Biodiesel Sustainability in Malik, Senegal

A Baccalaureate thesis submitted to the
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Bachelor of Science

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by

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ABSTRACT

This report was created to show the ideation, research, design, analysis, fabrication and testing requirements of the mechanical oil expeller, power transmission system, and engine exoskeleton created for the *"Biodiesel Sustainability in Malik, Senegal"* multi-disciplinary project. The project was completed to fulfill the requirements necessary to receive a degree in *Bachelors of Science for Mechanical Engineering Technology* from the *University of Cincinnati, School of Dynamic Systems, College of Engineering and Applied Science*. The engineering group for this interdisciplinary project focused on creating this three part system to yield a bio-diesel fuel source for small farms in Malika Senegal, Africa.

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INTRODUCTION

PROBLEM DEFINITION

The UC Forward initiative is a program that pairs students, faculty, and outside experts together to create innovative solutions to real world problems. Recently, they have secured funding to establish a pilot farm site in *Malika, Senegal, West Africa*. This farm site would utilize biodiesel technology as an entire system to harvest the jatropha plant, refine its oil into biodiesel fuel, and use that fuel to power equipment and other applications used within the community. A more streamlined, comprehensive, and independent bio-fuel process would allow them to create a sustainable and price-effective fuel source, providing additional jobs in agriculture, manufacturing, and maintenance, as well as set a precedent in regards to the full capabilities of a bio-fuel-centered community.

This interdisciplinary project, consisting of students from *College of Engineering and Applied Science, the College of Design, Art, Architecture, and Planning*, the *College of Business*, and the *College Conservatory of Music E-Media Program*, would focus on a practical and feasible method to implement this pilot farming site. The students from the *College of Engineering and Applied Science* would primarily be responsible for three separate segments. The first is a power transmission system for changing output speed and torque on a five horse power diesel engine, to be completed by James Anderson. The second is an exoskeleton to protectively house and transport the engine, to be completed by Reece McNulty. The last segment is a mechanical oil press to extract oil from the jatropha seeds, to be completed by Mike Compton. This project provides the opportunity to not only solve a problem that will have a resounding effect within the community of *Malika, Senegal*, but also foster positive relations and general good-will between two starkly different cultures.

Using strictly mechanical means, the price of this project will be kept low and functionality high. With the mindset of longevity as a priority, the people of Senegal can be given a tool that will be of benefit for many years and allow future advancements as other applications are implemented. Initial prototypes will be produced in Cincinnati, with a goal of reproduction of these products in Senegal. Designing components capable of being produced in country will aid in creating jobs and refining technical skills and knowledge, which can then be translated into multiple educational opportunities to be self-supportive in the design of other attachments, furthering the advancement of Senegal.

BACKGROUND

The current machinery used to operate the applications used to farm jatropha are un-optimized, out of date, and are mostly human powered, all of which are causing a loss of profit through fuel costs and wasted time. Additionally, the price of oil presses used to expel the seed's rich oil, exceeds the reasonable price limits of the community. Creating this three part system, will provide the community with an optimized system that will drive fuel costs down, provide ease of use for the workers, allow for a greater crop yield and be within the community price cap.

MECHANICAL OIL PRESS

OIL PRESS RESEARCH AND CUSTOMER REQUIREMENTS

Research

The most basic method for expelling oil from organic matter is with a screw type oil expeller. The basic components, as laid out in a study by the University of Ilorin, Nigeria (1) consist of a cylindrical barrel, a shaft with an external thread that is fitted into the barrel, an inlet for the organic material that is to be pressed, an outlet for the expelled oil, and an outlet for the excess material, or "cake". As the material is fed into one end of the barrel, the screw is turned, and the material is pushed toward the other end of the barrel. As the material is condensed, friction causes the heat in the material to rise, and with the increasing pressure, oil is separated from the rest of the organic matter. This oil drains down into the oil outlet, where is it collected for any additional processing, and the newly formed cake is expelled out the cake outlet.

Figure 1- Screw-driven oil press

The original method for this process, as described in the Wikipedia page for expeller pressing (2) is referred to as "continuous screw", as the helical thread is continuous and uninterrupted from the beginning of the shaft to the end. In 1900, a new method known as "Interrupted Screw" was invented by Valerius Anderson. As the name implies, interruptions were put into the thread of the shaft, which allows material to accumulate more before being moved down the barrel, thus increasing pressure and friction, yielding greater oil output and a more consistent cake. A further modification to this design has since been implemented, referred to as resistor teeth. These are teeth fitted onto the inside of the barrel, in the gaps left by the interrupted screw. These teeth provide further agitation while material collects between the gaps in the threads, increasing friction and further increasing oil yield.

Hydraulic Cylinder Driven Oil Press

A more common means of extracting oil today is through use of a hydraulic cylinder. As high pressure is needed to extract oil from organic matter, the high pressures that can be achieved with hydraulic cylinders makes them ideal for this application. Most commercial oil presses utilize this type of system. A hydraulic cylinder is used to drive a piston down a shaft into which organic matter is fed. As the piston compresses the material, the oil is expelled (3). As with the screw type method, the oil drains into an oil outlet while the cake is expelled through a separate outlet. The advantage with hydraulics is the greater ease at which the forces needed to expel the oil can be generated. Commercial hydraulic pumps are must more complex than a basic screw type oil expeller, requiring a higher level of knowledge to maintain and repair.

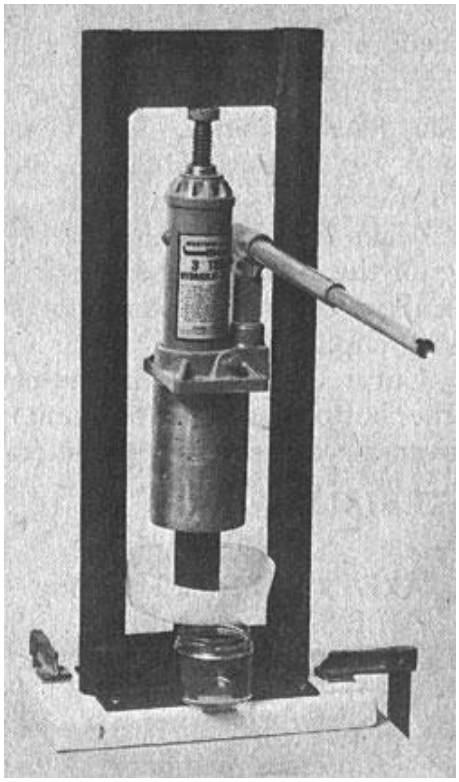


Figure 2 - Homemade hydraulic oil press

Customer Surveys

Surveys were conducted in order to assess the needs of the people of Malika, Senegal. Because of the language and distance barriers, only three surveys were able to be conducted for the oil expeller, but the information contained therein proved invaluable in terms of determining customer needs. Each survey lists different aspects of the product, and the survey taker rates how important each aspect is on a scale of 1 to 5, with 5 being the highest importance.

How important is each feature to you for the design of an oil press?
Please circle the appropriate answer. 1 = low importance 5 = high importance

							Average
Safety	1(1)	2(1)	3	4(1)	5	N/A	2.33
Durability	1	2	3	4	5(3)	N/A	5
Ease of Operation	1	2	3	4	5(3)	N/A	5
Price	1	2	3	4	5(3)	N/A	5
Ease of Manufacture	1	2	3(2)	4	5(1)	N/A	3.66
Efficiency	1	2	3	4	5(3)	N/A	5
Oil Output	1	2	3	4	5(3)	N/A	5
Size	1	2(2)	3(1)	4	5	N/A	2.33

How satisfied are you with the current method of extracting biodiesel fuel from Jatropha seeds?
Please circle the appropriate answer. 1 = very UNsatisfied 5 = very satisfied

							Average
Safety	1	2(1)	3	4(1)	5(1)	N/A	9.66
Durability	1(1)	2(2)	3	4	5	N/A	1.66
Ease of Operation	1(1)	2(2)	3	4	5	N/A	1.66
Price	1(3)	2	3	4	5	N/A	1
Ease of Manufacture	1(1)	2	3(2)	4	5	N/A	2.33
Efficiency	1(1)	2	3(2)	4	5	N/A	2.33
Oil Output	1	2(1)	3(2)	4	5	N/A	2.66
Size	1	2	3(2)	4(1)	5	N/A	3.33

How much would you be willing to fund for this project? $50-$100(3), $100-$200, $200-$500, $500-$1000, $1000-$2000

Thank you for the feedback!

