are two M6 bolts which will resist the moment created. The other two bolts act as the pivot point and do no experience addition loading.

This calculation assumes the bolts are preloaded. It is also assumed that the bolts take the entire load, therefore the stiffness constant is equal to 1.

Load and Geometry

$W = 15.73$ *Weight of components mounted to gearbox in lbf*

$r = 7$ *Moment arm from components CoG to top of gearbox in inches*

$b = 2$ *Width of bolts in inches, serves as moment arm for bolts*

$n = 2$ *Number of bolts opposing moment of weight*

Sum of the moments about opposite bolts, P is load on one bolt

$$n \cdot P \cdot b - W \cdot r = 0$$

Tensile area of one M6 bolt in mm^2

$$A_t = 20.1$$

Minimum Proof Strength of Property Class 4.6 bolt in MPa

$$S_p = 225$$

Preload of bolt for nonpermanent connection

$$F_i = 0.75 \cdot A_t \cdot S_p$$

Normal Stress in MPa (conversion of P from lbf to N), assuming the bolt takes entire external load (stiffness constant c = 1)

$$\sigma = \frac{P \cdot 4.45 + F_i}{A_t}$$

Factor of Safety against exceeding Proof Strength

$$n_p = \frac{S_p}{\sigma}$$

Load factor against overloading

$$n_L = \frac{S_p \cdot A_t - F_i}{P \cdot 4.45}$$

The factor of safety shows that the bolt should not exceed the proof load. Additionally, the load factor predicts that the static load can be increased by up to 9 times before overloading occurs.

SOLUTION

Unit Settings: SI C kPa kJ mass deg

$A_t = 20.1$	$b = 2$	$F_i = 3392$	$n = 2$
$n_L = 9.23$	$n_p = 1.287$	$P = 27.53$	$r = 7$
$\sigma = 174.8$	$S_p = 225$	$W = 15.73$	

No unit problems were detected.

Press Fit Torque Capacity for Coupling Shaft

File:ME429 Splined Coupler Press Fit Torque Capacity.EES 4/1/2015 11:32:44 AM Page 1
EES Ver. 9.699: #552: For use by Mech. Engin. Students and Faculty at Cal Poly

Press Fit Torque Capacity for Coupling Shaft Splines

The coupling shaft (which couples the planetarty gearset output to the pump input) will have internal splines. In order to create these blind splines, a separate internaly splined segment will be made (with through splines). This segment will then be press fit into a socket in the coupling shaft. The press fit needs to carry the expected system torque. At this point in the front drivetrain, the peak torque is expected to be 80 in-lbf (the rider input toque of 1200 in-lbf after 1:12 gearing).

Pressure in Fit in psi

$$p = \frac{E \cdot \delta}{2 \cdot d^3} \cdot \left[\frac{(d_o^2 - d^2) \cdot (d^2 - d_i^2)}{d_o^2 - d_i^2} \right]$$

Young's Modulus of Steel in psi

$E = 2.9 \times 10^7$ [lbf/in^2]

Nominal shaft diameter in inches

$d = 1$ [in]

Inside shaft (segment) diameter in inches (from nominal spline diameter of 18mm)

$d_i = 0.709$ [in]

Outside hub diameter in inches

$d_o = 1.5$ [in]

Diametral interferance in inches (segment diameter must be larger than the nominal shaft diameter by this abount)

$\delta = 0.001$ [in]

Length of fit in inches

$l = 0.787$ [in]

Assumed coefficient of friction for dry steel on steel

$f = 0.74$

Torque capacity in in-lbf

$$T = \frac{\pi}{2} \cdot f \cdot p \cdot l \cdot d^2$$

Even with only .001 inches of interferance, the torque capacity of the fit will be more than sufficient. Care must be taken to precisely and accurately machine a the parts to ensure an interferance fit.

SOLUTION

Unit Settings: SI C kPa kJ mass deg

$d = 1$ [in]	$\delta = 0.001$ [in]	$d_i = 0.709$ [in]
$d_o = 1.5$ [in]	$E = 2.900E+07$ [lbf/in^2]	$f = 0.74$
$l = 0.787$ [in]	$p = 5159$ [psi]	$T = 4719$ [in-lbf]

No unit problems were detected.

Direct Drive Fluid Loss Calculations

Direct Drive Fluid Loss Calculations

This calculates the fluid losses in the system from the inlet of the pump to just before the inlet of the motor. In direct drive there is approximately 3 ft of hydraulic lines and one check valve contributing to losses.

Constants (Fluid Properties Assumed Constant)

Density of Hydraulic Fluid in slugs/ft³

$\rho = 0.921 \cdot 1.94$

Dynamic Viscosity

$\mu = 36.78 \cdot 1.45 \times 10^{-7} \cdot 0.921$

Gravity in ft/s⁵

$g = 32.2$

Geometry/System Parameters

Length in inches

$L = 36$

Diameter in inches

$D = 0.272$

Tube Relative Roughness in inches

$\varepsilon = 0.000005 \cdot 12$

Check Valve Loss Coefficient

$K_{check} = 2$

Inputs

Pressure in psi

$P_1 = 14.7$

Work into the system in hp

$\dot{W}_{in} = 0.25 \cdot \eta_{pump}$

Speed of input in rpm

$speed = 80 \cdot 15$

Efficiency of the Pump

$\eta_{pump} = 0.9$

Volumetric Flowrate in in³/sec

$$Q = \frac{5 \cdot 0.06102 \cdot speed}{60}$$

Average Velocity in in/s

$$\overline{V} = \frac{Q}{\pi \cdot \left[\frac{D}{2}\right]^2}$$

Reynold's Number

$$Re = \frac{\rho \cdot \overline{V} \cdot D \cdot \left[\frac{1}{12}\right]^4}{\mu}$$

Extended Bernoulli's Equation, using inches

$$\frac{[P_1 - P_2] \cdot 12^3}{\rho \cdot g} = h_l + h_{lm} + \delta h_{shaft}$$

Head gained from pump in inches

$$\delta h_{shaft} = \frac{-\dot{W}_{in} \cdot 550 \cdot 12}{Q \cdot \rho \cdot g \cdot \left[\frac{1}{12}\right]^3}$$

Major head losses in inches

$$h_l = f \cdot \frac{L}{D} \cdot \frac{\overline{V}^2}{2 \cdot g} \cdot \frac{1}{12}$$

Minor head losses in inches

$$h_{lm} = K_{check} \cdot \frac{\overline{V}^2}{2 \cdot g} \cdot \frac{1}{12}$$

Colebrook Correlation

$$\frac{1}{\sqrt{f}} = -2 \cdot \textbf{log}\left[\frac{\varepsilon}{D \cdot 3.7} + \frac{2.51}{Re \cdot \sqrt{f}}\right]$$

Power in Fluid at Motor Entrance

$$\dot{W}_{out} = Q \cdot [P_2 - P_1] \cdot \frac{1}{550 \cdot 12}$$

Fluid Efficiency in Direct Drive

$$\eta_{system} = \frac{\dot{W}_{out}}{\dot{W}_{in}}$$

The fluid efficiency at the rider's top pedaling speed of 80 rpm is 97.5%

SOLUTION

Unit Settings: SI C kPa kJ mass deg

$D = 0.272$	$\delta h_{shaft} = -7309$	$\varepsilon = 0.00006$
$\eta_{pump} = 0.9$	$\eta_{system} = 0.9751$	$f = 0.08128$
$g = 32.2$	$h_l = 153.5$	$h_{lm} = 28.54$
$K_{check} = 2$	$L = 36$	$\mu = 0.000004912$
$P_1 = 14.7$	$P_2 = 252$	$Q = 6.102$
$Re = 501.1$	$\rho = 1.787$	$speed = 1200$
$\overline{V} = 105$	$\dot{W}_{in} = 0.225$	$\dot{W}_{out} = 0.2194$

No unit problems were detected.

Right Rear Dropout Stiffening Member Calculations

Right Rear Dropout Bending

The right rear dropout has several key components attached. While it is a 3/16in plate of 4130 steel, there is a danger of bending the plate.

This bending force could come from a combination of the gear meshing forces and the riding forces on the frame

This calculation treats the rear plate as a cantilevered beam. The portion of the plate to which the motor mount is bolted is considered rigid.

The remaining length of plate is considered a cantelivered beam. Below is a simple estimation of the maximum force allowable if it were bending at the end of the plate (e.g. closest to the ground) as a point load. The calculation has been done twice, one for no stiffening member, and one for two stiffening members of 1 inch wide strips of the same material.

$w = 5.5$ [in] *width of plate*

$L = 4$ [in] *length of plate considered cantilevered*

$t = 0.1875$ [in] *thickness of the plate*

$a = 1$ [in] *height of stiffening strips*

$S_y = 85000$ [psi] *minimum yield strength of 4130 steel*

$I_1 = \frac{1}{12} \cdot w \cdot t^3$ *moment of inertia of plate*

I_2 and $y_{centroid}$ found with SolidWorks section area evaluation

$I_2 = 5.25$ [in4] *moment of inertia of plate with stiffening members*

$y_{centroid} = 0.25$ [in] *distance from top face of plate (to which motor mount bolts) to the neutral axis*

$$\frac{F_{max,1} \cdot L \cdot \frac{t}{2}}{I_1} = S_y$$ *stress calculation to get max point force at end of cantilever without stiffening*

$$\frac{F_{max,2} \cdot L \cdot [a + t - y_{centroid}]}{I_2} = S_y$$ *stress calculation to get max point force at end of cantilever with stiffening*

$$N_{stiff} = \frac{F_{max,2}}{F_{max,1}}$$ *stiffness ratio*

From the results it is clear that adding the stiffening members makes huge improvements to overall strength. While the plate may have been okay without them; it will certianly be okay with them, at a cost of an additional half pound of weight.

SOLUTION

Unit Settings: SI C kPa kJ mass deg

$a = 1$ [in]	$F_{max,1} = 684.8$ [lbf]	$F_{max,2} = 119000$ [lbf]
$I_1 = 0.003021$ [in^4]	$I_2 = 5.25$ [in^4]	$L = 4$ [in]
$N_{stiff} = 173.8$	$S_y = 85000$ [psi]	$t = 0.1875$ [in]
$w = 5.5$ [in]	$y_{centroid} = 0.25$ [in]	

No unit problems were detected.

Rear Shaft Sizing Calculations

Rear Shaft Sizing Calculation

See attached graphic for Free Body Diagram with axes

Loads and System Parameters

Torque into the Shaft in in-lbf, peak human input after gearing

$$T_{in} = 1200 \cdot \frac{4.375}{15}$$

Radius of the Wheel in inches

$$r_{wheel} = 26$$

Average Constant Acceleration, 0 to 20mph in 20s, in ft/s²

$$a_{ave} = 1.467$$

Maximum Motive Force (accelerating with max power)

$$F_{Motive} = \frac{T_{wheel}}{r_{wheel}}$$

Weight of the Bicycle in lbf, assume half of total bicycle weight is on rear

$$Weight = \frac{120}{2}$$

Sum of Forces and Moments, all in lbf and inches (and in-lbf)

Sum of the Torques (in z-dir)

$$T_{in} - T_{wheel} = 0$$

Sum of Forces in x-dir

$$F_{Motive} - R1_x - R2_x = \frac{Weight}{32.2} \cdot a_{ave}$$

Sum of Moments at D about y

$$R1_x \cdot 7 - F_{Motive} \cdot 3.5 = 0$$

Sum of Forces in y-dir

$$Weight - R1_y - R2_y = 0$$

Sum of Moments at D about x-axis

$$R1_y \cdot 7 - Weight \cdot 3.5 = 0$$

See attached Shear and Moment Diagrams

Alternating Moment at Critical Point (C) in in-lbf

$$M_a = \sqrt{[R1_x \cdot 3.5]^2 + [R1_y \cdot 3.5]^2}$$

Mean Torque at Critical Point (C) in in-lbf

$$T_m = T_{in}$$

Material Properties

Ultimate Strength of Steel in psi

$$S_{ut} = 58000$$

Yield Strength of Steel in psi

$$S_y = 36000$$

Nominal Endurance Strength of Steel in psi

$$S'_e = 0.5 \cdot S_{ut}$$

Endurance Strength Modifiers

$$k_a = a \cdot \left[\frac{S_{ut}}{1000}\right]^b$$

$$a = 2.7$$

$$b = -0.265$$

$$k_b = 1.24 \cdot d^{-0.107}$$

$$d = 17$$

$$k_e = 0.814$$

Endurance Strength in psi, Modified

$$S_e = S'_e \cdot k_a \cdot k_b \cdot k_e$$

Shaft Diameter Size Calculations in inches

$n = 1$ *Factor of Safety, set to 1 for sizing estimates*

$K_f = 1$ *No stress concentration*

$K_{fs} = 1$ *No stress concentration*

$M_m = 0$ *No mean moment*

$T_a = 0$ *No alternating torque*

DE-Goodman

$$d_{Goodman} = \left[\frac{16 \cdot n}{\pi} \cdot \left(\frac{1}{S_e} \cdot [4 \cdot (K_f \cdot M_a)^2 + 3 \cdot (K_{fs} \cdot T_a)^2]^{[1/2]} + \frac{1}{S_{ut}} \cdot [4 \cdot (K_f \cdot M_m)^2 + 3 \cdot (K_{fs}\right.\right.$$

$$\cdot T_m)^2\Big]^{[1/2]}\Big)\Big]^{[1/3]}$$

DE-Gerber

$$d_{Gerber} = \left[\frac{8 \cdot n \cdot A_{Gerber}}{\pi \cdot S_e} \cdot \left(1 + \left[1 + \left(\frac{2 \cdot B_{Gerber} \cdot S_e}{A_{Gerber} \cdot S_{ut}}\right)^2\right]^{[1/2]}\right)\right]^{[1/3]}$$

$$A_{Gerber} = \sqrt{4 \cdot [K_f \cdot M_a]^2 + 3 \cdot [K_{fs} \cdot T_a]^2}$$

$$B_{Gerber} = \sqrt{4 \cdot [K_f \cdot M_m]^2 + 3 \cdot [K_{fs} \cdot T_m]^2}$$

DE-ASME Elliptic

$$d_{elliptic} = \left[\frac{16 \cdot n}{\pi} \cdot \left(4 \cdot \left[\frac{K_f \cdot M_a}{S_e}\right]^2 + 3 \cdot \left[\frac{K_{fs} \cdot T_a}{S_e}\right]^2 + 4 \cdot \left[\frac{K_f \cdot M_m}{S_y}\right]^2 + 3 \cdot \left[\frac{K_{fs} \cdot T_m}{S_e}\right]^2\right)^{(1/2)}\right]^{[1/3]}$$

DE-Soderberg

$$d_{Soderberg} = \left[\frac{16 \cdot n}{\pi} \cdot \left(\frac{1}{S_e} \cdot \left[4 \cdot (K_f \cdot M_a)^2 + 3 \cdot (K_{fs} \cdot T_a)^2\right]^{[1/2]} + \frac{1}{S_y} \cdot \left[4 \cdot (K_f \cdot M_m)^2 + 3 \cdot (K_{fs} \cdot T_m)^2\right]^{[1/2]}\right)\right]^{[1/3]}$$

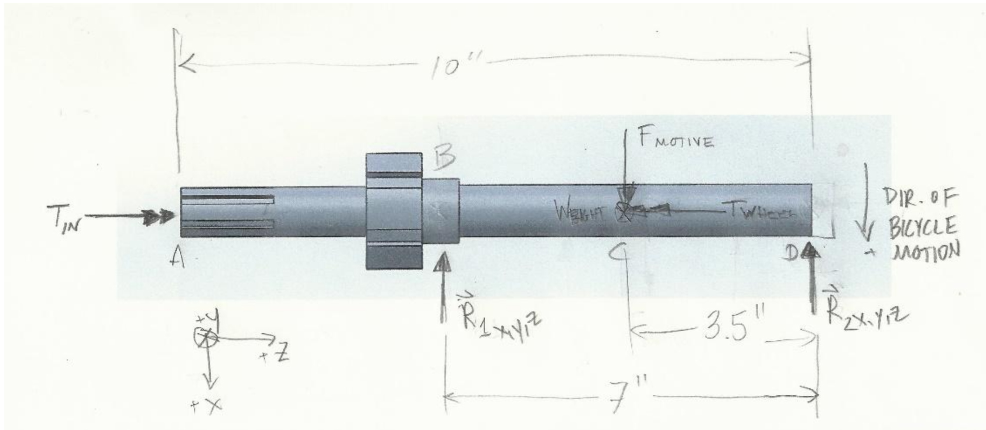
These results suggest a shaft sizing above a range of 0.44 to 0.55 inches. The current shaft from the XR80 clutch is 17mm (.67 in) in diameter.

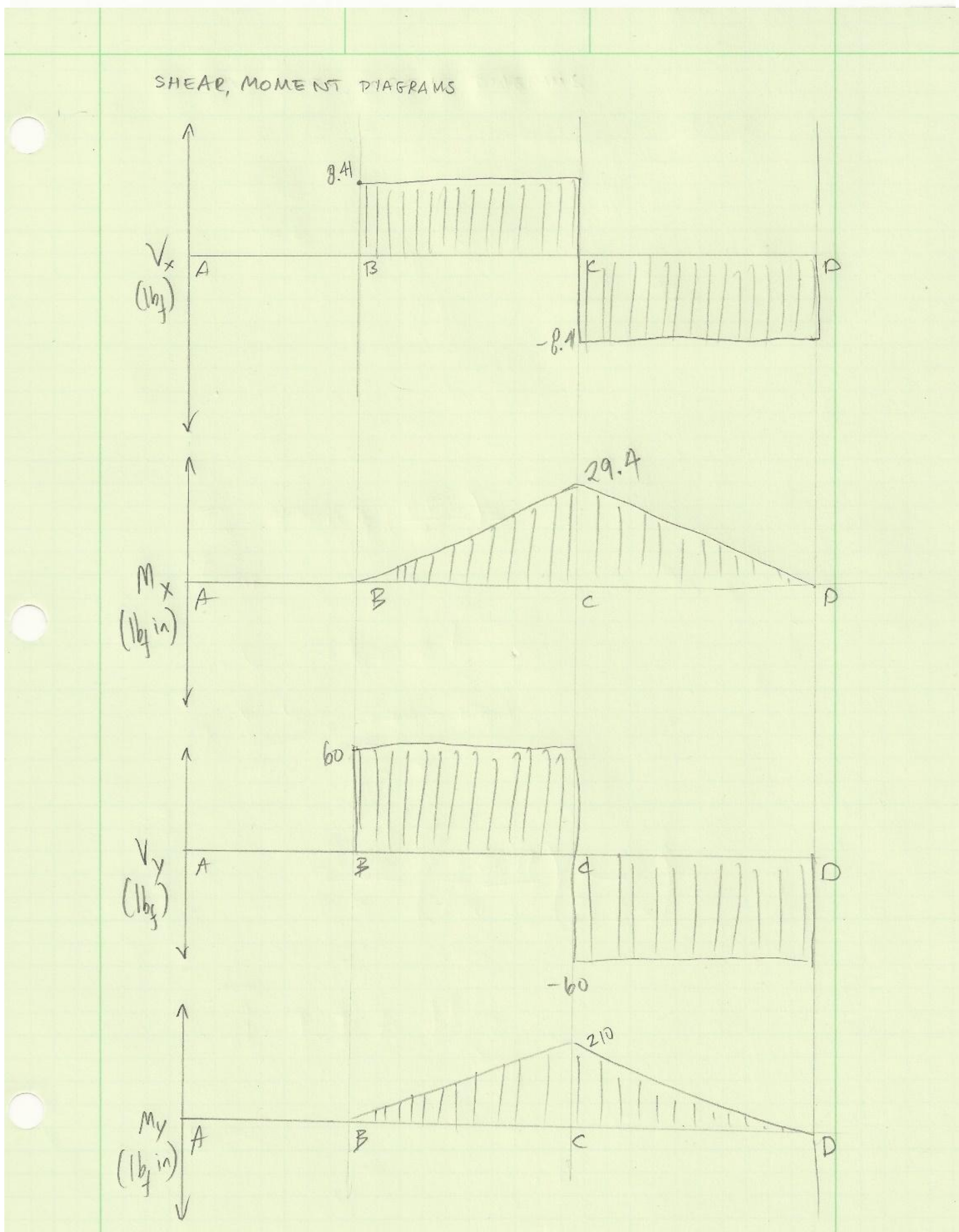
SOLUTION

Unit Settings: SI C kPa kJ mass deg

$a = 2.7$	$a_{ave} = 1.467$	$A_{Gerber} = 215.2$
$b = -0.265$	$B_{Gerber} = 606.2$	$d = 17$
$d_{elliptic} = 0.5481$	$d_{Gerber} = 0.4439$	$d_{Goodman} = 0.4767$
$d_{Soderberg} = 0.5203$	$F_{Motive} = 13.46$	$k_a = 0.9206$
$k_b = 0.9157$	$k_e = 0.814$	$K_f = 1$
$K_{fs} = 1$	$M_a = 107.6$	$M_m = 0$
$n = 1$	$R1_x = 6.731$	$R1_y = 30$
$R2_x = 3.997$	$R2_y = 30$	$r_{wheel} = 26$
$S_e = 19899$	$S'_e = 29000$	$S_{ut} = 58000$
$S_y = 36000$	$T_a = 0$	$T_{in} = 350$
$T_m = 350$	$T_{wheel} = 350$	Weight = 60

No unit problems were detected.


10"
F MOTIVE
T IN
WEIGHT
T WHEEL
DIR. OF BICYCLE MOTION
A
B
C
D
R1 x,y,z
R2 x,y,z
3.5"
7"
+y
+Z
+X


SHEAR, MOMENT DIAGRAMS
Vx (lbf)
8.4
-8.4
Mx (lbf in)
29.4
Vy (lbf)
60
-60
My (lbf in)
210
A
B
C
D

Rear Gear and Pinion Sizing Calculations

Gear Sizing Calculation

This sizing calculation is for the rear gears based on AGMA gear strength equations in Shigley's Mechanical Engineering Design

The loading case is based on the peak 1200 in-lbf input at the cranks, which goes through a 15:1 speed increse before befor being transferred through the fluid to the pinion attached to the motor output shaft

Parametric table summarizes results. P is pitch. F is face width. d_G is gear diameter. d_P is pinion diameter. S_F is pinion factor of safety against tooth bending stress. S_H is pinion factor of safety against contact stress. It is generally assumed that the gear is safe from bending and contact stress if the pinion is, since it sees lower loads

Minimum number of pinion teeth, comes out to 15.88, so 16 in minimum

$$N_{P,min} = \frac{2 \cdot k}{[1 + 2 \cdot m] \cdot \sin^2[PA]} \cdot \left[m + \sqrt{m^2 + (1 + 2 \cdot m) \cdot \sin^2(PA)} \right]$$

$k = 1$ *For full-depth teeth*

$PA = 14.5$ [deg] *Pressure angle*

$m = \frac{N_G}{N_P}$ *Gear Ratio*

Geometry

Common Diametral Pitches include 48, 32, 24, 20, 18, 16, 12, and 8 teeth/in

teeth/in

$N_P = 16$ *number of teeth of pinion*

$N_G = 90$ *number of teeth of gear*

Pitch Definitions

$P = \frac{N_G}{d_G}$ *diameter in in*

$P = \frac{N_P}{d_P}$

in

Lewis Form factor

$Y_P = 0.296$

$Y_G = 0.44$

Loading

$T = \frac{1200}{15}$ *in-lbf*

$$W_t = \frac{T}{\frac{d_P}{2}}$$ *lbf*

Pinion Bending Stress

$$\sigma_P = W_t \cdot K_{o,P} \cdot K_{v,P} \cdot K_{s,P} \cdot \frac{P}{F} \cdot \frac{K_{m,P} \cdot K_{B,P}}{J_P}$$

$K_{o,P} = 1.5$ *for medium shock source, uniform machine*

$K_{v,P} = 1$ *slow pitch line velocity*

$$K_{s,P} = 1.192 \cdot \left[\frac{F \cdot \sqrt{Y_P}}{P}\right]^{0.0535}$$

$$K_{m,P} = 1 + C_{mc} \cdot \left[C_{pf} \cdot C_{pm} + C_{ma} \cdot C_e\right]$$

$C_{mc} = 1$ *for uncrowned teeth*

$$C_{pf} = \frac{F}{10 \cdot d_P} - 0.025$$ *for F<1in*

$C_{pm} = 1.1$ *for off-center gear more than 18%*

$C_{ma} = 0.25$ *for open gearing, F = 1in*

$C_e = 1$ *for all other conditions (not adjusted at assembly)*

$K_{B,P} = 1$ *for not worrying about rim thickness*

$J_P = 0.27$

$$\sigma_{all} = \frac{S_t}{S_F} \cdot \frac{Y_N}{K_T \cdot K_R}$$

$S_t = 40000$ *psi for through-hardened steel, Brinell Hardness =~275*

$Y_N = 1$ *for 10^7 cycles*

$K_T = 1$ *for less than 250 degrees F*

$K_R = 1$ *for 0.99 reliability*

$$\frac{\sigma_{all}}{\sigma_P} = S_F$$

Pinion Contact Stress

$$\sigma_{c,P} = C_{p,P} \cdot \sqrt{W_t \cdot K_{o,P} \cdot K_{v,P} \cdot K_{s,P} \cdot \frac{K_{m,P}}{d_P \cdot F} \cdot \frac{C_{f,P}}{I_P}}$$

$C_{p,P} = 2300$ *sqrt(psi) for steel on steel*

$C_{f,P} = 1$ *for unknown, but satisfactory surface conditions*

$$I_P = \frac{\cos[20] \cdot \sin[20]}{2 \cdot 1} \cdot \left[\frac{m}{m+1}\right]$$

$$\sigma_{c,all} = \frac{S_c}{S_H} \cdot \frac{Z_N \cdot C_H}{K_T \cdot K_R}$$

S_c = 130000 *psi for through-hardened steel, Brinell Hardness =~275*

Z_N = 1 *for 10^7 cycles*

C_H = 1 *for pinion*

$$S_H = \frac{\sigma_{c,all}}{\sigma_{c,P}}$$

Parametric Table: Table 1

	P [teeth/in]	F [in]	d_G [in]	d_P [in]	S_F	S_H
Run 1	8	0.5	11.25	2	2.127	1.307
Run 2	10	0.5	9	1.6	1.707	1.171
Run 3	12	0.5	7.5	1.333	1.426	1.07
Run 4	16	0.5	5.625	1	1.072	0.928
Run 5	18	0.5	5	0.8889	0.953	0.8751
Run 6	8	0.75	11.25	2	2.563	1.435
Run 7	10	0.75	9	1.6	2.054	1.285
Run 8	12	0.75	7.5	1.333	1.713	1.173
Run 9	16	0.75	5.625	1	1.285	1.016
Run 10	18	0.75	5	0.8889	1.141	0.9575
Run 11	20	0.75	4.5	0.8	1.026	0.9079
Run 12	24	0.75	3.75	0.6667	0.8524	0.8276
Run 13	8	1	11.25	2	2.921	1.532
Run 14	10	1	9	1.6	2.338	1.371
Run 15	12	1	7.5	1.333	1.948	1.251
Run 16	16	1	5.625	1	1.457	1.082
Run 17	18	1	5	0.8889	1.292	1.019
Run 18	20	1	4.5	0.8	1.16	0.9656
Run 19	24	1	3.75	0.6667	0.962	0.8793
Run 20	8	1.5	11.25	2	3.501	1.677
Run 21	10	1.5	9	1.6	2.796	1.499
Run 22	12	1.5	7.5	1.333	2.323	1.366
Run 23	16	1.5	5.625	1	1.729	1.179
Run 24	18	1.5	5	0.8889	1.531	1.109
Run 25	20	1.5	4.5	0.8	1.372	1.05
Run 26	24	1.5	3.75	0.6667	1.132	0.9539

The gear/pinion pair that corresponds with Run 8 (12P, ¾" face width) in the parametric table above is the final choice for the design.

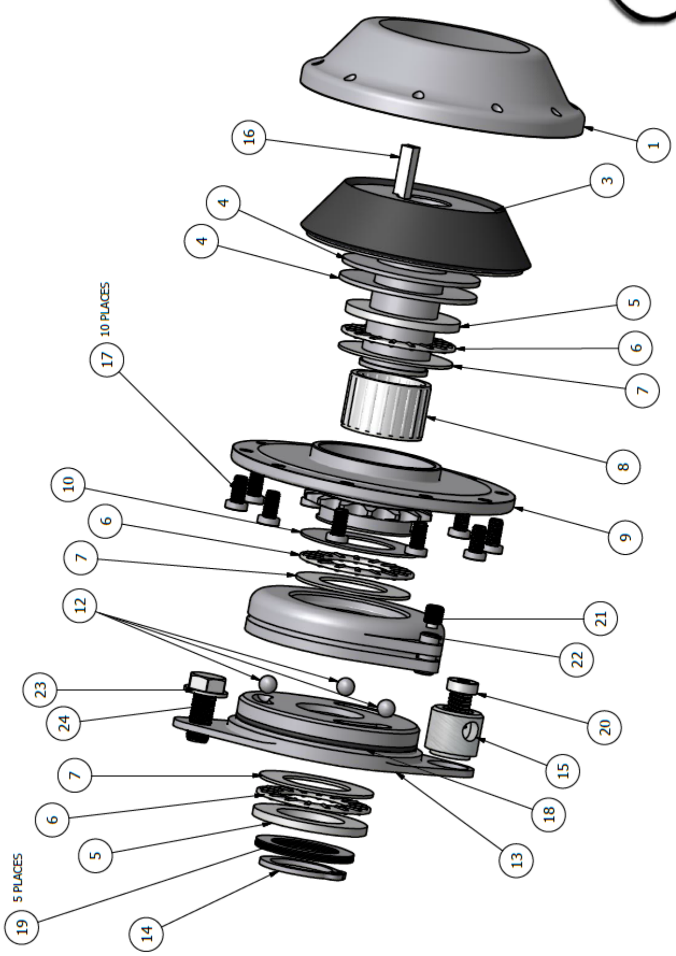
14. Appendix E: Manufacturer Catalog Information

Clutch (Downs Brothers Racing)

PARTS LIST

ITEM	QTY	DESCRIPTION
1	1	CLUTCH DRUM
3	1	CLUTCH DRIVER
4	1	BELLEVILLE DISC SPRING
5	2	THICK BEARING RACEWAY
6	3	THRUST BEARING
7	3	THIN BEARING RACEWAY
8	1	NEEDLE BEARING
9	1	SPROCKET FLANGE
10	1	OVERSIZE THRUST BEARING RACEWAY
11	1	ROTATING ACTUATOR
12	3	BEARING BALL
13	1	FIXED ACTUATOR
14	1	SNAP RING
15	1	CABLE HOUSING MOUNT
16	1	KEY
17	10	LOW HEAD CAP SCREW
18	1	O-RING
19	5	SHIM
20	1	LOW HEAD CAP SCREW
21	1	SET SCREW
22	1	CABLE STOP
23	1	WASHER
24	1	5/16-24 HEX BOLT

Downs Brothers Racing
OldMiniBikes.com



Planetary gear set (Harmonic Drive LLC)

Harmonic Drive LLC | 247 Lynnfield St. Peabody, MA 01960 | 800-921-3332
Harmonic Drive and Harmonic Planetary are registered trademarks of Harmonic Drive LLC.

HPG-14A-5

Motor selection

Motor Manufacturer **Motor Model Number** **Motor Bus Voltage**

Although the motor and gear data used on this site is believed to be accurate, no warranty is expressed or implied regarding the accuracy, adequacy, completeness, reliability or usefulness of this information. Additionally, this software matches the mechanical performance characteristics of the selected motors and gears. Not all matches represented here may be suitable for your intended application. Please confirm all specifications and performance calculations before ordering any product. Contact our engineering department if you should need any assistance.

Series	Size	Version	Ratio	Backlash	Input Bearing	Output Configuration	Motor Code
HPG	14	A	05				

- High Efficiency
- Low Backlash
- High load capacity output bearing
- Quick Connect™ coupling

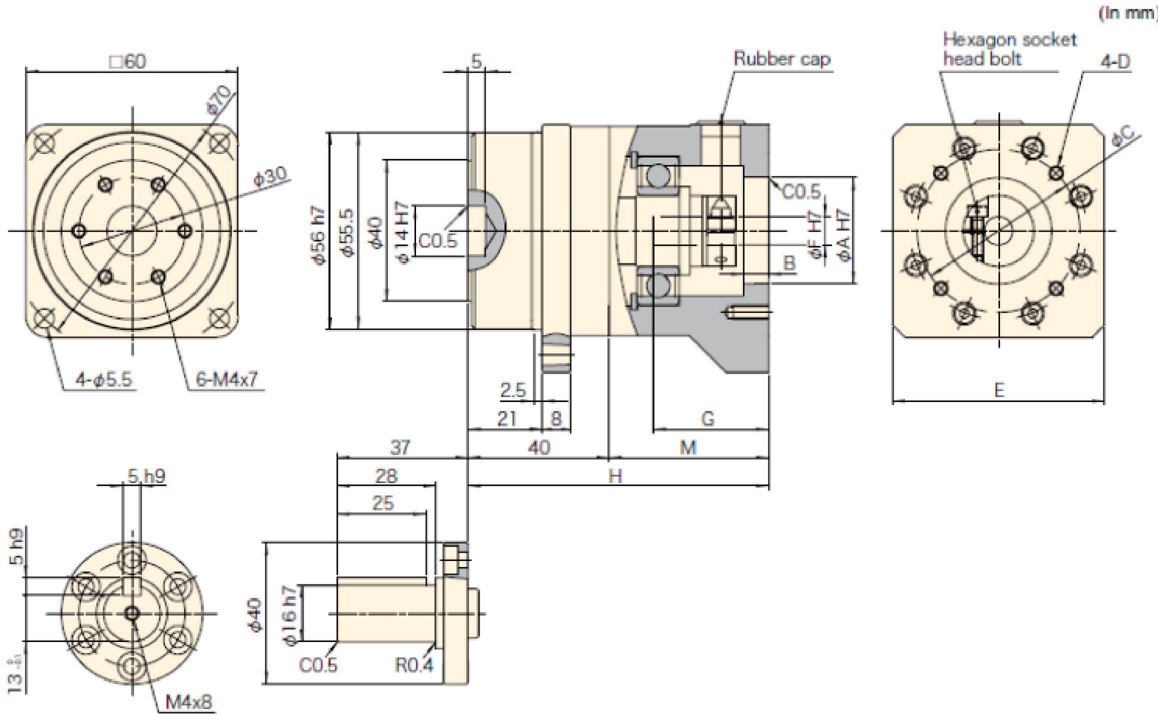
Gearhead Model Number

Gearhead Performance Data		
Rated Torque L10	5.9	Nm
Rated Torque L50	11	Nm
Limit for Average Torque	13	Nm
Limit for Repeated Peak Torque	30	Nm
Limit for Momentary Peak Torque	56	Nm
Starting Torque	8.6	Ncm
Backdriving Torque	0.43	Nm
Max. Input Speed	6,000	rpm

Gearhead Performance Data		
Ave. Input Speed	3,000	rpm
Standard Accuracy	4	arc min
Gear Backlash	< 3	arc min
Input Inertia	0.02	kg-cm^2

Torsional Stiffness		
K1	0.47	10^4 Nm / rad

Output Bearing		
Pitch Circle Diameter (dp)	0.04	m
Offset (R)	0.01	m
Basic Dynamic Rated Load (C)	51.1	10^2 N
Basic Static Rated Load (Co)	70.8	10^2 N
Allowable Moment Load (Mc)	32.3	Nm
Moment Stiffness (Km)	3	10^4 Nm / rad
Permissible Radial Load	470	N
Permissible Axial Load	700	N



HPG-14A

Note: Dimensions A, B, C, D, E, F, G, H depends on the chosen motor/adapter flange combination. Contact Harmonic Drive LLC for details.

Harmonic Drive LLC
Tel: 800-921-3332
www.harmonicdrive.net

Harmonic Drive LLC
800-921-3332
www.harmonicdrive.net

Pinion gear in rear drive train (McMaster-Carr)

14 1/2° Pressure Angle Spur Gears

About Gears

To work together, gears must have the same **pitch** and **pressure angle** (the angle between contact points of meshing teeth). A gear's pitch is its number of teeth divided by its pitch diameter. For example, a gear with 16 teeth and a pitch diameter of 1/2" will have a pitch of 32. For higher load capacities, choose 20° pressure angle gears over 14 1/2° gears.

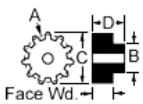
Use gauges to measure gear pitch and pressure angle.

Order 2069A67 for 14 1/2° Pressure Angle Gauge Set.