Quality Function Deployment

These survey results were then translated into a Quality Function Deployment, or QFD. A QFD is a tool that cross references aspects rated by the customers as mentioned in the survey above with engineering characteristics to help meet those requirements. The end result is a percentage associated with each product aspect based on that aspects relative weight. Using these relative weight ratings, each requirement then can be ordered based on its customer importance. From the QFD seen below in Table 1, it can be seen that the most important feature is Price, while the least important feature is Safety.

Table 1 - Oil Press Quality Function Deployment

Mike Compton UC Forward Senegal Oil Press 9 = Strong 3 = Moderate 1 = Weak	Guards for grinding mechanism and heating element	Recommended torque specifications for all bolts	Emergency Release	Material Selection	Designed for multiple loading conditions	Quick Disconnect/Coupler	Color Coding System for Different Applications	Maintenance Access Panel	Designed for User Manufacturing	Heating Element for Seeds	Calculat Feedrate for Highest Oil Output	Ability to be Disassembled		Customer importance	Current Satisfaction	Planned Satisfaction	Improvement ratio	Modified Importance	Relative weight	Relative weight %
Safety	9	3	9	1	1		3	1						2.3	3.6	4	1.1	2.6	0.04	4%
Durability	3	3		9	3			3						5	1.6	3.5	2.2	10.9	0.19	19%
Ease of Operation			3			9	9	1			1	3		5	1.6	4	2.5	12.5	0.22	22%
Price	1		1	3					1					5	1	3	3.0	15.0	0.26	26%
Ease of Manufacturability	1		1	1		1			9			1		3.6	2.3	3.5	1.5	5.5	0.10	10%
Oil Output								1		9	9			5	2.6	4	1.5	7.7	0.13	13%
Size	1		1	1						3				2.3	3.3	4.5	1.4	3.1	0.05	5%
Abs. importance	1.39	0.71	1.47	2.70	0.62	2.06	2.10	0.97	1.12	1.37	1.43	0.75	16.7					57.3	1.0	1.0
Rel. importance	0.08	0.04	0.09	0.16	0.04	0.12	0.13	0.06	0.07	0.08	0.09	0.04	1.0							

Customer Requirements

Using these relative weights to rank the importance of each product aspect, a general picture of customer requirements can then be constructed. For this oil press, it was determined that the customer, being the residents of Malika, Senegal, needed a durable, easy to operate, and cost effective means of efficiently expelling oil from Jatropha seeds for the purpose of fueling a sustainable farming community.

MECHANICAL OIL EXPELLER DESIGN

Preliminary Ideas and Concept Selection

Upon researching many US and International Patents, it was found that there were three main design variations with mechanical oil expellers. These variations are the number of screws present (one or two), the type of screw, and the drive system.

The first design decision that was made was the drive system. The three drive systems considered were a gear drive, a chain drive, and a belt drive. Ethnographic field research was conducted by students from the college of Design, Art, Architecture and Planning, with one of the main conclusions being the overabundance of dust, dirt, and sand particulates present in the air. Because of this environmental consideration, gear and chain drives were abandoned due to their lubrication requirements. It was feared that the large amounts of aforementioned particulates in the air stood too great a risk of compromising a gear or chair drive. For this reason, the chosen drive was a belt drive. Furthermore, it was decided that due to the high torque and low rotational speed required, 450 foot pounds and 55 rotations per minute, respectively, only a synchronized drive would be appropriate.

Initially, it was determined that two screws, rotating opposite each other, would be desired over one, on the notion that more material could pass through, per unit time, a chamber with two screws as opposed to one. It was later determined that using a synchronized belt to drive two screws in opposing directions would require not only a two sided synchronous belt, but two additional shafts. For the sake of cost and simplicity, it was decided that one screw was sufficient to achieve the end goal.

The final design variation consisted of continuous flights versus interrupted flights. The mechanical advantage of interrupted flights is a cavity, existing in the space of the interruption that allows material to build up, as there are no screw flights to drive material forward. This variation can be seen in figure 3, between sections M and L. This buildup of material further elevates pressure, theoretically increasing efficiency. For this reason, it was decided that a single flight interruption would be implemented in the middle of the helix.

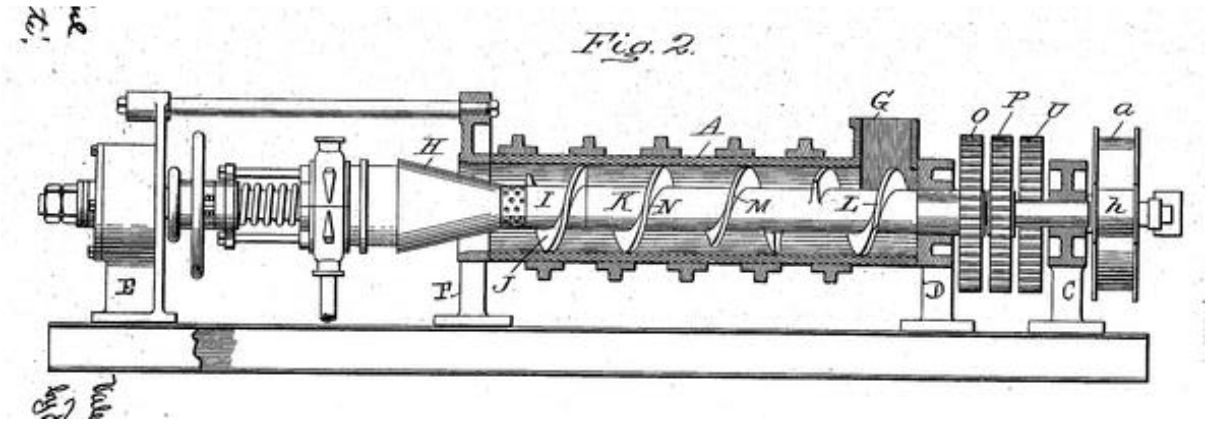


Figure 3- Traditional screw press example

In summary, the final concept selection consisted of a single screw oil expeller with interrupted flights, driven by a synchronous belt.

Drive Design

As stated before, due to the high torque and low rotational speed required for this application, a synchronous belt was thought to be the only suitable drive. This thought was confirmed upon attempting to select a V-Belt design. While numerous V-Belts were found that could handle the torque requirement, none were found that could meet 60 rotations per minute. The selected drive, utilizing a 1:2.25 reduction ratio, consisted of one Goodyear 28 tooth sprocket, one Goodyear 63 tooth sprocket, a Goodyear Y-1600 belt, and two QD style bushings.

Shaft Design

First, a nominal shaft length of 10.5 inches was estimated by allowing for two 5/8 inch thick bearings, four 1.8 inch thick bearing covers, a 2.5 inch section for the drive sprocket, and a 6.25 inch section for the screw. This length was later changed to 11 inches to allow for a retaining ring. Based on these initial assumed dimensions, along with a torque load of 450 foot pounds, a shaft minimum shaft diameter of 1.618 was found. A nominal value of 1.625 was chosen, as illustrated below.

$$\text{Minumum Diameter, D} = [\frac{32N}{\pi} * \sqrt{(\frac{Kt * M}{s'_n})^2 + \frac{3}{4} * (\frac{T}{S_y})^2}]^{1/3}$$

$$\text{D} = [\frac{32 * 2.5}{\pi} * \sqrt{\left(\frac{1.6 * 1363.25inlb}{14{,}288.4psi}\right)^2 + \frac{3}{4} * \left(\frac{5400inlb}{58000psi}\right)^2}]^{\frac{1}{3}} = 1.618\ inches$$

Nominal Value of 1.625 inches was chosen.