Order 2069A12 for 14 1/2° and 20° Pressure Angle Gauge Set.

Steel Plain-Bore 14 1/2° Pressure Angle Spur Gears and Racks


Gear on Rack



Spur Gears—Have a plain bore furnished in the minimum size listed and are machinable up to the maximum size listed. Made of 1144 steel.

Gear Racks and **Ready-Mount Gear Racks**—All are made of C1018 steel.

Note: Choose spur gears and racks with the same pressure angle and pitch. The 14 1/2° and 20° pressure angle spur gears and racks are not compatible.

Spur Gears

No. of Teeth	Pitch Dia. (A)	Hub Dia. (B)	OD (C)	O'all Lg. (D)	Bore Size Min.	Bore Size Max.		Each
32 Pitch—3/16" Face Width								
16	0.500"	13/32"	0.56"	1/2"	3/16"	3/8"	6325K89	$15.30
18	0.562"	13/32"	0.63"	1/2"	3/16"	3/8"	6325K94	15.30
20	0.625"	17/32"	0.69"	1/2"	1/4"	7/16"	6325K95	16.74
24	0.750"	21/32"	0.81"	1/2"	5/16"	1/2"	6325K96	20.85
40	1.250"	7/8"	1.31"	9/16"	3/8"	1/2"	6325K97	31.40
24 Pitch—1/4" Face Width								
12	0.500"	3/8"	0.58"	9/16"	1/4"	1/4"	6325K31	16.16
15	0.625"	1/2"	0.71"	9/16"	1/4"	1/4"	6325K63	16.82
16	0.666"	35/64"	0.75"	9/16"	5/16"	5/16"	6325K75	17.21
20	0.833"	23/32"	0.92"	9/16"	5/16"	5/16"	6325K87	21.09
21	0.875"	3/4"	0.96"	9/16"	3/8"	3/8"	6325K64	21.57

No. of Teeth	Pitch Dia. (A)	Hub Dia. (B)	OD (C)	O'all Lg. (D)	Bore Size Min.	Bore Size Max.		Each
12 Pitch—3/4" Face Width								
12	1.000"	3/4"	1.17"	1 1/4"	1/2"	1/2"	6325K22	$21.57
14	1.167"	29/32"	1.33"	1 1/4"	1/2"	1/2"	6325K23	24.39
15	1.250"	1"	1.42"	1 1/4"	5/8"	5/8"	6325K79	27.03
16	1.333"	1 1/16"	1.50"	1 1/4"	5/8"	5/8"	**6325K24**	28.02

Product Detail

Steel Plain Bore 14-1/2 Degree Spur Gear, 12 Pitch, 16 Teeth, 1.333" Pitch Diameter, 5/8" Bore

Each

ADD TO ORDER

You ordered 1 each on 12/04/14.

In stock

Raw stock for motor mount (McMaster-Carr)

Multipurpose 6061 Aluminum

Rectangular Bars—Unpolished

Wd.	Bars	1/2 ft.	1 ft.	2 ft.	3 ft.	6 ft.	Bars with Certification	1/2 ft.	1 ft.	2 ft.	3 ft.	6 ft.
1 3/4"	9008K51	14.43	22.67	42.25	58.74	103.05	9872T219	24.43	32.67	52.25	68.74	113.05
2" Thick (±0.024")												
2"	9008K53	16.09	25.28	47.12	65.50	114.92	9872T221	26.09	35.28	57.12	75.50	124.92
2 1/2"	8975K262	31.31	49.20	91.69	127.47	223.63	9872T222	41.31	59.20	101.69	137.47	233.63
3"	8975K264	33.55	52.72	98.24	136.58	239.62	9872T223	43.55	62.72	108.24	146.58	249.62
3 1/2"	8975K629	36.42	57.22	106.65	148.26	260.11	9872T224	46.42	67.22	116.65	158.26	270.11
4"	8975K266	45.67	71.77	133.75	185.94	326.21	9872T225	55.67	81.77	143.75	195.94	336.21
5"	8975K631	50.11	78.75	146.76	204.03	357.95	9872T226	60.11	88.75	156.76	214.03	367.95
6"	8975K268	—	100.36	187.04	260.03	456.19	9872T227	64.29	110.36	197.04	270.03	466.19
2 1/4" Thick (±0.024")												
2 1/4"	9008K55	23.08	36.28	67.60	93.99	164.89	9872T228	33.08	46.28	77.60	103.99	174.89
2 1/2" Thick (±0.024")												
2 1/2"	9008K57	27.25	42.83	79.81	110.96	194.66	9872T229	37.25	52.83	89.81	120.96	204.66
3"	8975K271	51.00	80.14	149.36	207.65	364.29	9872T231	61.00	90.14	159.36	217.65	374.29
3 1/2"	8975K273	55.94	87.91	163.83	227.77	399.59	9872T232	65.94	97.91	173.83	237.77	409.59
4"	8975K275	61.41	96.50	179.83	250.01	438.62	9872T233	71.41	106.50	189.83	260.01	448.62
5"	8975K277	71.36	112.13	208.97	290.52	509.69	9872T234	81.36	122.13	218.97	300.52	519.69
2 3/4" Thick (±0.024")												
2 3/4"	9008K59	32.23	50.65	94.39	131.22	230.21	9872T235	42.23	60.65	104.39	141.22	240.21
3" Thick (±0.024")												
3"	9008K62	35.56	55.88	104.13	144.77	253.98	9872T236	45.56	65.88	114.13	154.77	263.98
3 1/2"	8975K279	62.15	97.66	182.01	253.04	443.93	9872T237	72.15	107.66	192.01	263.04	453.93
4"	8975K282	66.07	103.83	193.50	269.01	471.94	9872T238	76.07	113.83	203.50	279.01	481.94
4 1/2"	8975K284	74.47	117.03	218.10	303.22	531.96	9872T239	84.47	127.03	228.10	313.22	541.96
5"	8975K286	82.58	129.76	241.83	336.21	589.84	9872T241	92.58	139.76	251.83	346.21	599.84
6"	**8975K288**	95.89	150.69	280.83	390.43	684.96	9872T242	105.89	160.69	290.83	400.43	694.96

Product Detail

Each

ADD TO ORDER

Multipurpose 6061 Aluminum, 3" Thick x 6" Width

Length
- 1/2 ft.
- 1 ft.
- 2 ft.
- 3 ft.
- 6 ft.

You ordered 1 each on 01/21/15.

Raw stock for brake bridge (McMaster-Carr)

Easy-to-Weld 4130 Alloy Steel

Round Tubes—Unpolished (Cold Drawn)

- Yield Strength: 70,000 psi
- Hardness: Hard (Rockwell C19)
- Can be hardened up to Rockwell C49

Meet AMS-T-6736 and MIL-T-6736B. Straightness tolerance is 0.030" per 3 feet. Length tolerance is ±1".

OD	ID	OD Tolerance		1 ft.	3 ft.	6 ft.
0.028" Wall Thick. (±0.004")						
0.250"	0.194"	±0.005"	89955K439	$5.87	$13.61	$23.46
0.375"	0.319"	±0.005"	89955K469	5.94	13.77	23.75
0.500"	0.444"	±0.005"	89955K479	5.98	13.88	23.93
0.625"	0.569"	±0.005"	89955K489	6.71	15.57	26.84
0.750"	0.694"	±0.005"	89955K438	6.53	15.14	26.11
1.000"	0.944"	±0.010"	89955K499	8.64	20.04	34.56
0.035" Wall Thick. (±0.005")						
0.188"	0.118"	±0.005"	89955K119	5.79	13.42	23.14
0.250"	0.180"	±0.005"	89955K219	5.65	13.11	22.60
0.313"	0.243"	±0.005"	89955K239	6.35	14.73	25.40
0.375"	0.305"	±0.005"	89955K149	6.35	14.74	25.41
0.500"	0.430"	±0.005"	89955K259	6.53	15.16	26.13
0.625"	0.555"	±0.005"	89955K279	6.58	15.28	26.34
0.750"	0.680"	±0.005"	89955K299	6.80	15.78	27.21
0.875"	0.805"	±0.005"	89955K849	7.91	18.36	31.65
1.000"	0.930"	±0.005"	89955K339	8.20	19.02	32.79
1.125"	1.055"	±0.005"	89955K919	10.06	23.35	40.26
1.250"	1.180"	±0.011"	89955K369	10.36	24.02	41.42
1.375"	1.305"	±0.012"	89955K979	11.40	26.45	45.61
1.500"	1.430"	±0.013"	89955K389	13.23	30.70	52.93
0.049" Wall Thick. (±0.007")						
0.250"	0.152"	±0.005"	89955K229	6.08	14.09	24.30
0.313"	0.215"	±0.005"	89955K249	6.40	14.85	25.60
0.375"	0.277"	±0.005"	89955K159	6.18	14.35	24.74
0.438"	0.340"	±0.005"	89955K739	6.61	15.34	26.44
0.500"	0.402"	±0.005"	**89955K269**	6.46	14.98	25.83

Product Detail

Easy-to-Weld 4130 Alloy Steel Round Tube, .500" OD, .049" Wall Thickness

Each

ADD TO ORDER

Length

1 ft.

3 ft.

6 ft.

You ordered 1 each on 02/05/15.

OD	ID	OD Tolerance		1 ft.	3 ft.	6 ft.
0.065" Wall Thick. (±0.010")						
0.250"	0.120"	±0.005"	89955K519	$6.13	$14.23	$24.53
0.313"	0.183"	±0.005"	89955K539	6.33	14.69	25.32
0.375"	0.245"	±0.005"	89955K449	6.51	15.11	26.06
0.438"	0.308"	±0.005"	89955K749	7.06	16.38	28.24
0.500"	0.370"	±0.005"	89955K559	7.39	17.14	29.56
0.563"	0.433"	±0.005"	89955K789	7.60	17.63	30.40
0.625"	0.495"	±0.005"	89955K579	7.66	17.77	30.63
0.750"	0.620"	±0.005"	89955K599	8.07	18.73	32.30
0.875"	0.745"	±0.005"	89955K869	8.12	18.83	32.47
1.000"	0.870"	±0.005"	89955K639	8.62	20.00	34.48
1.125"	0.995"	±0.005"	89955K939	11.61	26.93	46.43
1.250"	1.120"	±0.005"	89955K659	12.96	30.06	51.82
1.375"	1.245"	±0.005"	89955K129	13.96	32.38	55.83
1.500"	1.370"	±0.005"	89955K679	15.09	35.01	60.36
1.625"	1.495"	±0.010"	89955K248	16.78	38.93	67.12
1.750"	1.620"	±0.010"	89955K199	17.27	40.07	69.09
2.000"	1.870"	±0.010"	89955K699	20.05	46.53	80.22
0.083" Wall Thick. (±0.012")						
0.375"	0.209"	±0.005"	89955K459	7.93	18.39	31.71
0.438"	0.272"	±0.005"	89955K759	8.72	20.22	34.87
0.500"	0.334"	±0.005"	89955K569	9.08	21.06	36.31
0.563"	0.397"	±0.005"	89955K799	9.65	22.40	38.62
0.625"	0.459"	±0.005"	89955K589	9.36	21.70	37.42
0.750"	0.584"	±0.005"	89955K629	9.38	21.75	37.50
0.875"	0.709"	±0.005"	89955K879	10.39	24.10	41.55
1.000"	0.834"	±0.005"	89955K649	10.39	24.10	41.55
1.125"	0.959"	±0.005"	89955K949	11.99	27.83	47.98
1.250"	1.084"	±0.005"	89955K669	14.38	33.37	57.54
1.375"	1.209"	±0.005"	89955K139	15.00	34.80	60.00
1.500"	1.334"	±0.005"	89955K689	16.82	39.02	67.28
1.625"	1.459"	±0.010"	89955K258	18.85	43.73	75.40
1.750"	1.584"	±0.010"	89955K319	19.70	45.72	78.82
2.000"	1.834"	±0.010"	89955K719	19.77	45.87	79.09
0.095" Wall Thick. (±0.014")						
0.313"	0.123"	±0.005"	89955K268	9.33	21.65	37.32
0.438"	0.248"	±0.005"	89955K278	10.04	23.29	40.16
0.500"	0.310"	±0.005"	89955K288	14.32	33.23	57.29
0.750"	0.560"	±0.005"	89955K298	15.17	35.19	60.67
1.000"	0.810"	±0.005"	89955K318	16.05	37.23	64.19

Raw stock for rear shaft spacers (McMaster-Carr)

Multipurpose 6061 Aluminum

Round Tubes—Unpolished

- Yield Strength: 35,000 psi
- Hardness: Soft (95 Brinell)
- Temper: Heat Treated (T6511, unless noted)
- Meet ASTM B241, unless noted

OD tolerance for 1/4" to 7/8" dia. tubes is ±0.010". OD tolerance for 1" to 6 1/2" dia. tubes is ±0.035". Straightness tolerance for 1/4" dia. tubes is 0.500" per foot. Straightness tolerance for 3/8" to 6 1/2" dia. tubes is 0.020" per foot. Length tolerance for 1/2-ft. lengths is ±1/2". Length tolerance for 1-ft. to 6-ft. lengths is ±1".

OD	ID		1/2 ft.	1 ft.	2 ft.	3 ft.	6 ft.
0.035" Wall Thick. (±0.006")							
1/4"	0.180"	9056K61 ‡	$3.08	$5.13	$9.24	$12.11	$20.53
3/8"	0.305"	9056K63 ‡	3.59	5.99	10.78	14.13	23.95
1/2"	0.430"	9056K92 ‡	3.93	6.55	11.79	15.46	26.21
5/8"	0.555"	9056K91 ‡	4.57	7.61	13.70	17.96	30.44
7/8"	0.805"	9056K72 ‡	4.90	8.16	14.69	19.26	32.65
1"	0.930"	9056K89 ‡	5.30	8.83	15.90	20.84	35.33
0.049" Wall Thick. (±0.008")							
1/4"	0.152"	9056K62 ‡	3.67	6.12	11.01	14.44	24.47
3/8"	0.277"	9056K64 ‡	4.00	6.67	12.01	15.75	26.69
1/2"	0.402"	9056K65 ‡	4.30	7.17	12.91	16.93	28.69
3/4"	0.652"	9056K69 ‡	4.69	7.82	14.07	18.44	31.26
1"	0.902"	9056K74 ‡	5.51	9.18	16.53	21.67	36.73
1 1/2"	1.402"	9056K78 ‡	8.00	13.33	24.00	31.46	53.33
2"	1.902"	9056K82 ‡	13.02	21.70	39.07	51.22	86.82
0.058" Wall Thick. (±0.007")							
5/8"	0.509"	9056K67 ‡	5.89	9.82	17.68	23.19	39.30
0.065" Wall Thick. (±0.009")							
1/2"	0.370"	9056K66 ‡	4.70	7.83	14.09	18.47	31.30
5/8"	0.495"	9056K68 ‡	5.03	8.38	15.09	19.79	33.54
3/4"	0.620"	9056K71 ‡	5.47	9.11	16.40	21.51	36.45
7/8"	0.745"	9056K73 ‡	5.71	9.52	17.13	22.46	38.07
1"	0.870"	9056K75 ‡	5.81	9.68	17.42	22.84	38.71
1 1/4"	1.120"	9056K76 ‡	6.74	11.23	20.21	26.50	44.91
1 1/2"	1.370"	9056K79 ‡	9.35	15.58	28.05	36.77	62.33
1 3/4"	1.620"	9056K81 ‡	8.44	14.07	25.32	33.20	56.27
2"	1.870"	9056K83 ‡	9.53	15.88	28.59	37.49	63.54
2 1/2"	2.370"	9056K84 ‡	12.26	20.43	36.77	48.21	81.72
3"	2.870"	9056K85 ‡	20.96	34.93	62.87	82.43	139.72

‡ Temper is T6. Meet ASTM B210.
° Temper is T6. Meet ASTM B221.
⌂ Meet ASTM B429.
⊥ Temper is T6.

OD	ID		1/2 ft.	1 ft.	2 ft.	3 ft.	6 ft.
0.083" Wall Thick. (±0.009")							
1 1/4"	1.084"	9056K77 ‡	$9.61	$16.02	$28.84	$37.82	$64.10
0.125" Wall Thick. (±0.020")							
3/4"	1/2"	9056K33 ‡	3.58	5.97	10.75	14.09	23.88
1"	3/4"	**9056K36 ‡**	4.67	7.78	14.00	18.35	31.11

Product Detail

Multipurpose 6061 Aluminum Tube, 1" OD, .125" Wall Thickness

Each

ADD TO ORDER

Length, ft.

1/2

1

2

3

6

You ordered this on 03/04/15.

OD	ID		1/2 ft.	1 ft.	2 ft.	3 ft.	6 ft.
1 1/4"	1"	9056K37 ‡	5.61	9.36	16.84	22.08	37.43
1 1/2"	1 1/4"	9056K38 °	7.24	12.06	21.71	28.47	48.25
1 3/4"	1 1/2"	9056K39 ‡	8.38	13.96	25.14	32.96	55.86
2"	1 3/4"	9056K4 ‡	9.34	15.57	28.04	36.76	62.30
3"	2 3/4"	9056K41 ‡	12.64	21.07	37.92	49.71	84.26
4"	3 3/4"	9056K42 ‡	16.11	26.85	48.33	63.36	107.39
0.250" Wall Thick. (±0.025")							
1"	1/2"	9056K28 ⌂	5.70	9.50	17.10	22.42	38.00
1 1/4"	3/4"	9056K29 ⌂	7.58	12.63	22.74	29.82	50.54
1 1/2"	1"	9056K27	11.62	19.37	34.87	45.72	77.49
2"	1 1/2"	9056K13	16.00	26.66	47.99	62.92	106.64
2 1/4"	1 3/4"	9056K16	19.35	32.24	58.04	76.09	128.97
2 1/2"	2"	9056K18	21.13	35.22	63.39	83.11	140.86

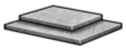
Raw stock for L-Bracket and rear dropouts, .190” thick, (McMaster-Carr)

Easy-to-Weld 4130 Alloy Steel

Its carbon content is low enough for good weldability but high enough to give this steel abrasion and impact resistance. 4130 is often used for gears, fasteners, and structural applications.

Warning: Physical, mechanical, and chemical properties are not guaranteed and are intended only as a basis for comparison.

Sheets with Material Certification—Unpolished



- Yield Strength: 50,000 psi
- Hardness: Medium (up to Rockwell B85)
- Can be hardened up to Rockwell C60

Meet AMS 6350-6351. Each sheet comes with a traceable lot number and material test report. All sheets are annealed. 0.025" to 0.125" thick sheets are cold rolled. 0.160" and thicker are hot rolled. Width and length tolerances are ±1/16".