Next, the design of the threads was considered. Data on thread design in terms of efficiency and effectiveness could not be obtained. Because of this, a mathematical model was developed for the sole purpose of this project. Several approaches were taken, with the one deemed most appropriate relating the volumetric throughput of material per unit time along the helix to the volumetric output per unit time of oil. This model calculates an "ideal volumetric oil output", assuming 100% of the helix volume is taken up at any given time and 100% efficiency is present in expelling oil from the material. Based upon the specifications of similar products, it is estimated that the actual oil output will be 10% to 15% of the ideal volumetric output. A secondary goal of this project will be to test for the presence of a constant ratio between the ideal volumetric output and the actual oil output for a variety of oil seeds. This mathematical model can be seen below.

$$\frac{16.33in^3}{6.25\ revolutions} = 2.61in^3/_{rev}$$

$$\frac{2.61in^3}{rev} * 55\frac{rev}{minute} = \frac{143.704in^3}{min} = \frac{8622.24in^3}{hour}$$

Solid Density of jatropha Seeds: 0.435 ounces/in^3

$$\frac{8622.24in^3}{hour} * \frac{0.435ounces}{in^3} = 3793.79\frac{ounces}{hour}$$

Average Oil Content of Jatropha Seeds: 37%

$$3793.79\frac{ounces}{hour} * 0.37 = 1403.70\frac{ounce\ of\ oil}{hour}$$

Jatropha Oil Density: 0.53 ounces/in^3

$$1403.70\frac{ounces}{hour} * \frac{1in^3}{0.53\ ounces} = \frac{2648.49in^3}{hour}$$

$$\frac{2648.49in^3}{hour} * 0.0164\frac{liters}{in^3} = 43.40\frac{liters}{hour}$$