		6" × 12"		6" × 36"		12" × 12"		12" × 24"		24" × 24"		36" × 36"	
Thick.	Thick. Tolerance		Each		Each		Each		Each		Each		Each
0.025"	±0.005"	4459T14	$13.91	4459T18	$25.68	4459T16	$21.40	4459T24	$28.53	4459T26	$53.50	4459T28	$89.17
0.032"	±0.005"	4459T141	14.83	4459T181	27.38	4459T161	22.82	4459T241	30.42	4459T261	57.04	4459T281	95.07
0.040"	±0.009"	4459T142	15.48	4459T182	28.58	4459T162	23.82	4459T242	31.76	4459T262	59.54	4459T282	99.24
0.050"	±0.009"	4459T143	16.62	4459T183	30.69	4459T163	25.57	4459T243	34.10	4459T263	63.94	4459T283	106.56
0.063"	±0.009"	4459T144	18.39	4459T184	33.95	4459T164	28.29	4459T244	37.72	4459T264	70.73	4459T284	117.89
0.071"	±0.009"	4459T145	19.13	4459T185	35.32	4459T165	29.43	4459T245	39.24	4459T265	73.58	4459T285	122.64
0.080"	±0.009"	4459T146	20.05	4459T186	37.02	4459T166	30.85	4459T246	41.14	4459T266	77.13	4459T286	128.55
0.100"	±0.009"	4459T147	22.58	4459T187	41.69	4459T167	34.74	4459T247	46.32	4459T267	86.84	4459T287	144.74
0.125"	±0.009"	4459T148	25.65	4459T188	47.35	4459T168	39.46	4459T248	52.61	4459T268	98.64	4459T288	164.40
0.160"	±0.009"	4459T149	28.23	4459T189	52.11	4459T169	43.42	4459T249	57.90	4459T269	108.56	4459T289	180.93
0.190"	±0.009"	**4459T15**	30.93	4459T19	57.10	4459T17	47.58	4459T25	63.44	4459T27	118.95	4459T29	198.25

Product Detail

Easy-to-Weld 4130 Alloy Steel Sheet with Material Certification, .190" Thick, 6" x 12"

Each

ADD TO ORDER

In stock

You ordered 1 each on 11/07/14.

0.250"	±0.009"	4459T151	36.04	4459T191	66.54	4459T171	55.45	4459T251	73.93	4459T271	138.62	4459T291	231.03

Raw stock for clutch/gear adapter plate, .125” thick, (McMaster-Carr)

Easy-to-Weld 4130 Alloy Steel

Its carbon content is low enough for good weldability but high enough to give this steel abrasion and impact resistance. 4130 is often used for gears, fasteners, and structural applications.

Warning: Physical, mechanical, and chemical properties are not guaranteed and are intended only as a basis for comparison.

Sheets with Material Certification—Unpolished

- Yield Strength: 50,000 psi
- Hardness: Medium (up to Rockwell B85)
- Can be hardened up to Rockwell C60

Meet AMS 6350-6351. Each sheet comes with a traceable lot number and material test report. All sheets are annealed. 0.025" to 0.125" thick sheets are cold rolled. 0.160" and thicker are hot rolled. Width and length tolerances are ±1/16".

		6" × 12"		6" × 36"		12" × 12"		12" × 24"		24" × 24"		36" × 36"	
Thick.	Thick. Tolerance		Each		Each		Each		Each		Each		Each
0.025"	±0.005"	4459T14	$13.91	4459T18	$25.68	4459T16	$21.40	4459T24	$28.53	4459T26	$53.50	4459T28	$89.17
0.032"	±0.005"	4459T141	14.83	4459T181	27.38	4459T161	22.82	4459T241	30.42	4459T261	57.04	4459T281	95.07
0.040"	±0.009"	4459T142	15.48	4459T182	28.58	4459T162	23.82	4459T242	31.76	4459T262	59.54	4459T282	99.24
0.050"	±0.009"	4459T143	16.62	4459T183	30.69	4459T163	25.57	4459T243	34.10	4459T263	63.94	4459T283	106.56
0.063"	±0.009"	4459T144	18.39	4459T184	33.95	4459T164	28.29	4459T244	37.72	4459T264	70.73	4459T284	117.89
0.071"	±0.009"	4459T145	19.13	4459T185	35.32	4459T165	29.43	4459T245	39.24	4459T265	73.58	4459T285	122.64
0.080"	±0.009"	4459T146	20.05	4459T186	37.02	4459T166	30.85	4459T246	41.14	4459T266	77.13	4459T286	128.55
0.100"	±0.009"	4459T147	22.58	4459T187	41.69	4459T167	34.74	4459T247	46.32	4459T267	86.84	4459T287	144.74
0.125"	±0.009"	**4459T148**	25.65	4459T188	47.35	4459T168	39.46	4459T248	52.61	4459T268	98.64	4459T288	164.40

Product Detail

Easy-to-Weld 4130 Alloy Steel Sheet with Material Certification, .125" Thick, 6" x 12"

Each

ADD TO ORDER

In stock

0.160"	±0.009"	4459T149	28.23	4459T189	52.11	4459T169	43.42	4459T249	57.90	4459T269	108.56	4459T289	180.93
0.190"	±0.009"	4459T15	30.93	4459T19	57.10	4459T17	47.58	4459T25	63.44	4459T27	118.95	4459T29	198.25
0.250"	±0.009"	4459T151	36.04	4459T191	66.54	4459T171	55.45	4459T251	73.93	4459T271	138.62	4459T291	231.03

Steel drive shaft with keyway (McMaster-Carr)

Fully Keyed Drive Shafts

For a secure hold in higher-torque applications, an ANSI keyway runs the entire length of the shaft (keys not included; see our selection of key stock). Straightness tolerance is 0.012" per foot for inch sizes and 0.305 mm per 300 mm for metric sizes. Ends are beveled.

Steel Shafts—Made of 1045 steel, these shafts are stronger than stainless steel and aluminum shafts but are less corrosion resistant. Rockwell hardness is B95.

Type 304 Stainless Steel Shafts—Have good corrosion resistance but are not as strong as steel shafts. Rockwell hardness is B85.

Type 316 Stainless Steel Shafts—Have excellent corrosion resistance but are not as strong as steel shafts. Rockwell hardness is B85.

Aluminum Shafts—Made of Alloy 2024-T4 aluminum, these shafts are lightweight and have more corrosion resistance than steel shafts. Hardness is Brinell 120.

CAD For technical drawings and 3-D models, click on a part number.

Steel Shafts—Inch Sizes

Shaft Dia.	Dia. Tolerance
5/15"- 2 1/2"	-0.0025" to -0.001"

Lg.		Each
5/16" (Keyway: 3/32" Wd. × 3/64" Dp.)		
3"	1497K1	$7.36
6"	1497K2	11.52
9"	1497K3	15.67
12"	1497K111	16.33
18"	1497K951	18.79
24"	1497K11	24.50
36"	1497K12	33.00
48"	1497K13	38.59
60"	1497K14	46.56
72"	1497K15	51.03
3/8" (Keyway: 3/32" Wd. × 3/64" Dp.)		
3"	1497K4	6.25
6"	1497K5	9.82
9"	1497K6	13.37
12"	1497K121	15.14
18"	1497K952	20.45
24"	1497K21	27.33
36"	1497K22	34.52
48"	1497K23	44.35

Lg.		Each
3/4" (Keyway: 3/16" Wd. × 3/32" Dp.)		
3"	1497K114	$8.76
6"	1497K115	14.24
9"	1497K116	19.70
12"	1497K161	21.20
18"	**1497K956**	26.77
24"	1497K61	33.23
36"	1497K62	42.88
48"	1497K63	54.55
60"	1497K64	64.26

Product Detail CAD

Fully Keyed 1045 Steel Drive Shaft, 3/4" OD, 3/16" Keyway Width, 18" Length

Each

ADD TO ORDER

In stock

You ordered 1 each on 12/04/14.

Lg.		Each
1" (Keyway: 1/4" Wd. × 1/8" Dp.)		
3"	1497K143	$10.18
6"	1497K144	17.12
9"	1497K145	24.09
12"	1497K281	27.60
18"	1497K961	35.48
24"	1497K101	41.82
36"	1497K102	58.73
48"	1497K103	75.70
60"	1497K104	87.31
72"	1497K105	99.74
1 1/8" (Keyway: 1/4" Wd. × 1/8" Dp.)		
3"	1497K146	10.76
6"	1497K147	18.33
9"	1497K148	25.76
12"	1497K291	32.73
18"	1497K962	40.30
24"	1497K201	48.43
36"	1497K202	57.69
48"	1497K203	80.00

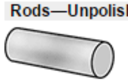
Raw stock for coupler housing (McMaster-Carr)

Hard High-Strength 7075 Aluminum

Originally developed for aircraft frames, uses for Alloy 7075 now include keys, gears, and other high-stress parts. It is nonmagnetic and heat treatable. Temperature range is -320° to 210° F.

Yield strength is approximate and may vary based on size and shape.

Rods—Unpolished



- Yield Strength: 62,000 psi
- Hardness: Medium (150 Brinell)
- Temper: Heat Treated (T6 or T651)
- Meet ASTM B211

These rods also meet AMS QQ-A-225/9. Length tolerance is ±1/8".

Dia.	Dia. Tolerance		1/2 ft.	1 ft.	2 ft.	3 ft.	6 ft.
5 1/2"	±0.016"	**90465K75**	$183.96	$321.93	$567.21	$843.15	$1,533.00

Product Detail

Hard High-Strength 7075 Aluminum, Rod, 5-1/2" Diameter

Each

ADD TO ORDER

Length, ft.

1/2

1

2

3

6

Short Rods—

Rear reduction gear (Motion Industries)

Martin Sprocket & Gear C1290

Product Categories Mechanical Power Transmission > Gears & Gear Racks > Spur & Change Gears > External Tooth Spur Gears > C1290

C1290, External Tooth Spur Gear, Pitch: 12", Pressure Angle: 14.5°, Cast Iron, 90 Teeth, Pitch Diameter: 7-1/2"

Mfr Part #: C1290
Mi Item #: 00372803
Mfr Description: C1290 14 1/2 SPUR GEAR

View Larger

How can we make our Product Content better?

Price $117.55 (EACH) Available to Order

Quantity QTY

Add to Cart

ADDITIONAL INFORMATION

Country of Origin: UNITED STATES
Country of Origin is subject to change.

Scheduled Item

Product Description | Substitute Items

General Information

Pitch:	12"
Pressure Angle:	14.5 °
Face Width:	3/4"
Number of Teeth:	90
Pitch Diameter:	7-1/2"
Bore Type:	Rough Stock Bore
Bore Diameter:	3/4 to 1-11/16"
Material:	Cast Iron
Hub Type:	Hub with Set Screw
Outside Diameter:	7.666"
Hub Diameter:	2-3/4"
Gear Construction:	Spoked
Overall Width:	1-1/2"
Hub Projection:	3/4"
Standards Met:	ANSI Standard B15.1

Hydraulic Oil

Mobil

Mobil EAL 224H

Hydraulic Fluid

Product Description

Mobil EAL 224H is a premium performance environmentally aware hydraulic fluid designed to provide outstanding performance in hydraulic and circulation systems operating at moderate conditions. It provides excellent anti-wear and film strength characteristics necessary for hydraulic systems operating under high load and high pressures. Its 12-stage rating in the FZG Gear Load test demonstrates a high level of protection against wear and scuffing and the suitability of this product to protect gears and bearings used in conjunction with hydraulic systems. Mobil EAL 224H provides excellent protection against corrosion and ensures very good multi-metal compatibility allowing its use in systems employing various metallurgy that may be used in pump and component designs. It also provides very good thin oil film protection against rusting. In addition to its exceptional performance capability, it satisfies the requirements for ready biodegradability and non-toxicity making it a desirable product where leakage or spillage of conventional oils could result in damage to the environment.

It is formulated from select, high-quality, high-VI vegetable oils and a specifically engineered additive system to meet or exceed the performance requirements of most hydraulic pump and system builders while satisfying the stringent criteria for biodegradability and toxicity.

Features and Benefits

Mobil EAL 224H provides excellent anti-wear, lubricity, and film strength performance in hydraulic and circulation systems operating under moderate operating conditions. The ready biodegradability and virtually non-toxic nature of this product makes it an excellent choice where leakage or spillage could enter environmentally sensitive areas. The inadvertent leakage of spillage of this product in environmentally sensitive areas could result in easier clean-up and lower remediation costs.

Features	Advantages and Potential Benefits
Ready Biodegradability and Non-Toxicity	Reduces potential for environmental damage Lowers potential remediation and clean-up costs caused by spills or leakage Becomes an integral part of plant environmental program
Outstanding Load-Carrying and Anti-Wear Properties	Protects system components against wear and scuffing Provides long equipment life
Exceptional Corrosion Protection	Reduces corrosion of internal system components
Excellent Multi-Metal Compatibility	Will not react with steel or copper alloys
Good Elastomer Compatibility	Works well with same elastomers used with conventional mineral based oils. No need for special seals or elastomers

Applications

- Hydraulic systems where spills or leakage could result in damage to the environment
- In systems where readily biodegradable and virtually non-toxic fluids may be required
- Gear systems requiring either an ISO VG 32 or 46 oil with mild extreme-pressure characteristics
- Systems containing servo-valves
- Hydraulic systems operating with oil temperatures in the range of 0 F to 180 F
- Marine and mobile equipment operating in environmentally sensitive areas
- Circulation systems operating under mild to moderate service conditions
- Industrial hydraulic systems where leaked or spilled fluids could get into plant effluent
- Air line oilers and some limited oil-mist generating systems
- Air-over-hydraulic fluid systems operating in environmentally sensitive areas

Specifications and Approvals

Mobil EAL 224H recommended for use in applications requiring:
Environmentally friendly characteristics
Anti-wear protection
Compatibility with system components

Typical Properties

Mobil EAL 224H	
Viscosity, ASTM D 445	
cSt @ 40º C	36.78
cSt @ 100º C	8.3
Viscosity Index, ASTM D 2270	212
Specific Gravity @ 15º C/15º C, ASTM D 1298	0.921
FZG Gear Test, DIN 51354, Fail Stage	12
4-Ball Wear, ASTM D 4172, 40 kg, 93º C, 600 rpm Scar Diameter., mm	0.35
Pour Point, ºC, ASTM D 97	-34
Flash Point, ºC, ASTM D 92	294
Vickers V-104C Pump Wear, ASTM D2882, mg	10
Biodegradability, CO2 Conversion, EPA 560/6-82-003, wt %	›70
Aquatic Toxicity, LC50, Trout, OECD 203, ppm	›5000

Health and Safety

Based on available information, this product is not expected to produce adverse effects on health when used for the intended application and the recommendations provided in the Material Safety Data Sheet (MSDS) are followed. MSDS's are available upon request through your sales contract office, or via the Internet. This product should not be used for purposes other than its intended use. If disposing of used product, take care to protect the environment.

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9-2014

Exxon Mobil Corporation
3225 Gallows Road
Fairfax, VA 22037

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Typical Properties are typical of those obtained with normal production tolerance and do not constitute a specification. Variations that do not affect product performance are to be expected during normal manufacture and at different blending locations. The information contained herein is subject to change without notice. All products may not be available locally. For more information, contact your local ExxonMobil contact or visit www.exxonmobil.com
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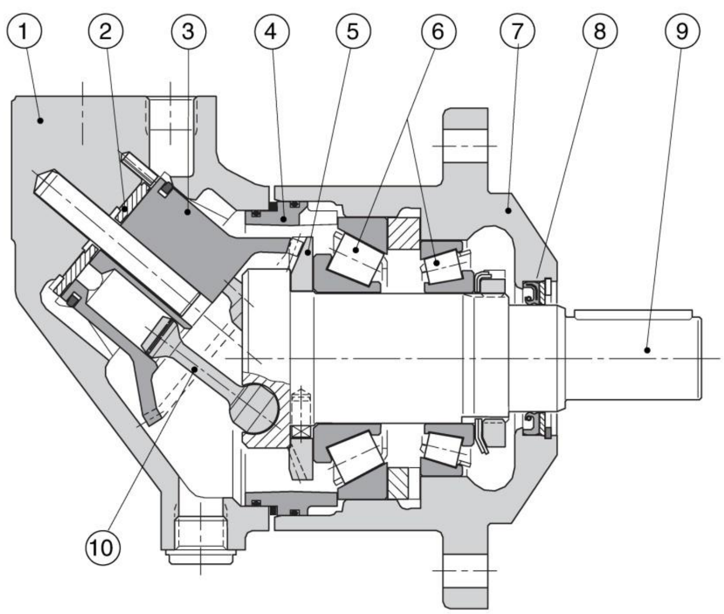
Parker F11-5 Hydraulic Pump

F11 and F12 are bent axis, fixed displacement heavy-duty motor/pump series. They can be used in numerous applications in both open and closed loop circuits.

- Series F11 is available in the following frame sizes and versions:
  - F11-5, -10, -14, 19 and -150 with CETOP mounting flange and shaft end
  - F11-14 with ISO flange and shaft
  - F11-14, -19, -150 and -250 with SAE flange and shaft
- Series F12 conforms to current ISO and SAE mounting flange and shaft end configurations. A very compact cartridge version is also available.
- Thanks to the unique spherical piston design, F11/F12 motors can be used at unusually high shaft speeds. Operating pressures to 480 bar provides for the high output power capability.
- The 40° angle between shaft and cylinder barrel allows for a very compact, lightweight motor/pump.
- The laminated piston ring offers important advantages such as low internal leakage and thermal shock resistance.
- The pump version has highly engineered valve plates for increased selfpriming speed and low noise, available with left and right hand rotation.
- The F11/F12 motors produce very high torque at start-up as well as at low speeds.
- Our unique timing gear design synchronizes shaft and cylinder barrel, making the F11/F12 very tolerant to high 'G' forces and torsional vibrations.
- Heavy duty roller bearings permit substantial external axial and radial shaft loads.
- The F11's and F12's have a simple and straight-forward design with very few moving parts, making them very reliable motors/pumps.
- The unique piston locking, timing gear and bearing set-up as well as the limited number of parts add up to a very robust design with long service life and, above all, proven reliability.