Solid model

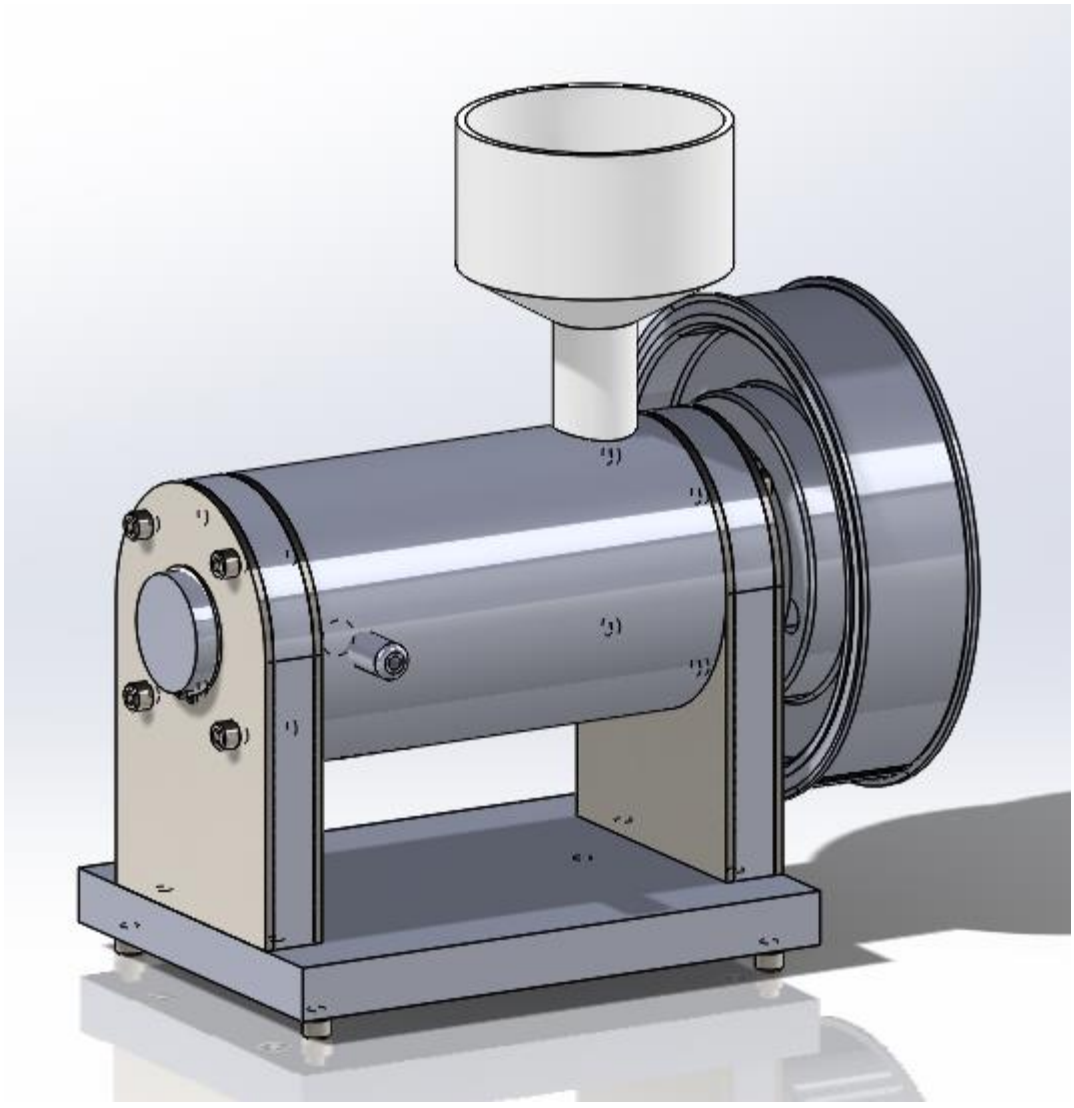


Figure 4 - Solid model of oil press

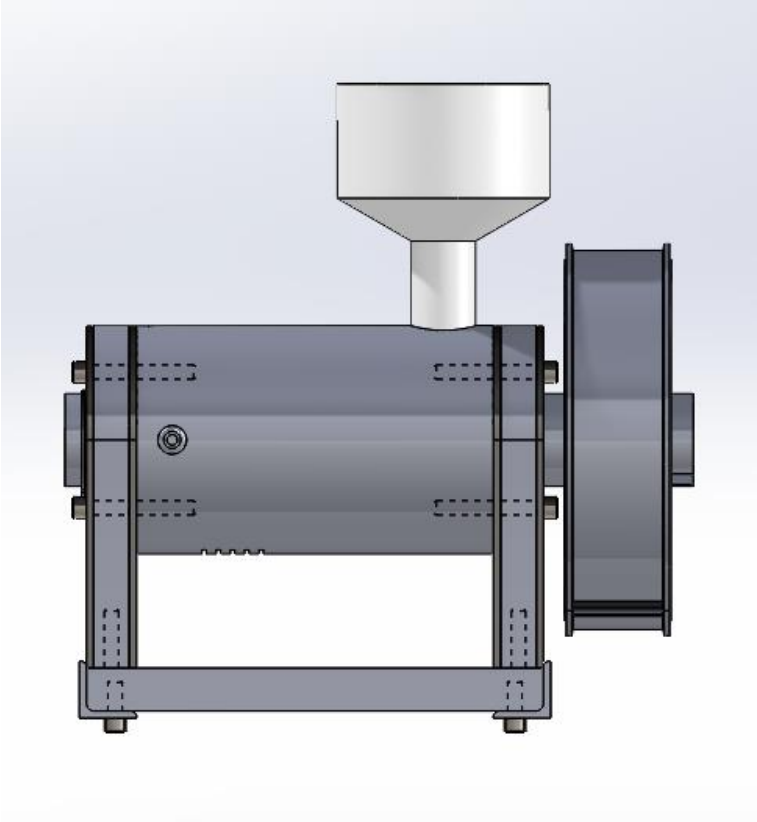


Figure 5 - Side View

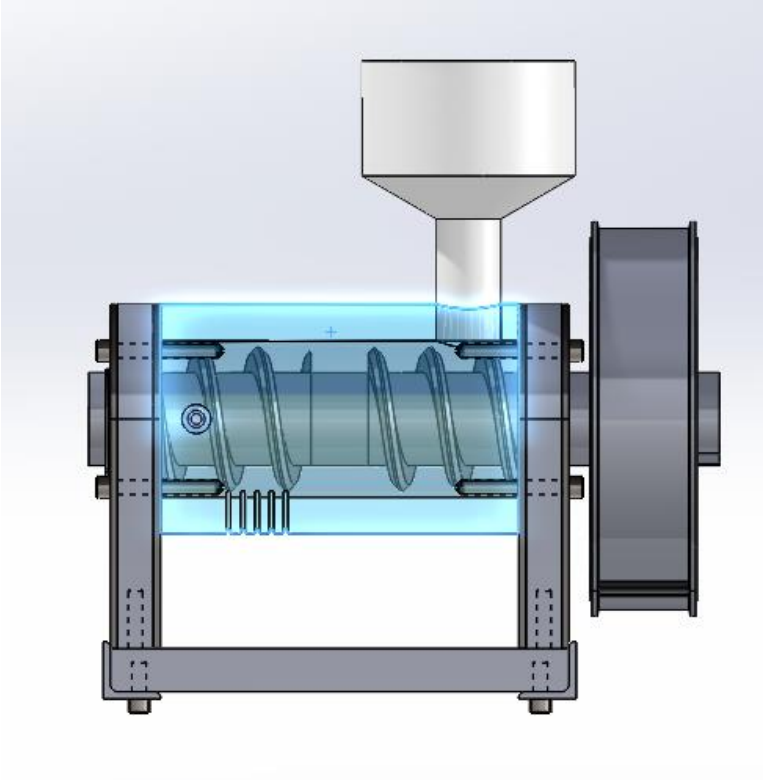


Figure 6 - Side View, Transparent Chamber

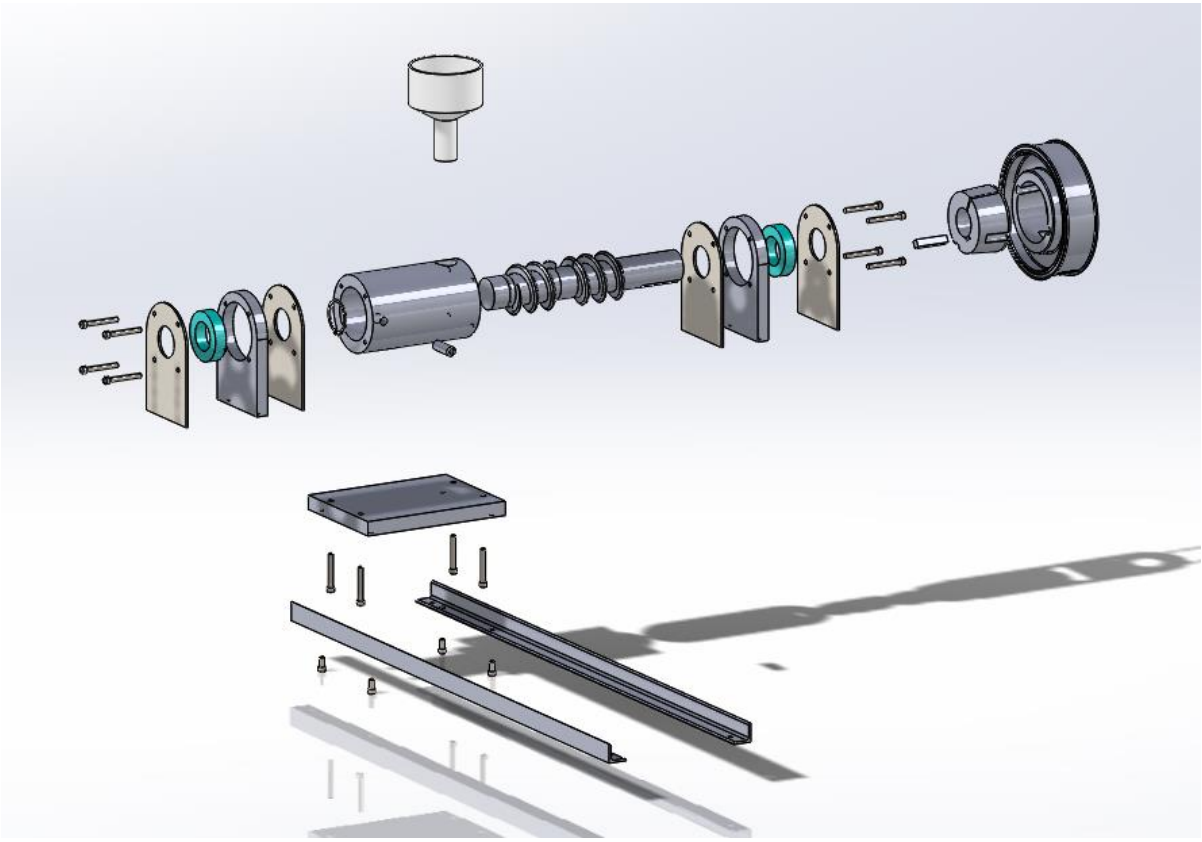
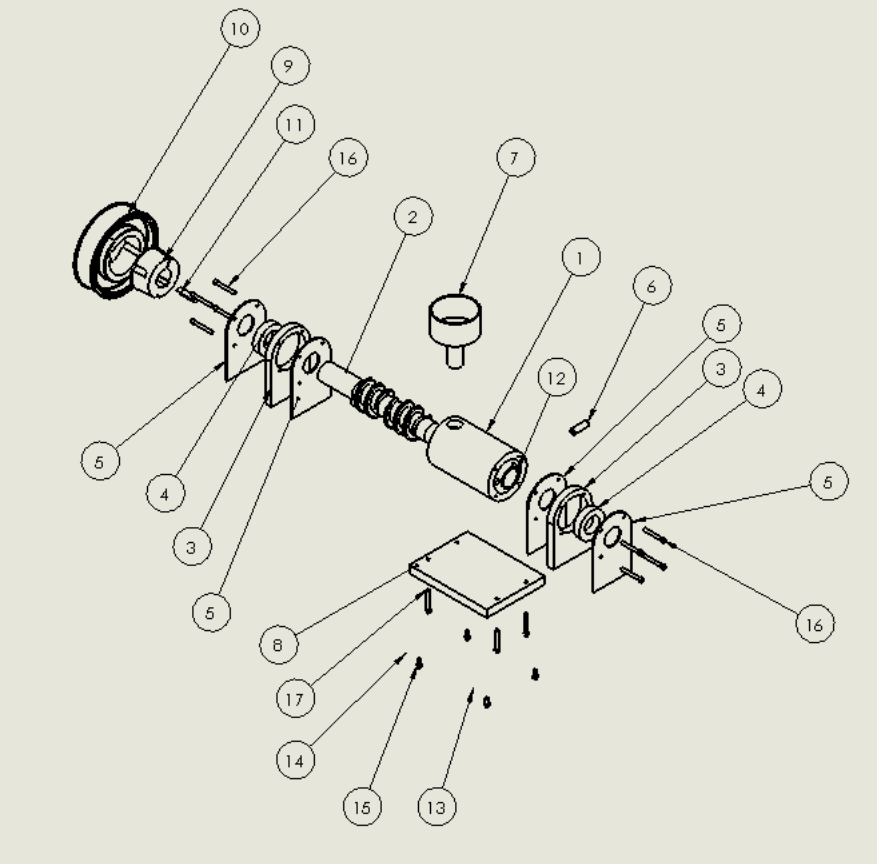


Figure 7 - Exploded View

Bill of Materials

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	Chamber		1
2	Screw		1
3	bearing housing		2
4	Bearing		2
5	bearing cover		4
6	bullet nose		1
7	funnel		1
8	Base		1
9	2517_1_1_2		1
10	8mx_67s_36		1
11	Key		1
12	RetainingRing		1
13	leck bracket		1
14	right bracket		1
15	HX-SHCS 0.25-20x0.5x0.5-N		4
16	HX-SHCS 0.25-20x1.875x1.375-N		8
17	HX-SHCS 0.25-20x1.75x1.25-N		4



Loading Conditions and Design Analysis

Based upon published research, it was determined that an operating pressure range of 50-150 bar (4) should be designed for. Additionally, it was determined, based upon video footage of other mechanical screw type oil expellers, that an operational speed of 60 rotations per minute was desired. Due to a lack of available design examples and/or guidelines, a maximum torque of 450 foot-pounds was designed for, based upon the engine horsepower and desire rotational speed of 60 rotations per minute. These were the loading conditions used for analysis of the mechanical screw type oil expeller design. It should be noted that after the pulley reduction system design was completed, the desired rotational speed was increased to 93 rotations per minute due to limitations present within the pulley reduction system design. Finite Element Analysis under the aforementioned loading conditions of 450 foot pounds and 725 and 2175 PSI (50 and 150 bar) was conducted using Solidworks SimulationXpress. It was found that the resulting Von Mises stress generated from mild 1020 steel acted upon by the given loading conditions fell at 13,889 PSI(96,453,088 N/m^2), which is 27% of the yield stress for 1020 steel, being 50,991 PSI(351,571,000 N/m^2). At 725 PSI (50 bar), it was found that the maximum foreseeable Von Mises stress was 4629 PSI (32,151,028 N/m2), which is 9% of the aforementioned yield strength for mild 1020 Steel. The same analysis was conducted for 1060 Aluminum alloy, with the resulting stress being 13730 PSI (95,349,480 N/m^2) at 2175 PSI (150 bar), and 4577 PSI (31,783,158 N/m^2) at 725 PSI (50 bar). These results are 345% and 115%, respectively, of the Yield Strength of 3971 PSI (27,574,200 N/m^2) for 1060 Aluminum Alloy. This result eliminated aluminum as a feasible choice for the chamber.

Table 2 - Loading Conditions

Preliminary Torque	450 ft-lbs
Preliminary Rotational Speed	60 RPM
Actual Rotational Speed	93 RPM
Actual Torque	254 ft-lbs
Predicted Operating Pressure	50-150 Bar

Table 3 - Fininte Element Analysis Results

1060 Aluminum	
Yield Strength	3,971 PSI
Maximum Stress at 725 PSI	4,577 PSI
Maximum Stress at 2175 PSI	13,730 PSI
1020 Steel	
Yield Strength	50,991 PSI
Maximum Stress at 725 PSI	4,629 PSI
Maximum Stress at 2175 PSI	13,889 PSI