F11 cross section

1. Barrel housing
2. Valve plate
3. Cylinder barrel
4. Guide spacer with O-rings
5. Timing gear
6. Roller bearing
7. Bearing housing
8. Shaft seal
9. Output/input shaft
10. Piston with laminated piston ring



Frame size	F11-5	-10	-14	-19	F12-30	-40	-60	-80	-110	F11-150	-250
Displacement [cm³/rev]	4.9	9.8	14.3	19.0	30.0	40.0	59.8	80.4	110.1	150	242
[cu in/rev]	*0.30*	*0.60*	*0.87*	*1.16*	*1.83*	*2.44*	*3.65*	*4.91*	*6.72*	*9.15*	*14.8*
Operating pressure											
max intermittent[1)] [bar]	420	—	—	420	480	—	—	—	480	420	420
[psi]	*6 000*	—	—	*6 000*	*7 000*	—	—	—	*7 000*	*6 000*	*6 000*
max continuous [bar]	350	—	—	350	420	—	—	—	420	350	350
[psi]	*5 000*	—	—	*5 000*	*6000*	—	—	—	*6000*	*5 000*	*5 000*
Motor operating speed [rpm]											
max intermittent[1)]	12 000	11 000	10 000	9 000	7 100	6 400	5 600	5 200	4 700	3 000	2 700
max continuous	10 800	9 900	9 000	8 100	5 600	5 000	4 300	4 000	3 600	2 600	2 400
min continuous	50	—	—	—	—	—	—	—	—	—	50
Max pump selfpriming speed[2)]											
L or R function; max [rpm]	4 600	4 200	3 900	3 500	3150	2870	2500	2300	2290	1 700	1 500
Motor input flow											
max intermittent[1)] [l/min]	58	108	143	171	213	256	335	418	517	450	650
[gpm]	*15.3*	*28.5*	*37.8*	*45.2*	*56.3*	*67.6*	*88.5*	*110.4*	*136.6*	*119.0*	*172.8*
max continuous [l/min]	52	97	128	153	168	200	257	322	396	390	580
[gpm]	*13.7*	*25.6*	*33.8*	*40.4*	*44.4*	*52.8*	*67.9*	*85.1*	*104.6*	*103.0*	*153.6*
Main circuit temp.[3)], max [°C]	80	—	—	—	—	—	—	—	—	—	80
[°F]	*175*	—	—	—	—	—	—	—	—	—	*175*
min [°C]	-35	-35	-30	-35	-35	—	—	—	-35	-35	-35
[°F]	*-31*	*-31*	*-22*	*-31*	*-31*	—	—	—	*-31*	*-31*	*-31*
Mass moment of inertia											
(x10⁻³) [kg m²]	0.16	0.39	0.42	1.1	1.7	2.9	5	8.4	11.2	40	46
(x10⁻²) *[lbft²]*	*0.38*	*0.92*	*1.00*	*2.61*	*4.03*	*6.88*	*11.86*	*19.93*	*26.58*	*94.92*	*109.16*
Weight [kg]	5	7.5	8.3	11	12	16.5	21	26	36	70	77
[lb]	*11*	*16.5*	*18.8*	*24*	*26*	*36*	*46*	*57*	*79*	*154*	*170*

1) Intermittent: max 6 seconds in any one minute.
2) Selfpriming speed valid at sea level.
3) See also installation information, operating temperature.

Basic formulas for hydraulic motors

Flow (q)

$$q = \frac{D \times n}{1000 \times \eta_v} \text{ [l/min]}$$

Torque (M)

$$M = \frac{D \times \Delta p \times \eta_{hm}}{63} \text{ [Nm]}$$

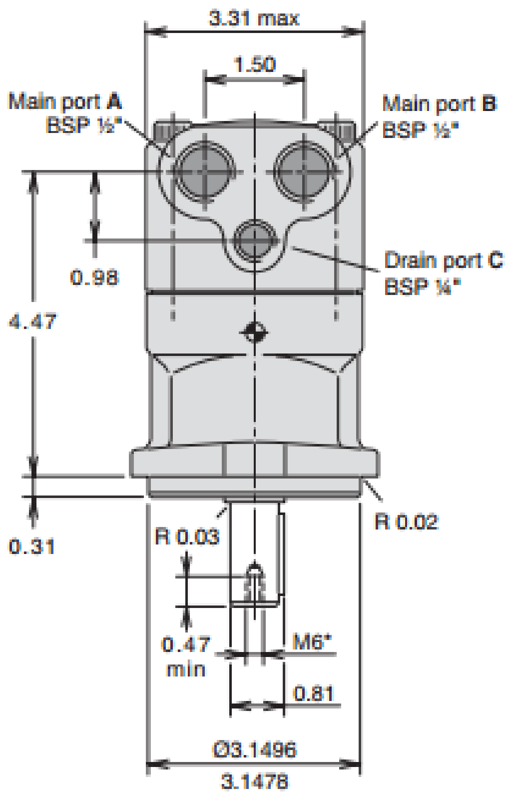
Power (P)

$$P = \frac{q \times \Delta p \times \eta_t}{600} \text{ [kW]}$$

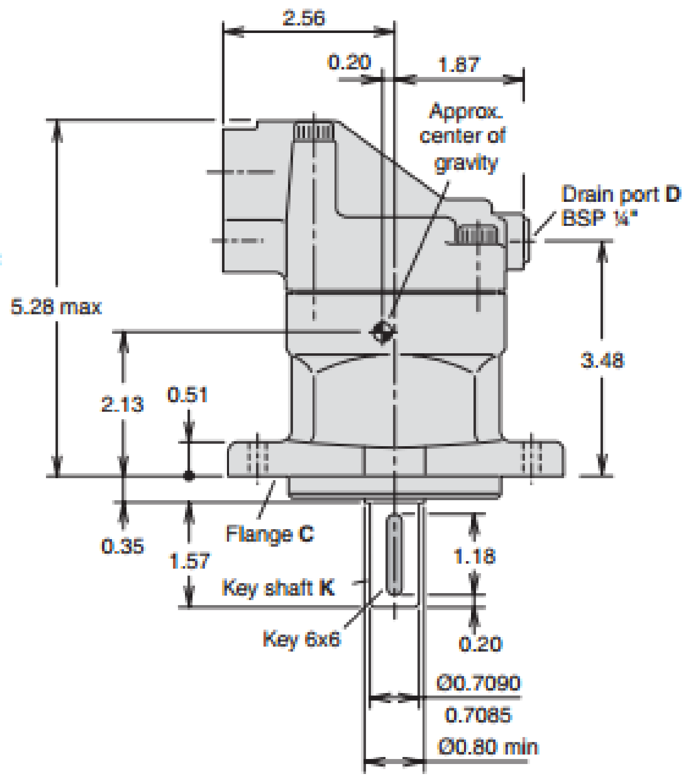
D - displacement [cm³/rev]
n - shaft speed [rpm]
η_v - volumetric efficiency
Δp - differential pressure [bar] (between inlet and outlet)
η_{hm} - mechanical efficiency
η_t - overall efficiency ($\eta_t = \eta_v \times \eta_{hm}$)

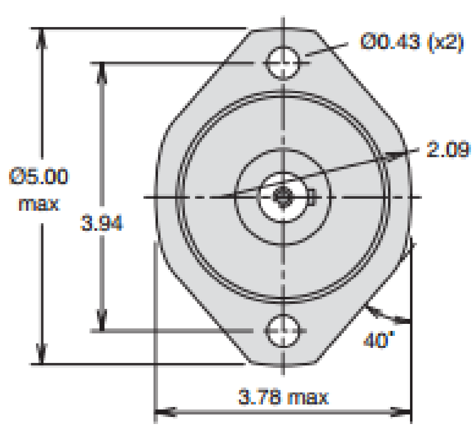
F11-5

(CETOP versions)

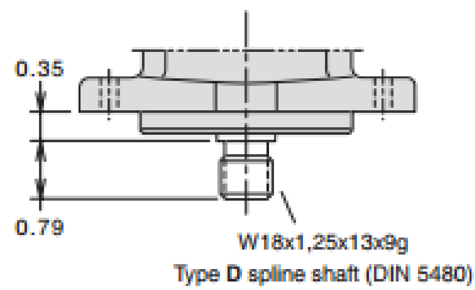


* Metric thread





Mounting flange C



MSR Dromedary Bag

DROMEDARY® BAGS

OUR BURLIEST MEDIUM-TO-LARGE CAPACITY WATER STORAGE AND DELIVERY SYSTEMS.

Completely collapsible for optimal packing efficiency, our burliest medium-to-large capacity water storage and delivery systems are perfect for everything from alpine- to road warrior-style expeditions. Armed with incredibly tough 500 and 1000-denier Cordura® exteriors and laminated with a BPA-Free and food-grade polyurethane lining, our Dromedary Bags can handle freezing and just about any kind of abuse you can imagine. Accessories like the Shower or Hydration Kits add do-all versatility.

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Choose style/color ▼

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DETAILS SPECS REVIEWS AWARDS

View specs in printer-friendly format

	2 LTR	4 LTR	6 LTR	10 LTR
Color	Black	Black	Black	Black
Capacity	70 fl. oz / 2 liters	140 fl. oz / 4 liters	210 fl. oz / 6 liters	350 fl. oz / 10 liters
Weight	6.3 oz / 179 g	6.9 oz / 196 g	8.7 oz / 247 g	10 oz / 284 g
Width	8 in / 20 cm	10 in / 25 cm	11 in / 28 cm	13.5 in / 34 cm
Length	16.5 in / 42 cm	19 in / 48 cm	22 in / 56 cm	23.5 in / 60 cm
Film	Cordura-500 (Nylon) / TPU	Cordura-500 (Nylon) / TPU	Cordura-1000 (Nylon) / TPU	Cordura-1000 (Nylon) / TPU
Spout material	TPU	TPU	TPU	TPU
Cap material	HDPE	HDPE	HDPE	HDPE
Country of Origin	Made in Seattle, USA	Made in Seattle, USA	Made in Seattle, USA	Made in Seattle, USA

Parker Piston Accumulator

ACP Series Piston Accumulators

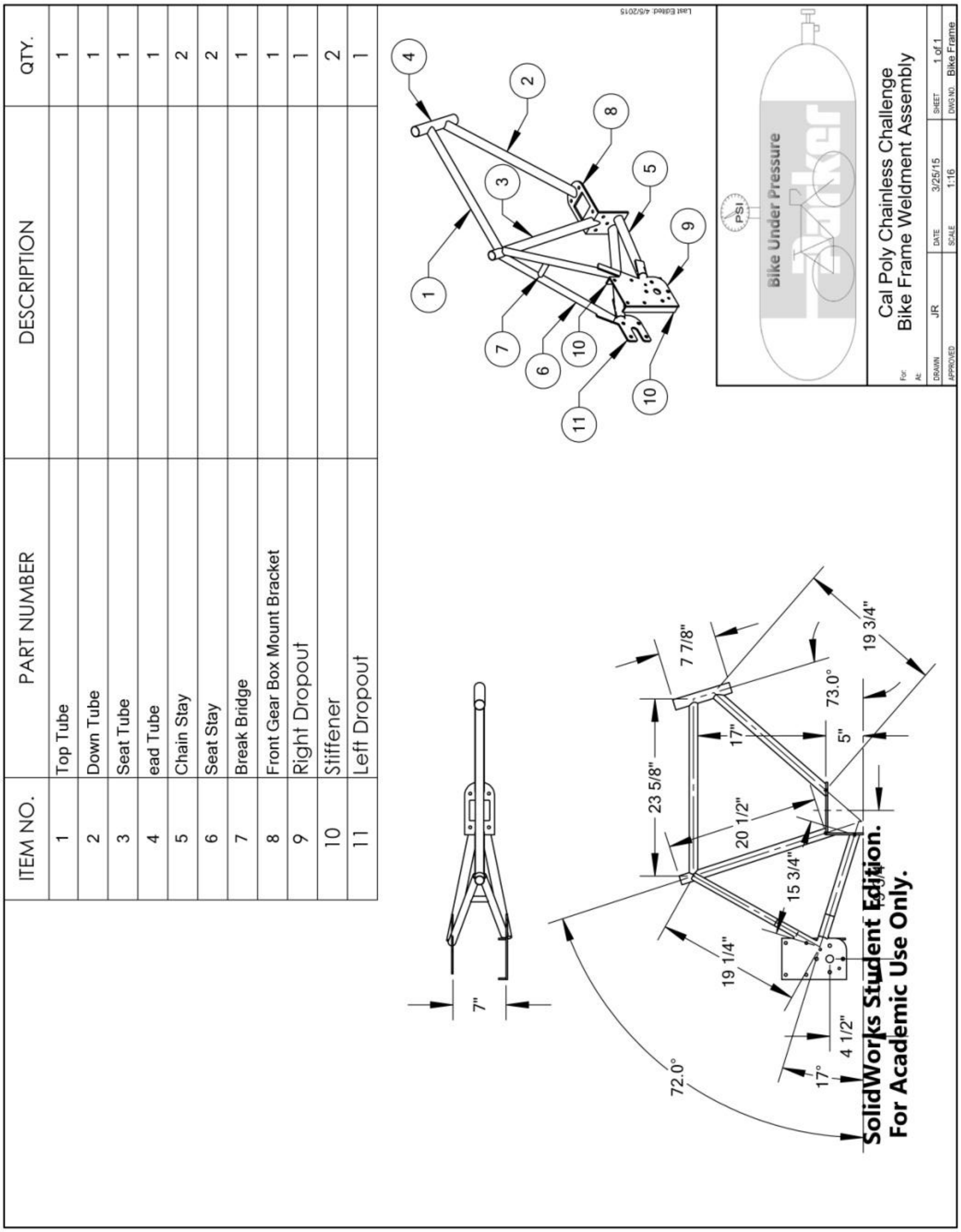
With Working Pressures of 3,770, 4000 and 5000 PSI