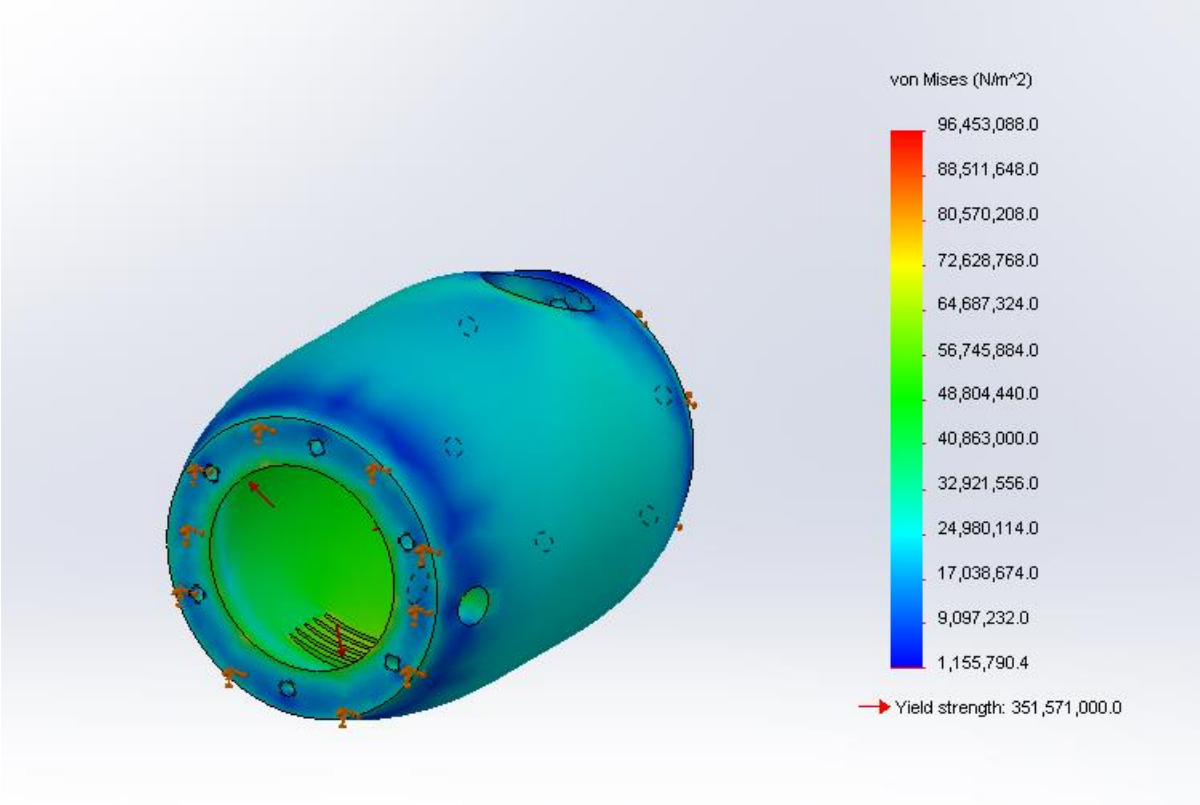


Figure 8 - Von Mises Stress for 1020 Steel at 2175 PSI

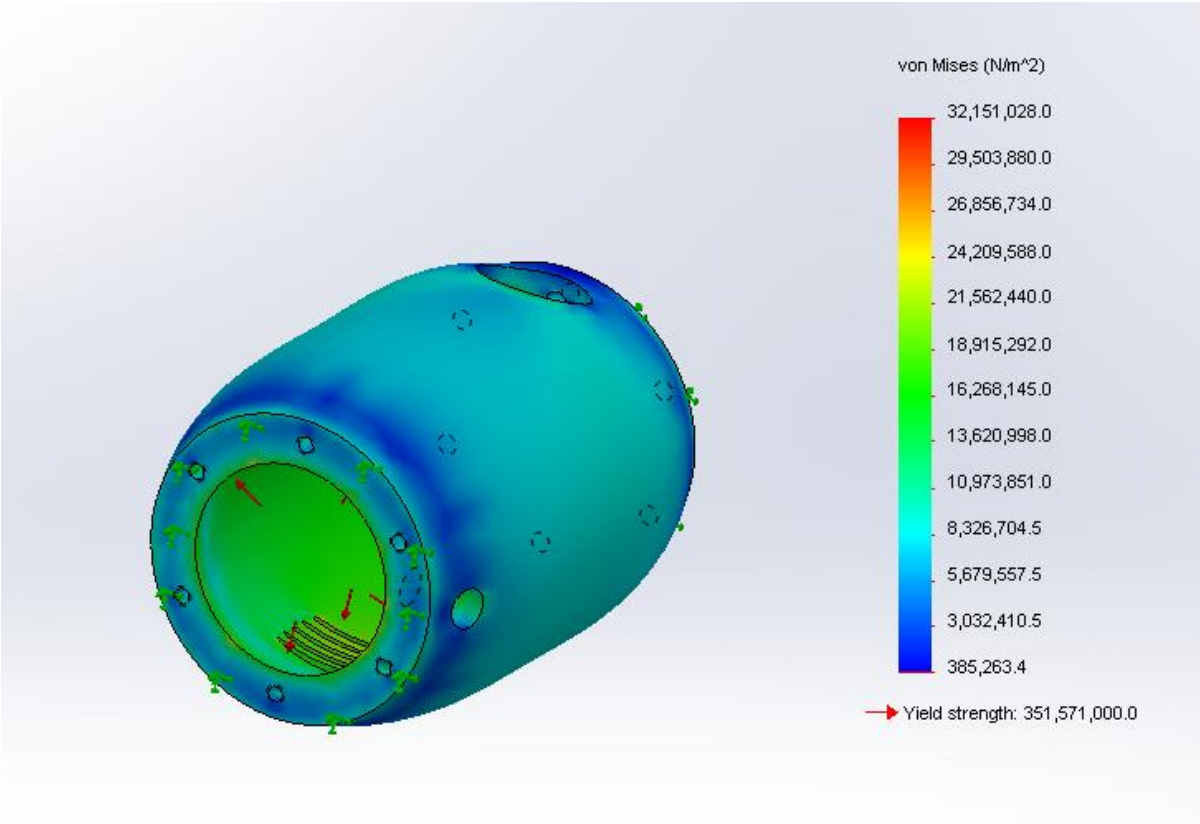


Figure 9 - Von Mises Stress for 1020 Steel at 725 PSI

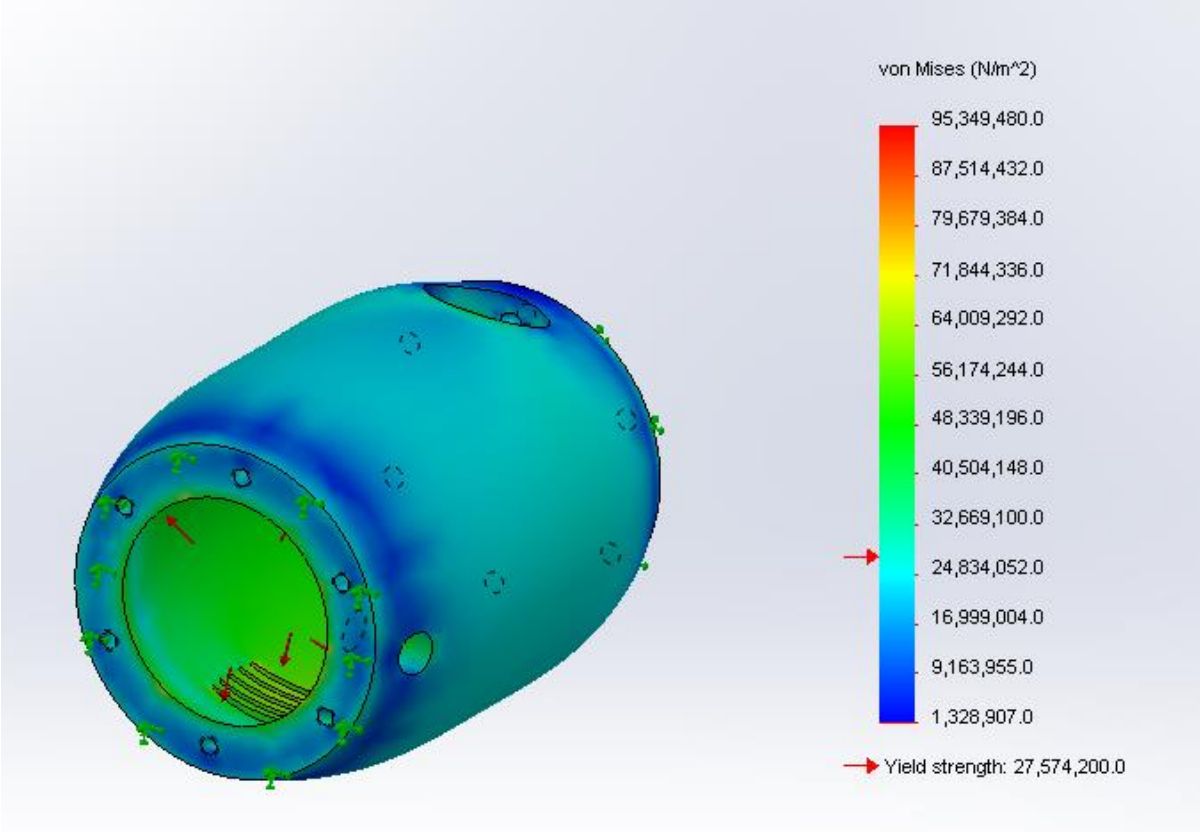


Figure 10 - Von Mises Stress for 1060 Aluminum Alloy at 2175 PSI

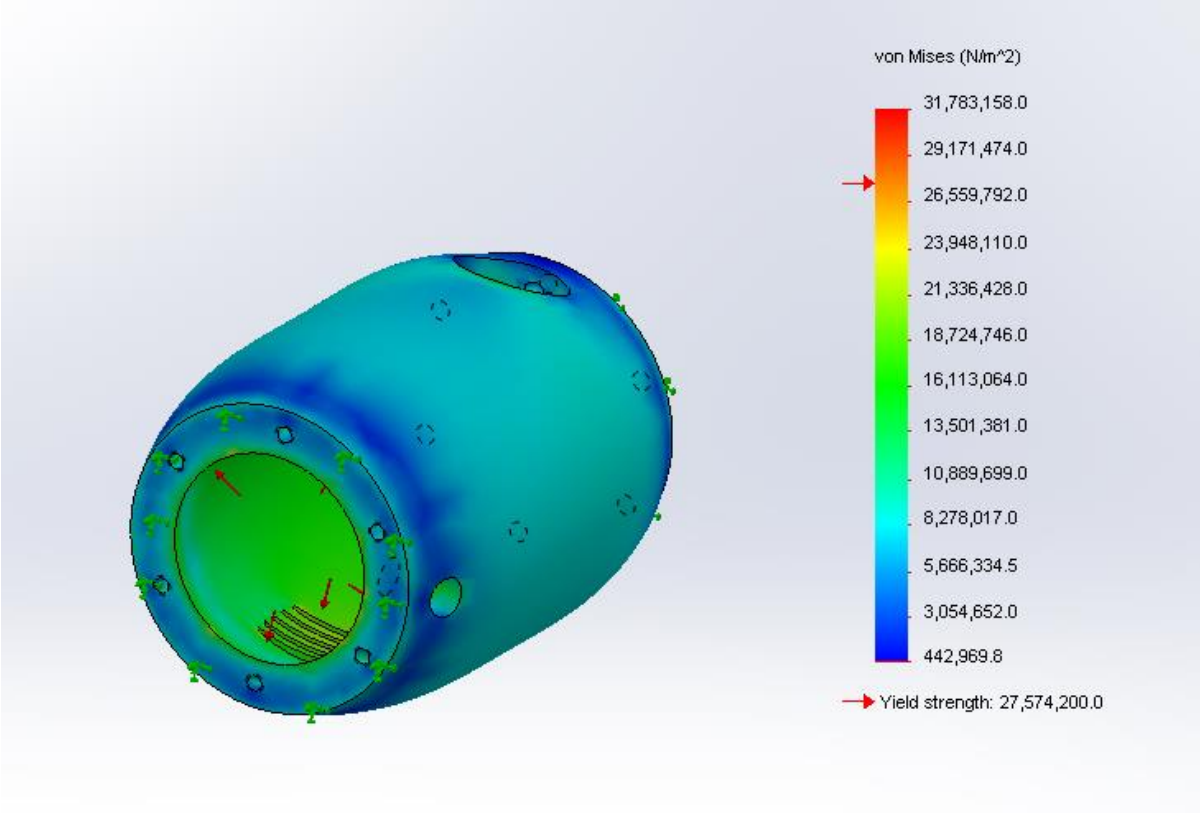


Figure 11 - Von Mises Stress for 1060 Aluminum Alloy at 725 PSI

OIL PRESS FABRICATION AND ASSEMBLY

A major consideration for this project was parts capable of being manufactured and repaired in Senegal. The town for which this press was made has available to it basic manufacturing resources such as lathes, drill presses, vertical mills, and welding equipment. All manufactured components were designed with creation from the aforementioned equipment in mind. Likewise, all manufactured components were manufactured using these components. One particularly noteworthy component was the screw. Originally, it was intended for this component to be turned down on a lathe from a single piece of steel round stock. After careful consultation from the Victory Parkway Machine Shop Manager, it was decided to machine the shaft to its nominal diameter of 1.625 inches, and then weld the flighting in place. After the shaft was machined, it was placed in a lathe, and a piece of 0.5 inch by 0.125 inch steel was tack welded onto the shaft such that the 0.5 inch side was vertical. The lathe was then put in neutral and spun by hand, while an arc welder was used to heat the flighting and wrap it around the shaft at a distance of one thread per inch. Once wrapped, the threads were then welded in place using Tungsten Inert Gas welding. No specialized assembly processes were needed.