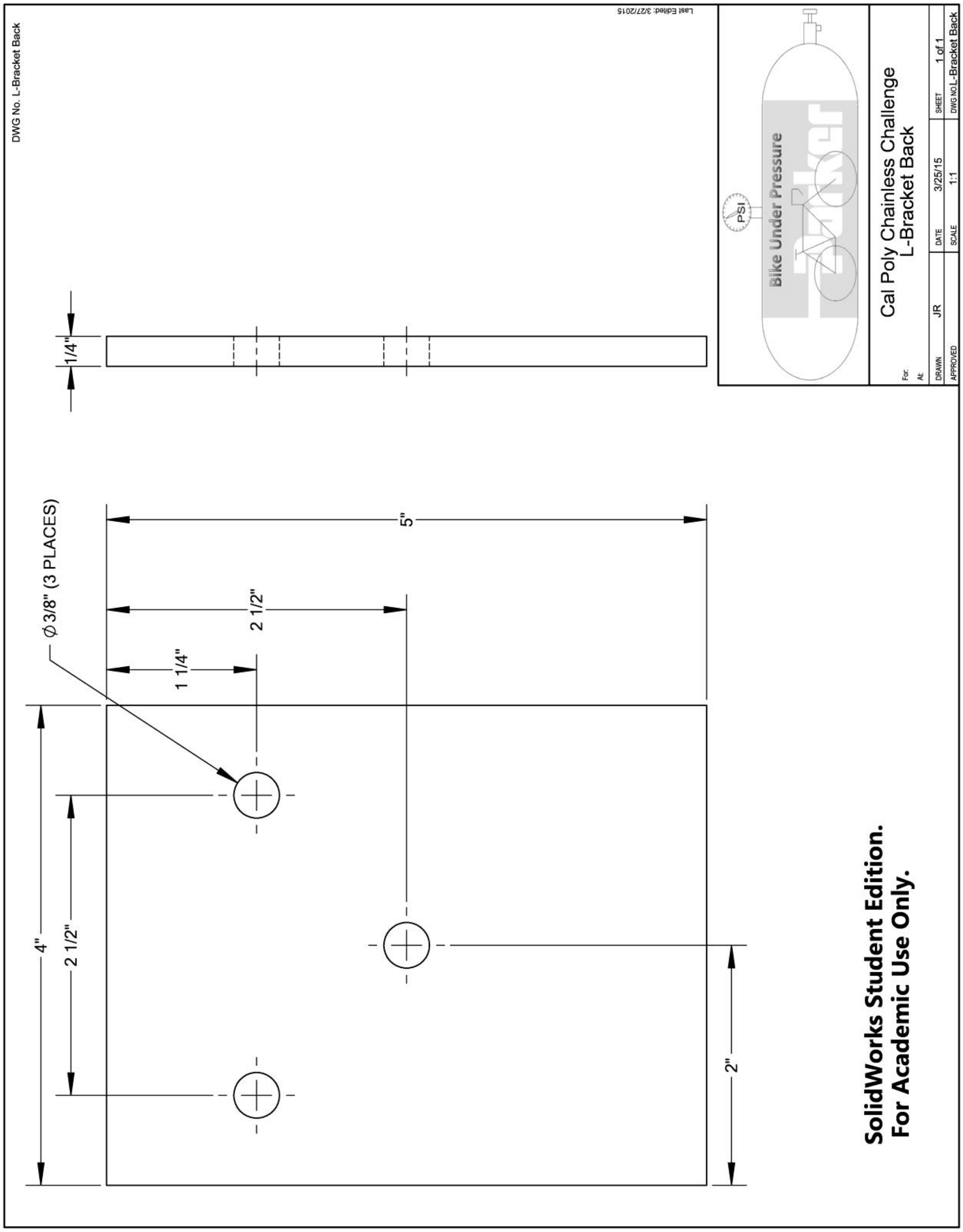
IN THIS SECTION

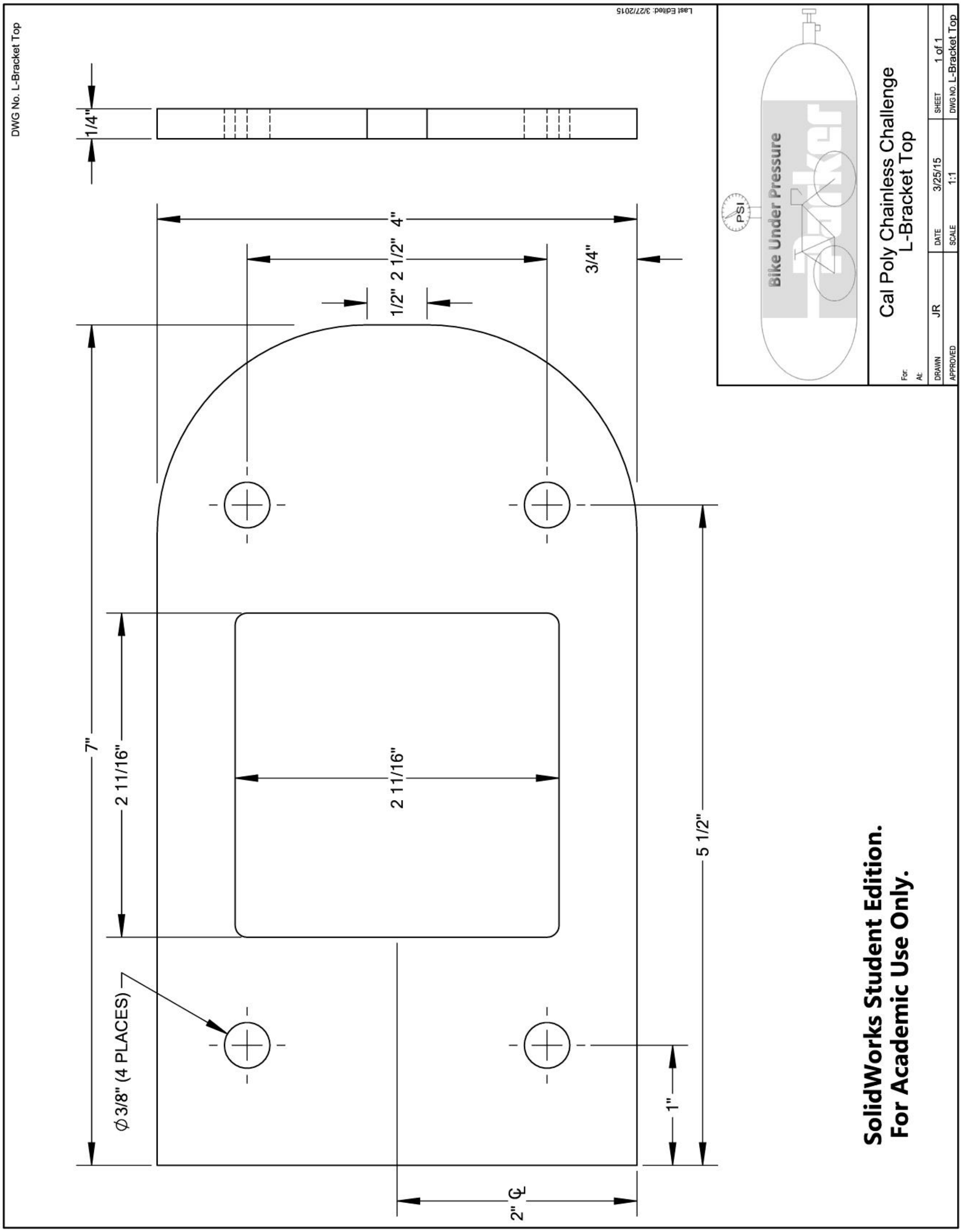
ACP Series Crimped Piston Accumulators are ideal for mobile and construction equipment applications in climates as rugged as those of northern Canada and Siberia. Rated at 3770, 4000 and 5000 PSI with a design factor of four to one or greater, these accumulators have been tested and proven at -40ºF/C or below, and meet or exceed CSA/CE/ASME standards.

A3E0058L1K model used by Bike Under Pressure with 63 cubic inches rated up to 4000 psi.

15. Appendix F: Drawings of Custom Parts




DWG No. L-Bracket Back
Last Edited: 3/27/2015
1/4"
⌀3/8" (3 PLACES)
5"
2 1/2"
1 1/4"
4"
2 1/2"
2"
SolidWorks Student Edition.
For Academic Use Only.
Bike Under Pressure
PSI
For:
At:
Cal Poly Chainless Challenge
L-Bracket Back
DRAWN JR
APPROVED
DATE 3/25/15
SCALE 1:1
SHEET 1 of 1
DWG NO. L-Bracket Back


DWG No. L-Bracket Top
Last Edited: 3/27/2015
1/4"
4"
2 1/2"
1/2"
3/4"
7"
2 11/16"
2 11/16"
5 1/2"
⌀3/8" (4 PLACES)
1"
2" ℄
Bike Under Pressure
PSI
Cal Poly Chainless Challenge
L-Bracket Top
For:
At:
DRAWN JR
DATE 3/25/15
SHEET 1 of 1
APPROVED
SCALE 1:1
DWG NO. L-Bracket Top
SolidWorks Student Edition.
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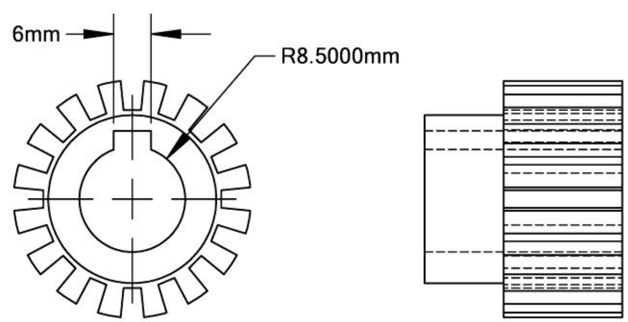
DWG No. 16T Pinion

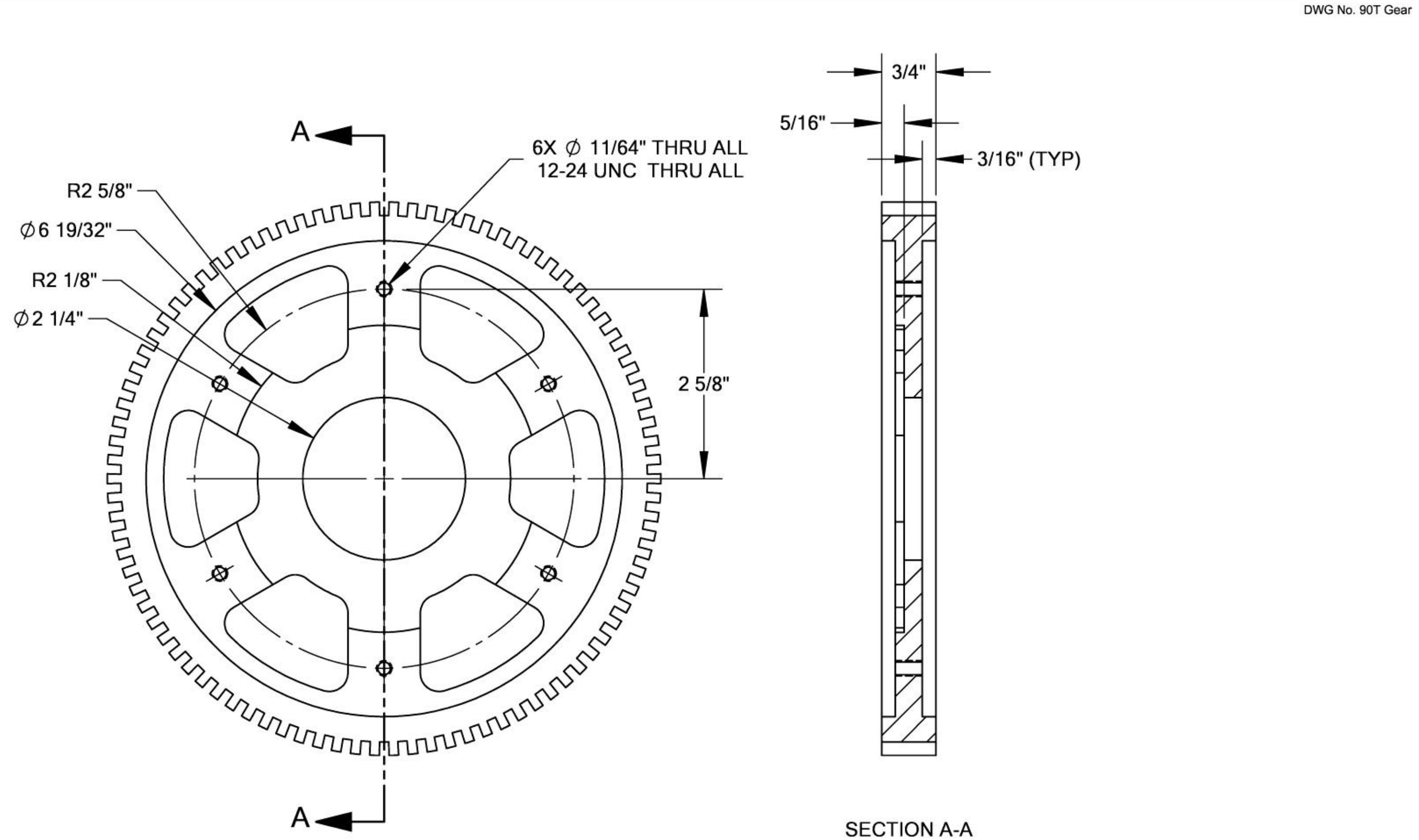


Last Edited: 3/27/2015

Bike Under Pressure

Cal Poly Chainless Challenge
16 Tooth Pinion

For:					
At:					
DRAWN	JR	DATE	3/25/15	SHEET	1 of 1
APPROVED		SCALE	1:1	DWG NO.	16T Pinion



SolidWorks Student Edition.
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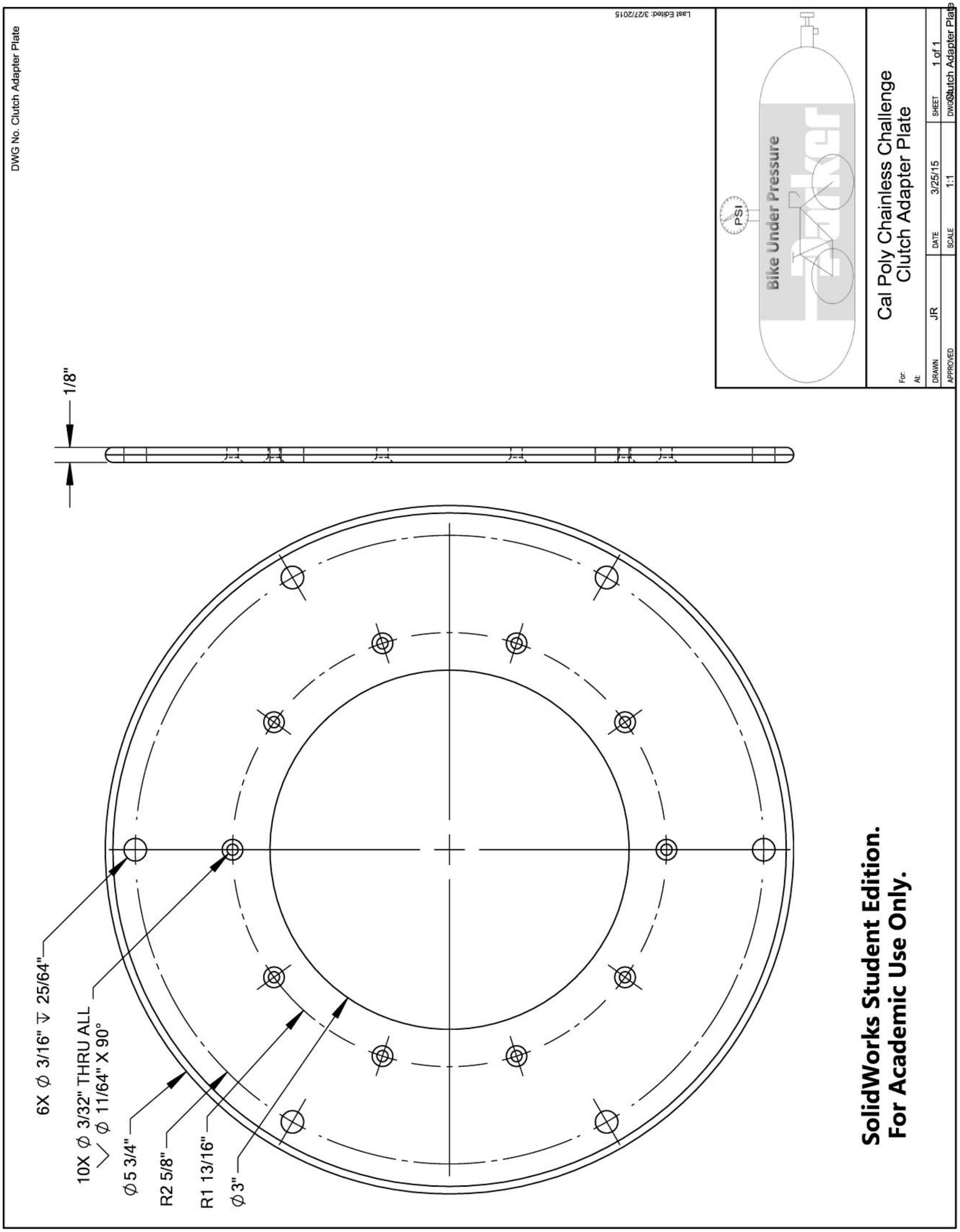
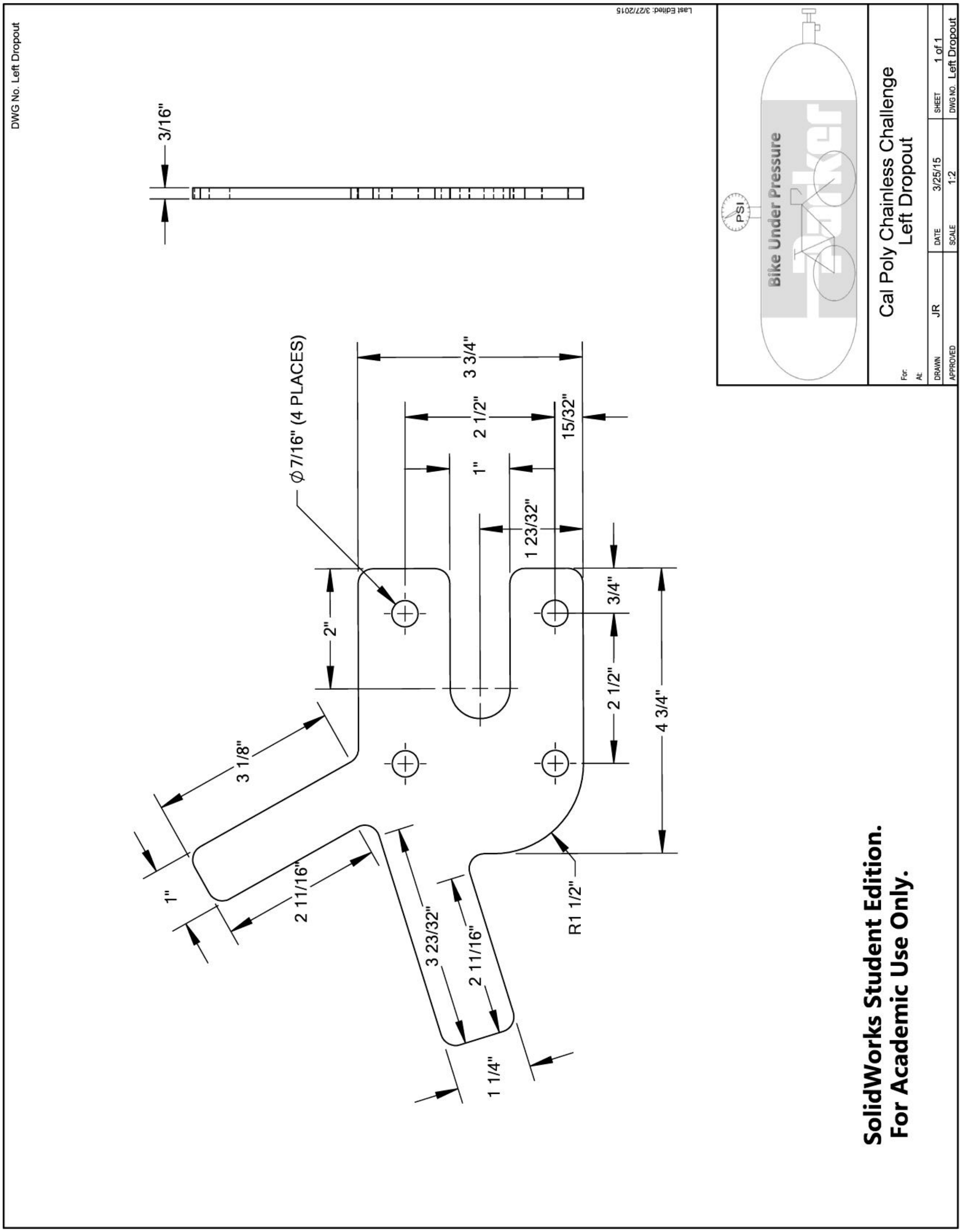
Bike Under Pressure

Cal Poly Chainless Challenge
90 Tooth Gear

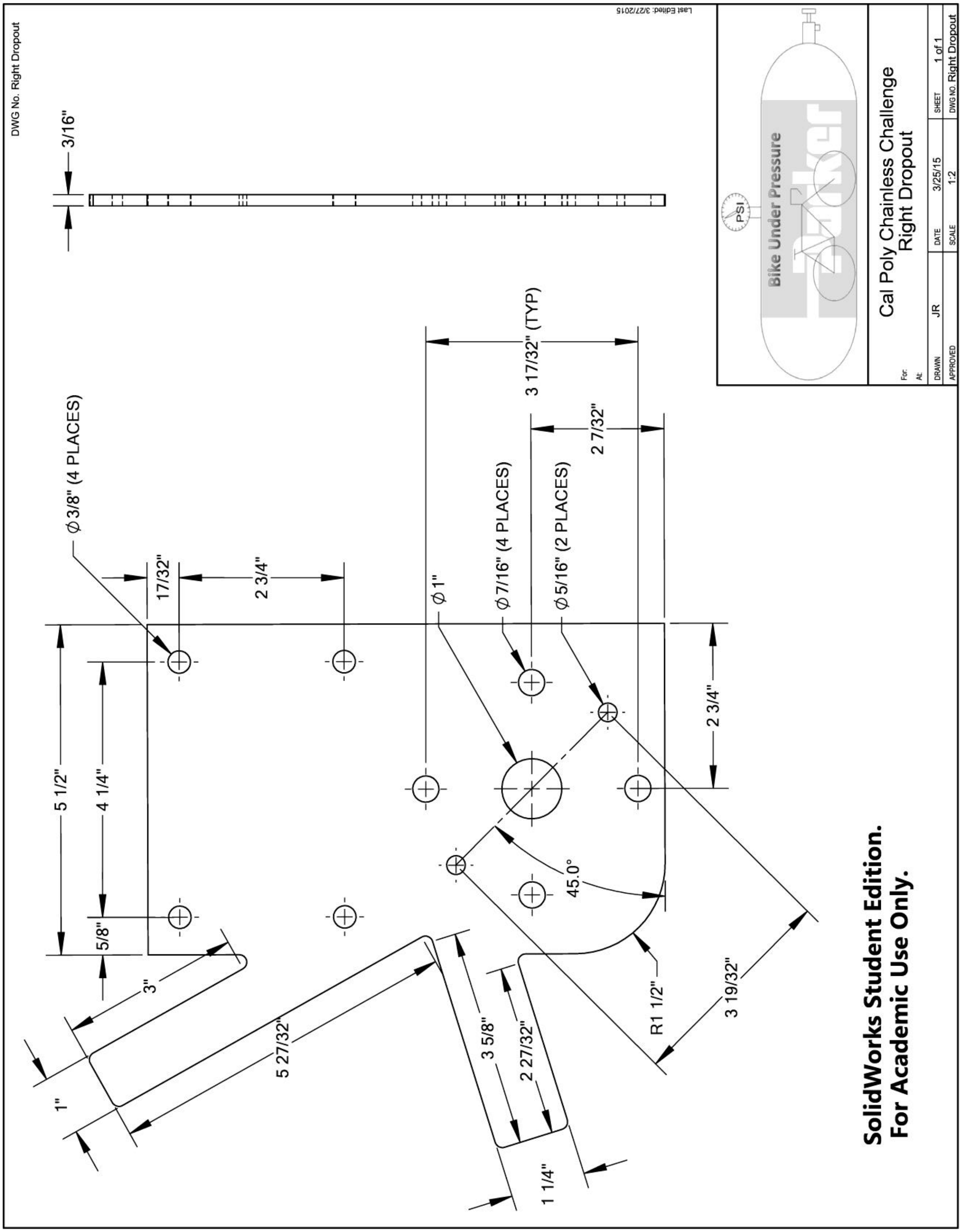
For:
At:

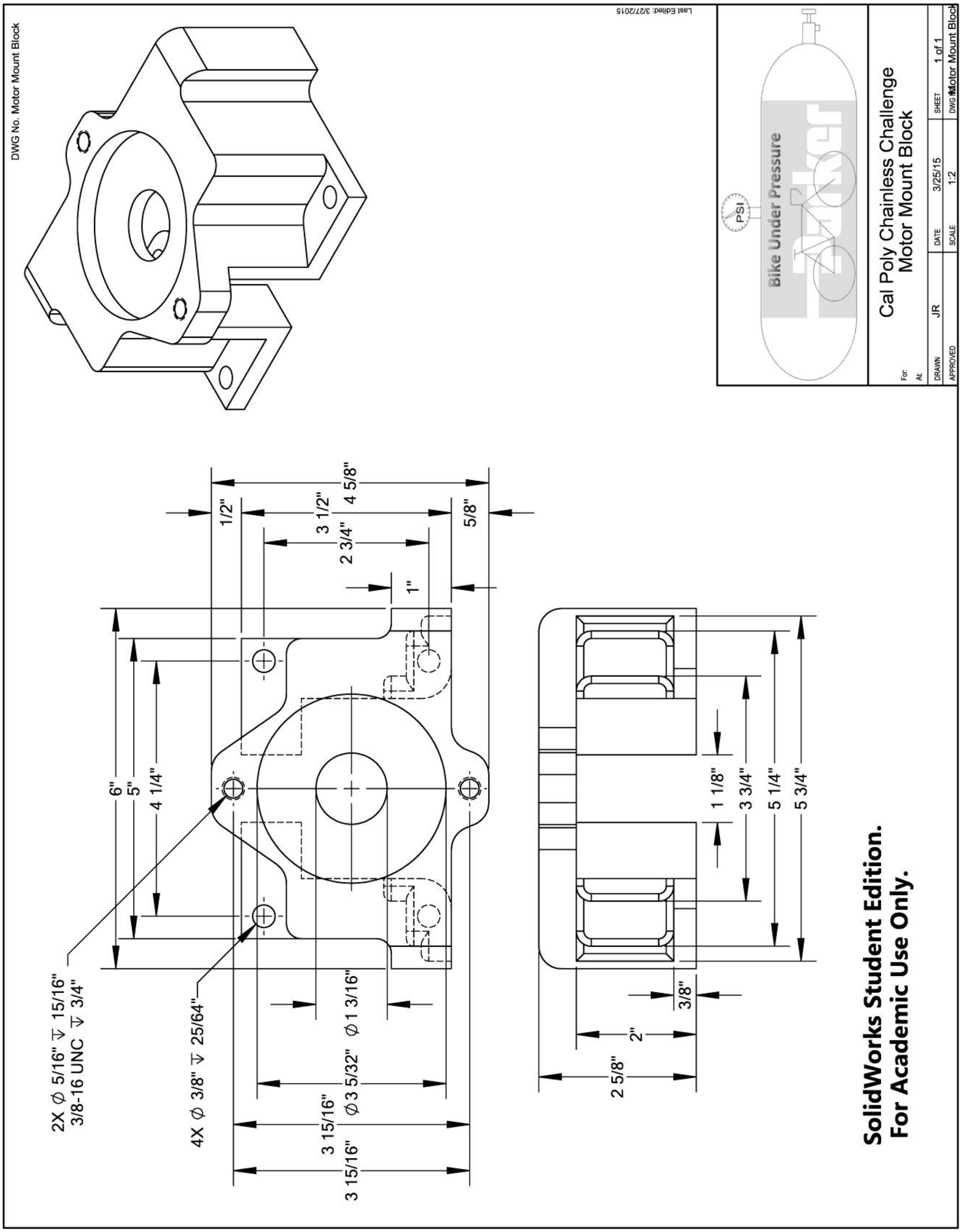
DRAWN	JR	DATE	3/25/15	SHEET	1 of 1
APPROVED		SCALE	1:2	DWG NO.	90T Gear


DWG No. Clutch Adapter Plate
Last Edited: 3/27/2015
1/8"
6X ⌀ 3/16" ↧ 25/64"
10X ⌀ 3/32" THRU ALL
⌵ ⌀ 11/64" X 90°
⌀ 5 3/4"
R2 5/8"
R1 13/16"
⌀ 3"
SolidWorks Student Edition.
For Academic Use Only.
PSI
Bike Under Pressure
Cal Poly Chainless Challenge
Clutch Adapter Plate
For:
At:
DRAWN JR
DATE 3/25/15
SHEET 1 of 1
APPROVED
SCALE 1:1
DWG No. Clutch Adapter Plate

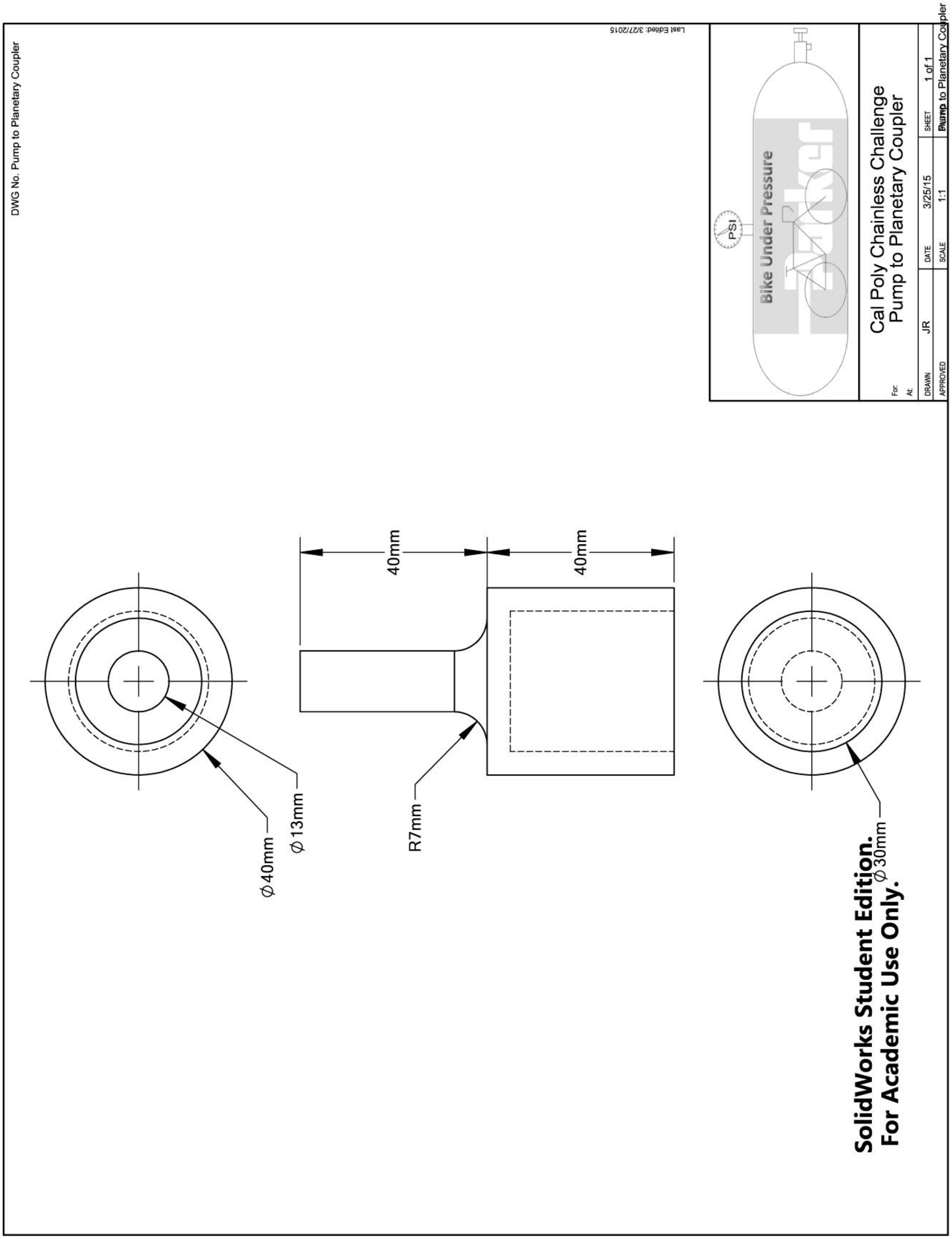


DWG No. Left Dropout
Last Edited: 3/27/2015
3/16"
Ø7/16" (4 PLACES)
3 3/4"
2 1/2"
15/32"
1"
1 23/32"
3/4"
2"
2 1/2"
4 3/4"
3 1/8"
1"
2 11/16"
3 23/32"
2 11/16"
R1 1/2"
1 1/4"
PSI
Bike Under Pressure
Cal Poly Chainless Challenge
Left Dropout
For:
At:
DRAWN JR
APPROVED
DATE 3/25/15
SCALE 1:2
SHEET 1 of 1
DWG NO. Left Dropout
SolidWorks Student Edition.
For Academic Use Only.

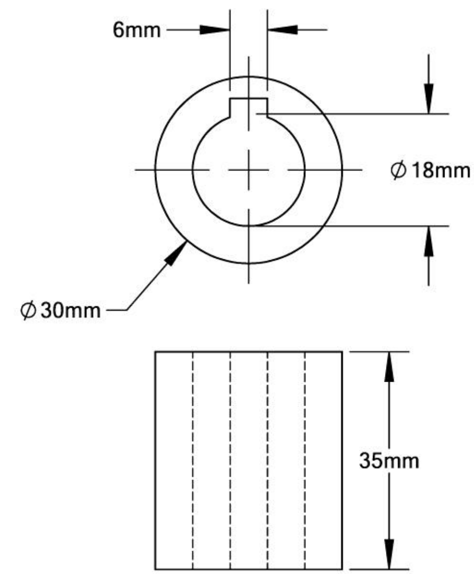

DWG No. Right Dropout
Last Edited: 3/27/2015
3/16"
Ø3/8" (4 PLACES)
17/32"
2 3/4"
Ø1"
Ø7/16" (4 PLACES)
Ø5/16" (2 PLACES)
3 17/32" (TYP)
2 7/32"
5 1/2"
4 1/4"
5/8"
2 3/4"
45.0°
R1 1/2"
3 19/32"
3"
5 27/32"
1"
3 5/8"
2 27/32"
1 1/4"
Bike Under Pressure
PSI
Cal Poly Chainless Challenge
Right Dropout
For:
At:
DRAWN
JR
DATE
3/25/15
SHEET
1 of 1
APPROVED
SCALE
1:2
DWG NO. Right Dropout
SolidWorks Student Edition.
For Academic Use Only.



DWG No. Motor Mount Block
Last Edited: 3/27/2015
Bike Under Pressure
PSI
Cal Poly Chainless Challenge
Motor Mount Block
For:
At:
DRAWN JR
DATE 3/25/15
SHEET 1 of 1
APPROVED
SCALE 1:2
DWG # Motor Mount Block
4 5/8"
1/2"
3 1/2"
5/8"
2 3/4"
1"
6"
5"
4 1/4"
1 1/8"
3 3/4"
5 1/4"
5 3/4"
2X Ø 5/16" ↧ 15/16"
3/8-16 UNC ↧ 3/4"
4X Ø 3/8" ↧ 25/64"
Ø1 3/16"
Ø3 5/32"
3 15/16"
3 15/16"
3/8"
2"
2 5/8"
SolidWorks Student Edition.
For Academic Use Only.


DWG No. Pump to Planetary Coupler
Last Edited: 3/27/2015
40mm
40mm
⌀40mm
⌀13mm
R7mm
⌀30mm
SolidWorks Student Edition.
For Academic Use Only.
PSI
Bike Under Pressure
Cal Poly Chainless Challenge
Pump to Planetary Coupler
For:
At:
DRAWN JR
DATE 3/25/15
SHEET 1 of 1
APPROVED
SCALE 1:1
Pump to Planetary Coupler

DWG No. Pump to Planetary Coupler Insert



Last Edited: 3/27/2015

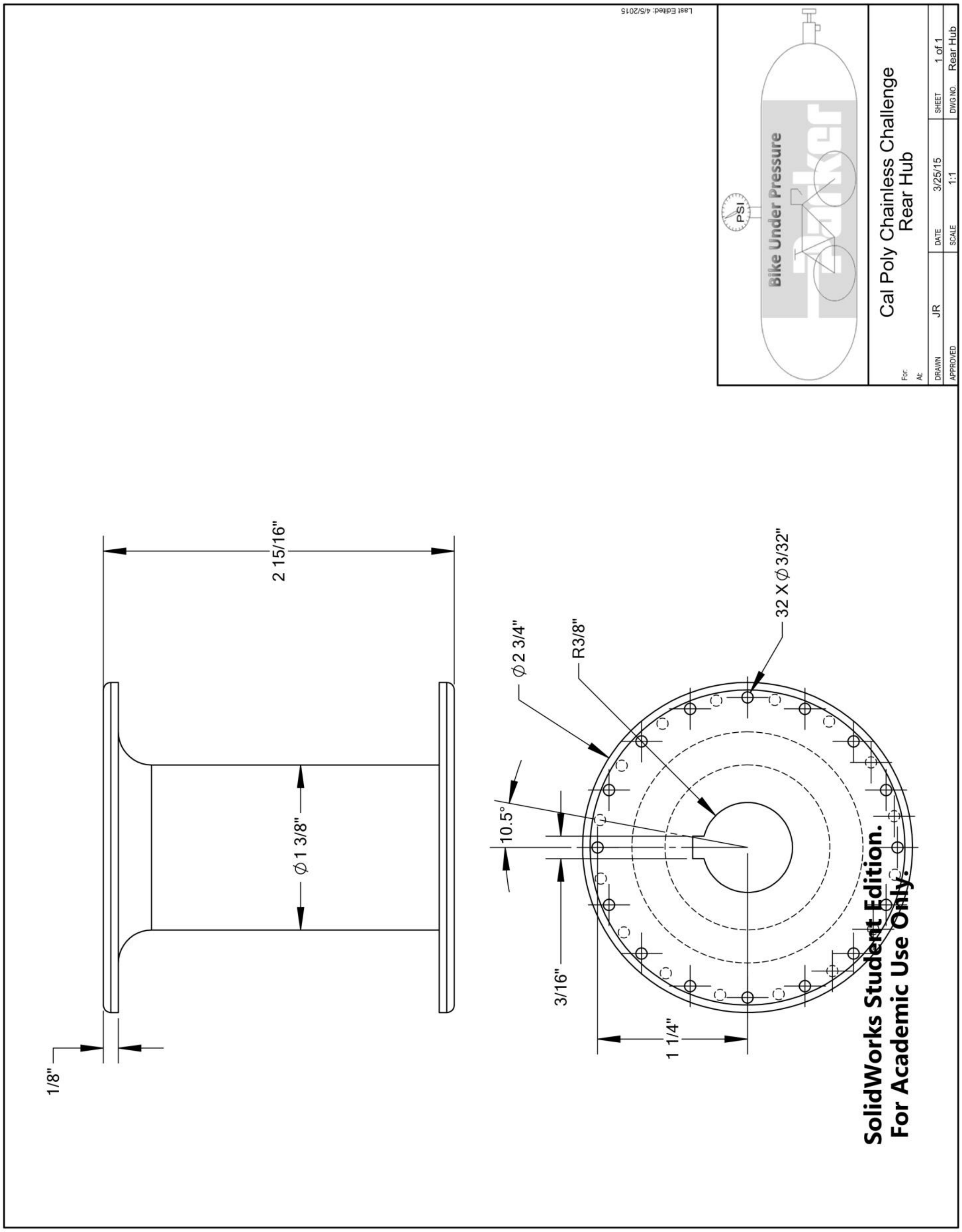
Bike Under Pressure

Cal Poly Chainless Challenge
Pump to Planetary Coupler Insert

For:
At:

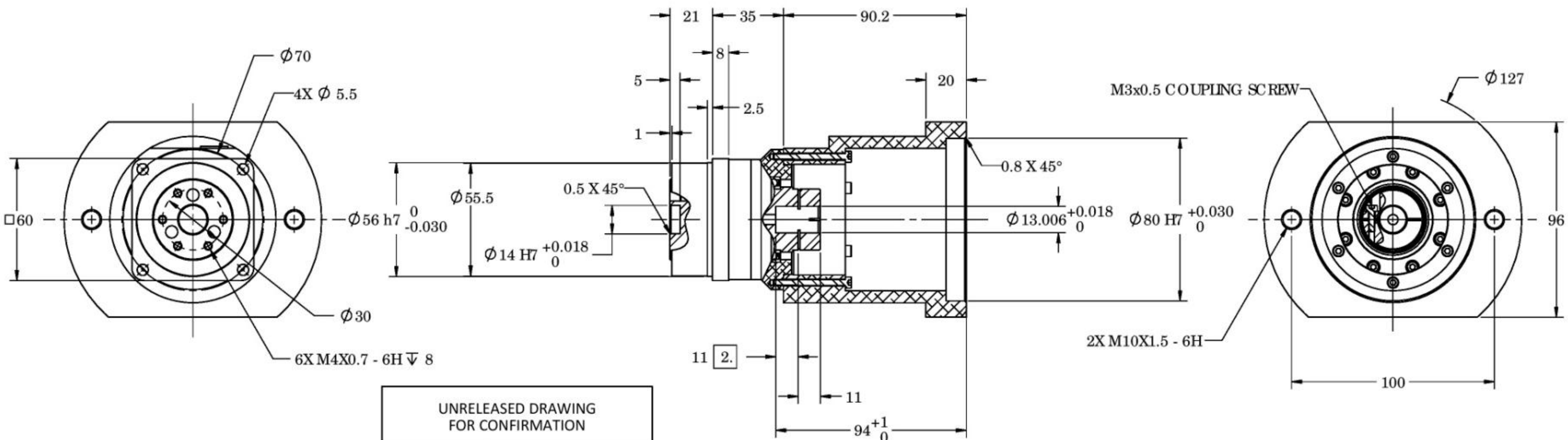
DRAWN	JR	DATE	3/25/15	SHEET	1 of 1
APPROVED		SCALE	1:1	Pump to Planetary Coupler Insert	

SolidWorks Student Edition.
For Academic Use Only.