TESTING PROCESS AND RESULTS

Because of the poisonous nature of Jatropha seeds, they cannot legally be imported without a permit. This restriction makes availability of Jatropha seeds very low. Therefore, a substitute was used for testing purposes. Initially, peanuts were deemed to be a sufficient substitute. Peanuts were to be placed into the hopper for a period of 1 hour, and the extracted oil measured. After initial testing, it was quickly determined that peanuts were an insufficient substitute. Next, sunflower seeds were fed into the hopper for 23 minutes. At the 23 minute mark, the sunflower seeds began to be rejected by the press.

It is believed that a much greater pressure is required to press oil from peanuts than that generated in the press. The resulting compound was peanut butter. Upon using sunflower seeds, a much more desirable result began to become apparent. What also became apparent is the inadequate sealing present in the press, as oil was seen leaking out primarily from the end of the shaft, opposite the drive sprocket. At the 23 minute mark, seeds began to be rejected from the hopper. Once the engine was stopped and the press disassembled, a hard black ring was discovered at the end of the screw shaft. Once the screw was removed, it was seen that hardened, dried, sunflower seeds, sans oil, were wrapped along the majority of the screw shaft, allowing no more seeds to enter. From this observation, it is believed that a critical design flaw can be determined. An initial assumption made, but not explicitly stated due to a lack of realization of its utter significance, is that the seed cake would be malleable enough to make a 90 degree turn at the very end of the shaft. After observing the dryness and hardness of the cake, along with how densely it was packed along the shaft, it is believed that the cake must exit the chamber in the direction of travel down the shaft, in order to prevent jamming.

RECOMMENDATIONS

Though the concept was proved absolutely through experimentation, two critical design flaws were detected, and recommendations to correct these flaws will now be addressed

- Commercially available presses should be disassembled in order to discern how the cake is allowed to exit along the length of the shaft
- More efficient methods of sealing the press such that oil and cake exit only through the intended outlets
- A lean manufacturing approach should be taken in order to minimize the amount of material used

In addition to these aforementioned recommendations, and due to the large scale of this oil expeller project, several smaller baccalaureate thesis topics (senior design projects) will now be suggested:

- A Lean Manufacturing design for the reduction of material used in a mechanical oil expeller
- A study of efficiency vs. cake outlet angle for a mechanical screw type oil expeller
- A study of temperature and pressure zones present within a chamber of a mechanical screw type oil expeller
- A study of efficiency vs. rotational speed for a mechanical screw type oil expeller
- A study of oil yield vs. oil seed used with a mechanical screw type oil expeller
- A study of the effect on oil yield and expeller efficiency from the implementation of a cooling system for a mechanical screw type oil expeller

ENGINE EXOSKELETON

RESEARCH, TECHNOLOGY, AND EXISTING PRODUCTS

Research

A portable generator is the closest thing found that resembled an engine exoskeleton. This portability allows for the user to transport the unit from site to site. The generator found had a similar design that goes with proposed concepts (6). The exoskeleton will serve the same duty along with much more. The problems with portable generators is that they are not built very robust. In Senegal the exoskeleton must be able to withstand a lot of abuse far more than the portable generator. Dan Bode was very informative with how the Senegalese work and that is they abuse machinery, because of this extensive durability is a must. Ease of operation and maintenance are the next two important design features that must be completed. These are objectives are important because of the obsolete machinery and processes that are available. The system cannot be overly designed so the Senegalese are able to repair the problem. Many of the Senegalese are illiterate so complicated or extensive procedures are impossible. Also the machinery in country is also primitive so all parts must be able to be replaced or manufactured with outdated machines.

Figure 12- Portable generator

Existing Technology

Below is a couple of pictures from the DAAP students that attended the farm in Malika. What they have created is very simple and only a starting point for the end result of a fully enclosed easy to use system. On the next page you will find pictures of what is currently in use in Senegal.

Figure 13-Senegalese Exoskeleton

There are many disadvantages to this design, with round tubing it is very difficult to fasten any new attachment to the system. This design has some downfalls including it takes two men to move the system which we are striving to go away from. With the setup currently used in Senegal they are only able to pump water and that is it. The final design will allow for this as well as be used a seed crusher and oil extractor.

Customer Needs, Survey, QFD

The customers for this product are very limited. It was designed for solely the Senegalese farmers that will be using this assembly. The customer needs were hard to attain since this is a completely new design. First a survey was submitted to our Colorado contact which he then forwarded to some of the farmers in Senegal the survey with results can be seen below in Table 4. From the survey it was shown that the Durability ease of operation and weight were the three biggest design drivers.

Table 4-Customer Survey

Please rate the following features with a level of importance

Please circle a number 1=lowest importance, 5=highest importance

							Average
Safety	1	2	3(2)	4	5(1)	N/A	3.66
Durability	1	2	3	4	5(3)	N/A	5
Ease of Operation	1	2	3	4(1)	5(2)	N/A	4.66
Ease of Maintenance	1	2	3	4	5(3)	N/A	5
Lightweight	1	2	3	4(1)	5(2)	N/A	4.66
Overall height	1(1)	2(2)	3	4	5	N/A	1.66
Ability to handle rough terrain	1	2	3	4	5(3)	N/A	5
Lifting mechanism for Engine	1	2(1)	3(1)	4(1)	5	N/A	3
Stabilizing mechanism	1	2	3(2)	4(1)	5	N/A	3.33
Ease of disassembly	1	2	3	4	5(3)	N/A	5
Versatile (different engines)	1	2	3	4	5(3)	N/A	5

How much would you be willing to fund for this project?

$50-$100x, $100-$200, $200-$500

With the survey results a Quality Function deployment table was created to weigh the design drivers these results are shown below in table 5.

Table 5- QFD

Reece McNulty Engine Exoskeleton Senegal Moving Forward 9 = Strong 3 = Moderate 1 = Weak	wheel size	Lifting mechanism	Brake selection	machineability	accessable Materials	lifespan	access doors/geometry	designed for multiple loading conditions	go-no-go gauge for pully guard	ground clearence		Customer importance	Designer's Multiplier	Current Satisfaction	Planned Satisfaction	Improvement ratio	Modified Importance	Relative weight	Relative weight %
Durability	9	9	3		3	9	9	9	1	1		5	1.1	1	4	4.0	25.0	0.25	25%
Safety	9	1			3		3	3	3	3		3.6	1	1	3.8	3.8	16.0	0.16	16%
Ease of Operation	9	9	1			9	3	9	3	3		4.66	1	1	4.8	4.8	21.0	0.21	21%
Ease of Maintenance		9		3	3	9	3					5	1	1	4	4.0	20.0	0.2	20%
Cost		1	1		3							2	1	1	5	5.0	11.0	0.11	11%
Engine Swap Capabilities	1			1								5	1	1	2	2.0	7.0	0.07	7%
Abs. importance	5.65	6.21	1.07	0.67	2.16	5.94	3.96	4.62	1.36	1.36	33.00						100.0	1	100%
Rel. importance	0.17	0.19	0.03	0.02	0.07	0.18	0.12	0.14	0.04	0.04									

Since durability was rated the highest the design multiplier was changed from 1 to 1.1 to further increase the overall durability of the system.