Last Edited: 4/5/2015
PSI
Bike Under Pressure
Parker
For:
At:
Cal Poly Chainless Challenge
Rear Hub
DRAWN JR
APPROVED
DATE 3/25/15
SCALE 1:1
SHEET 1 of 1
DWG NO. Rear Hub
2 15/16"
Ø 1 3/8"
1/8"
Ø 2 3/4"
R3/8"
32 X Ø 3/32"
10.5°
3/16"
1 1/4"
SolidWorks Student Edition.
For Academic Use Only.

PRELIMINARY DRAWING

REVISIONS					
REV	ECA NO.	BY	CHKD	APPD	DATE
1	---	WTH			



UNRELEASED DRAWING FOR CONFIRMATION	
ENG	WTH
DATE	11/10/2014
SALES	
DATE	
CUSTOMER	
DATE	

NOTES:

1. DESIGN AND/OR HANDLING MUST BE IN COMPLIANCE WITH TECHNICAL INFORMATION FOUND IN HD CATALOG.
2. H7 TOLERANCE ZONE; TOLERANCE IN CLAMP AREA CAN BE ±0.1.
3. DIMENSIONS ARE IN MILLIMETERS UNLESS OTHERWISE NOTED.
4. MOTOR MOUNTING HARDWARE INCLUDED WITH UNIT.
5. CUSTOMER ORDER P/N: 7334356005.

MATERIAL:

ISSUE DATE			
CONTRACT NO.			
DRAFTSMAN WTH	11/10/14	CHECKER	
MAT'L ENG		DESIGN ENG	
MFG. ENG.		QUALITY	
SUBMITTED			
SIMILAR TO			

HARMONIC DRIVE LLC
247 LYNNFIELD STREET
PEABODY MASS 01960-4905
www.harmonicdrive.net

TITLE: INSTALLATION DRAWING
HPG-14A-05-BL1-ZZ-F0-SP4356

SIZE	FSCM	DWG. NO.	REV
C	34519	7379634	1
SCALE 1:1.5	WEIGHT		SHEET 1 OF 1

16. Appendix G: Cost Analysis

Table of Prototype Cost (two pages long)

Custom Parts	Material	Standard parts	Fixturing	Tooling	Machining (Hours)	Fabrication (Hours)	Assembly (Hours)	Setup (Hours)	Machine	Labor Cost	Total
Front Drive Unit		$3.00					2.00			$120.00	$123.00
Planetary/pump coupler shaft	$0.00	$0.00	$0.00	$0.00	10.00	0.00	0.00	1.00	Haas ST30 CNC Lathe	$660.00	$660.00
Planetary/pump coupler insert	$0.00	$0.00	$0.00	$100.34	3.00	0.00	2.00	1.00	Haas ST30 CNC Lathe	$360.00	$460.34
Coupler housing	$150.00	$2.00	$0.00	$0.00	21.00	0.00	1.00	1.00	Haas ST30 CNC Lathe	$1,380.00	$1,532.00
Gearbox assembly	$0.00	$5.00	$0.00	$0.00	2.00	0.00	2.00	0.20	Haas VF2 CNC Mill	$252.00	$257.00
L-Bracket	$30.93	$0.00	$0.00	$0.00	6.00	0.00	0.20	0.20	Haas VF3 CNC Mill	$384.00	$414.93
Rear Drive Unit		$1.00					0.50			$30.00	$31.00
Rear dropout	$95.16	$0.00	$0.00	$0.00	15.00	0.00	0.50	0.10	Haas VF23 CNC Mill	$936.00	$1,031.16
Motor mount	$80.54	$4.00	$0.00	$0.00	22.00	0.00	0.20	1.00	Haas VF2 CNC Mill	$1,392.00	$1,476.54
Pinion	$28.02	$0.00	$0.00	$0.00	5.00	0.00	0.30	0.30	Haas TL1 CNC lathe	$336.00	$364.02
Gear	$117.02	$0.00	$0.00	$0.00	2.00	0.00	0.00	0.25	Haas TL1 CNC lathe	$135.00	$252.02
Clutch/gear adatper plate	$12.00	$0.00	$0.00	$0.00	25.00	0.00	1.50	1.00	Haas VF2 CNC Mill	$1,650.00	$1,662.00
Rear wheel	$0.00	$40.00	$0.00	$81.23	10.00	0.00	1.00	0.17	Haas TL1 CNC	$670.00	$791.23
Frame	$160.00	$0.00	$0.00	$0.00	0.00	55.00	0.00	0.00	Licoln Tig Welder	$3,300.00	$3,460.00
										Total, Custom Parts	$12,392.24

Purchased Components	Cost	Assembly (Hours)	Labor Cost	Total
Frame Components				
Front fork	$0.00	1.50	$90.00	$90.00
Brakes	$43.73	1.00	$60.00	$103.73
Seat	$25.00	0.75	$45.00	$70.00
Front Wheel	$0.00	0.10	$6.00	$6.00
Handle Bar	$0.00	0.20	$12.00	$12.00
Brake Cables	$10.00	1.00	$60.00	$70.00
Cable Housing	$15.00	2.00	$120.00	$135.00
Stem	$0.00	0.05	$3.00	$3.00
Front Drive Unit Components				

Assembly/ Setup Rate ($/hr)	60
Machining Rate ($/hr)	60
Fabrication Rate ($/hr)	60

Total Cost (1 unit):	$15,071.41

Planetary	$737.00	0.10	$6.00	$743.00
Cranks	$10.00	0.20	$12.00	$22.00
Rear Drive Train				
Mounted Bearings (2)	$50.00	0.20	$12.00	$62.00
Spacers	$4.67	0.10	$6.00	$10.67
Clutch	$175.00	3.00	$180.00	$355.00
Rear Drve Shaft	$26.77	0.50	$30.00	$56.77
Hydraulic Components		8.00	$480.00	$480.00
Pump	$0.00		$0.00	$0.00
Motor	$0.00		$0.00	$0.00
Accumulator	$0.00		$0.00	$0.00
Reservoir	$0.00		$0.00	$0.00
Ball Valves (4)	$0.00		$0.00	$0.00
Check Valves (4)	$0.00		$0.00	$0.00
Pressure Gauge	$0.00		$0.00	$0.00
JIC fittings	$400.00		$0.00	$400.00
Low Pressure hose	$0.00		$0.00	$0.00
High Pressure hose	$60.00		$0.00	$60.00
Pipe Fittings	$0.00		$0.00	$0.00
Hydralic fluid	$0.00		$0.00	$0.00
			Total, Purchased Parts:	$2,679.17

Table of Mass Production Unit Cost (two pages long)

Custom Parts	Material	Standard parts	Fixturing	Tooling	Machining (Hours)	Assembly (Hours)	Set Up (Hours)	Machine	Labor Cost	Total
Front Drive Unit		$1.50				0.50			$30.00	$31.50
Planetary/pump coupler shaft	$5.00	$0.00	$0.00	$0.44	0.07	0.00	0.03	Haas ST30 CNC Lathe	$6.00	$11.44
Planetary/pump coupler insert	$3.00	$0.00	$0.00	$0.14	0.05	0.00	0.03	Haas ST30 CNC Lathe	$5.00	$8.14
Coupler housing	$90.00	$0.00	$0.04	$0.20	0.42	0.00	0.03	Haas ST30 CNC Lathe	$26.80	$117.04
Gearbox assembly	$100.00	$40.00	$0.02	$0.20	1.00	0.50	0.13	Haas VF2 CNC Mill	$98.00	$238.22
L-Bracket	$3.00	$0.00	$0.00	$0.04	0.17	0.00	0.03	Haas VF3 CNC Mill	$12.00	$15.04
Rear Drive Unit		$1.00				0.50			$30.00	$31.00
Rear dropout	$25.00	$0.00	$0.00	$0.20	0.07	0.00	0.10	CNC Water Jet Cutter	$10.00	$35.20
Motor mount	$45.00	$0.00	$0.04	$0.20	0.75	0.00	0.03	Haas VF2 CNC Mill	$47.00	$92.24
Pinion	$2.00	$0.00	$0.00	$0.04	0.08	0.00	0.03	Haas TL1 CNC lathe	$6.80	$8.84
Gear	$80.00	$0.00	$0.00	$0.08	0.08	0.00	0.03	Haas TL1 CNC lathe	$6.98	$87.06
Clutch/gear adatper plate	$8.00	$0.00	$0.40	$0.04	2.00	0.00	0.33	Haas VF2 CNC Mill	$140.00	$148.44
Rear wheel	$10.00	$0.00	$0.00	$0.02	0.17	1.00	0.17	Haas TL1 CNC	$80.00	$90.02
Frame	$100.00	$0.00	$2.00	$0.20	10.00	0.00	1.00	Licoln Tig Welder	$660.00	$762.20
									Total, Custom Parts:	$1,644.88

Purchased Components	Cost	Assembly (Hours)	Labor Cost	Total
Frame Components		0.20	$12.00	$12.00
Front fork	$20.00	0.08	$5.00	$25.00
Brakes	$40.00	0.03	$2.00	$42.00
Seat Post	$5.00	0.02	$1.00	$6.00
Seat	$5.00	0.02	$1.00	$6.00
Front Wheel	$20.00	0.02	$1.00	$21.00
Handle Bar	$5.00	0.02	$1.00	$6.00
Hand Grips	$2.00	0.02	$1.00	$3.00
Stem	$4.00	0.02	$1.00	$5.00
Front Drive Components		0.17	$10.00	$10.00
Planetary	$600.00	0.05	$3.00	$603.00
Cranks	$5.00	0.02	$1.00	$6.00
Rear Drive Train		0.25	$15.00	$15.00
Mounted Bearings	$4.00	0.02	$1.00	$5.00
Spacers	$0.10	0.02	$1.00	$1.10
Clutch	$100.00	0.02	$1.00	$101.00

Total Cost (1 unit):	$3,865.32
Total Cost (500 units):	$1,932,660.00

Rear Drve Axle	$5.00	0.02	$1.00	$6.00
Hydraulic Components		0.83	$50.00	$50.00
Pump	$425.00		$0.00	$425.00
Motor	$425.00		$0.00	$425.00
Accumulator	$171.00		$0.00	$171.00
Reservoir	$10.00		$0.00	$10.00
Ball Valves (4)	$10.00		$0.00	$10.00
Check Valves (4)	$20.00		$0.00	$20.00
Pressure Gauge	$5.00		$0.00	$5.00
JIC fittings	$150.00		$0.00	$150.00
Low Pressure hose	$4.00		$0.00	$4.00
High Pressure hose	$60.00		$0.00	$60.00
pipe fittings	$15.00		$0.00	$15.00
Hydralic fluid	$2.34		$0.00	$2.34
			Total, Purchased Parts:	$2,220.44

17. Appendix H: Point Simulation

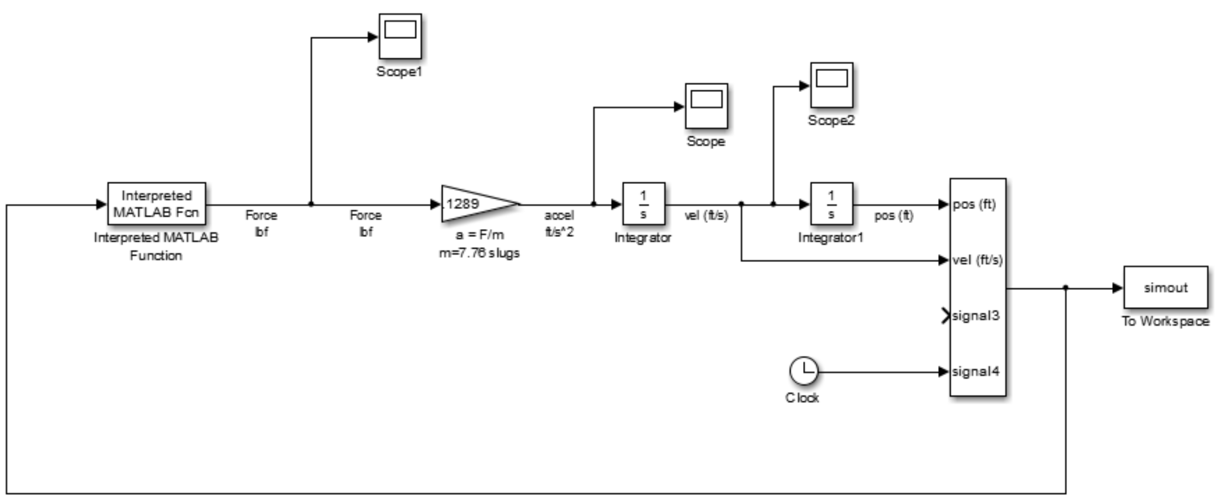
	Max points	2nd Overall	3rd Overall	4th Overall	Potential Win	Potential Win	Chain Deduct	2nd/3rd	No Regen	Poor Judging	Poor racing	Fail to Race
Innovation	100	75	50	25	100	75	75	75	75	10	100	100
Reliability and Safety	100	75	50	25	10	50	50	50	50	10	100	100
Manufacturability Workmanship	100	75	50	25	10	50	50	75	75	10	100	100
Best Design by Peers	100	75	50	25	10	25	25	50	50	10	100	100
Cost Analysis	100	75	50	25	10	50	50	75	75	10	100	100
Best Sprint Race Time	200	150	100	50	20	200	200	100	100	20	20	0
Best Relay Team	200	150	100	50	100	20	20	150	150	20	20	0
Efficiency Challenge	200	150	100	50	150	150	150	100	100	150	20	0
Winner of the Time Trial	200	150	100	50	200	100	100	150	150	200	20	0
Best Paper Presentation	200	150	100	50	200	100	100	100	100	200	20	0
Deductions for chains							-100					
Energy Recovery Bonus	200	200	200	200	200	200	200	200	0	200	200	200
TOTAL	1700	1325	950	575	1010	1020	920	1125	925	840	800	700

18. Appendix I: System Simulation

The system simulation was conducted with Simulink/MATLAB. The bicycle and rider were modeled as a 250 pound mass being pushed by a motive force on a frictionless surface. This simplification ignores all friction other than aerodynamic drag and rotational inertia of the wheels. There were several other key assumptions:

- Aerodynamic drag coefficient is constant at all speeds
- Rider torque is sinusoidal with a peak at 1200 in-lbf
- Rider torque is constant from 0 to 70 rpm, and linearly decreases to 0 at 130 rpm
- Pump efficiency is a curve fit from Parker's data for F11-5
- Pump efficiency is 10% if the curve fit reports less than 10%
- Only losses from cranks to rear wheel are in pump and motor; fluid losses are ignored
- Rider matches pedaling speed to bicycles actual speed (the cranks and rear wheel are rigidly connected, e.g. compressibility effects of fluid are ignored

The Simulink model calculates a motive force base on the bicycle's current speed, divides by the mass to get an acceleration and integrates twice to get position and velocity data. The Simulink model and the MATLAB function that is called are below.




```
function force = forceout(u)
%this funcion calculates a new force pushing the bicycle
%the rider matches the speed with pedaling
vel = u(2); %speed of bicycle in ft/s

r = 14; %rear wheel radius in inches
gear_front = 15; %front drive unit speed increase
gear_back = 4.29; %rear drive unit speed decrease
sys_gear = gear_front/gear_back; %system gearing

C_D = 1; %coefficient of drag for rider
```

```
rho = .00329; %air density, slugs/ft^3
A = 12; %frontal area of rider and bike, ft^2

%this logic starts the system
%this is rather than using a step input in the simulink model
if vel < .01
    vel = .01;
end

%the angular speed in the front and back are calculated
%from the velocity and differ by the system gear ratio
angular_speed_back = (vel/r)*12; %calc angular speed in rad/s
angular_speed_front = angular_speed_back*(1/sys_gear);

%curve fit F11-5 pump efficiency from Parker data
eta = .1586*log(angular_speed_front*gear_front)+.2388;

%if the efficiency function returns lower than 10%, keep 10%
%this is to prevent a zero or negative efficiency from returning
%furthermore the curve fit is less accurate at low speeds
if eta < .1
    eta = .1;
end

%This sets up the torque profile
%The peak torque input by the rider is 1200 in-lb up to 70 rpm (7.33 rad/s)
%this decreases linearly to zero at 130 rpm
if angular_speed_front <= 7.33
    torque = (1200*eta^2)/sys_gear;
elseif angular_speed_front > 7.33
    torque = ((-191*angular_speed_front+2600)*eta^2)/sys_gear;
    if torque < 0
        torque = 0;
    end
end

%calculate the motive force on the bike based on the peak torque (assumed
%sinusoidal) and the aerodynamic drag
force = (.707*.5*torque)/r - .5*C_D*rho*vel^2*A;

%this code was used during debugging to make sure the force was never less
%than zero, the code doesn't need it anymore.
% if force < 0
%     force = 0;
% end

end
```

```
Error using forceout (line 4)
Not enough input arguments.
```

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