Product Objectives

The DAAP students return brought more feedback from the field. The following is a summary from Peter Middleton,

- The ground is sandy everywhere similar to the beach
- The machinist are very talented with welding and machining
- In the co-op in Malika there are 250 women working there so the engine will be expected to be able to be used by all.
- The Senegalese are very industrious they are more likely to "hack" something together than purchase something for ten dollars since money is so scarce

This was all helpful information it was important that the machinists would be able to repair the system if it were to be damaged so it could continue in operation. When creating concepts the major goals for the system were durability, user friendliness and a minimal part count

Durability (25%)

a. Designed with shock loading in mind
b. Can withstand a 6' foot drop
c. Minimal maintenance
d. Design for operator manufacture and assembly

Ease of Operation (21%)

a. Able to be moved by one person from site to site
b. Easy connect and disconnect capabilities
c. Ability to handle rough terrain
d. Lifting mechanism (crank driven scissor jack)
e. Moved by one person either by pulling or pushing
f. Engine accessibility (starting and adjustments)

Ease of Maintenance (20%)

a. Disassembles quickly
b. Made of standard parts available to the Senegalese

Safety (16%)

a. Brakes for going down a steep decline
b. Made of easily disposable materials

Cost (11%)

a. Cost less than $500 to manufacture

Engine Swap Capabilities (6%)

a. Designed to house another engine within 20% of current engine

As the project continued it was shown that there are no inclines so a breaking system was rendered unusable as well as the lifting mechanism.

DESIGN

The design process began with the sketching of potential outer shell geometries shown below in figure 14.

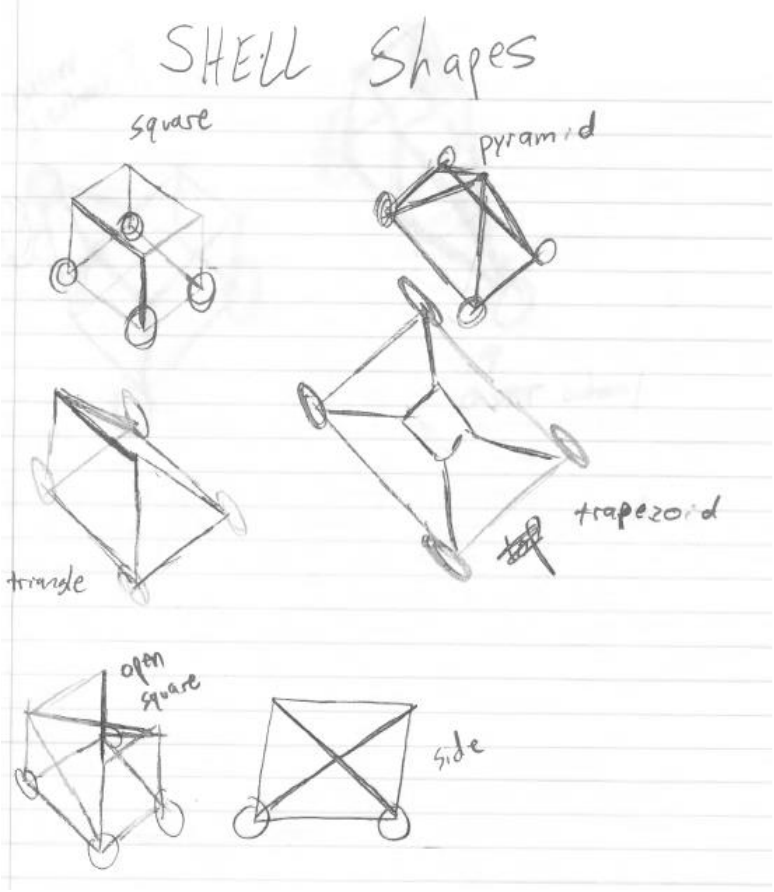


Figure 14- Initial shell geometries

These designs were discussed with Professor Salehpour, the square configuration was chosen to the ease of manufacture and also the engine accessibility. All of the other designs required precise angled cuts which may have been difficult for the Senegalese machinery.

Design Alternatives and Selection

Concept 1

After the square configuration was decided upon the next challenge was to determine the best way to configure the shafts to be incorporated into the exoskeleton. Below in figure 15 and 16 shows the first concept.

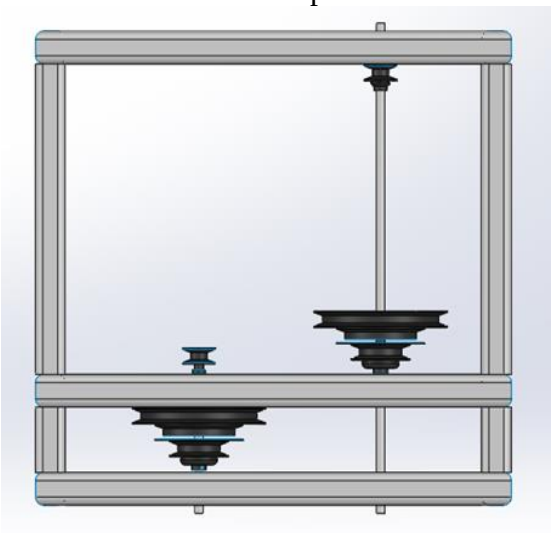


Figure 15- Top view

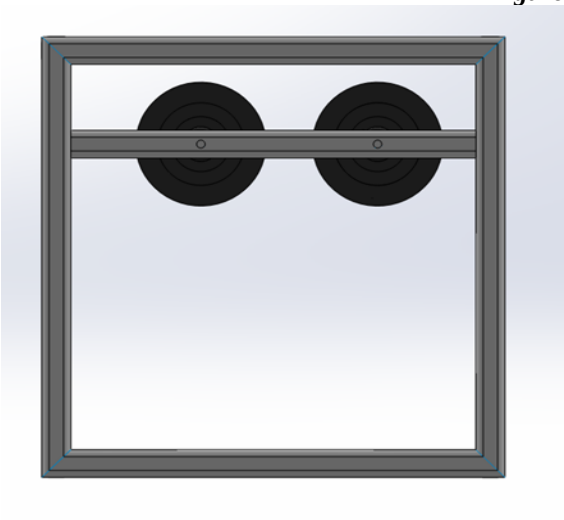


Figure 16- Side view

This design had many flaws the first that was a major concern was that this would have been very top heavy and likely to tip if bumped. Another problem was that the small shaft was susceptible to large shearing forces.

Concept 2

With the next concept the structural integrity was greatly improved. This next concept is shown below in figure 17.

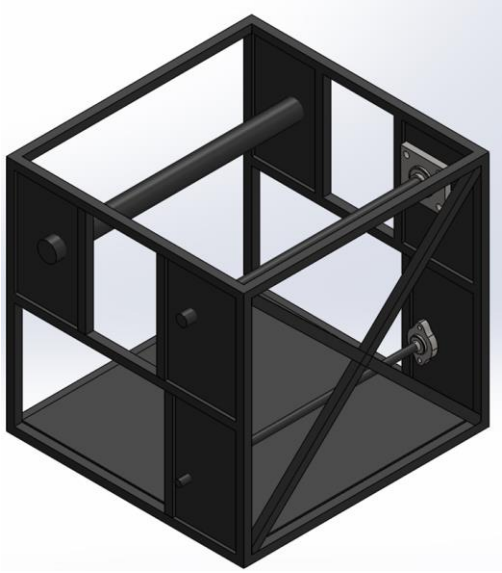


Figure 17-Concept 2

The major reason that this design was not chosen was because of the weight. This design contained 3 shafts, this drove up the price for materials and added additional unneeded weight.

Concept 3

This design cut down on the weight from the previous design as well as calling for only 2 shafts to transfer the power from high to low rpm while increasing the torque. Below is the second to last concept shown figure 18

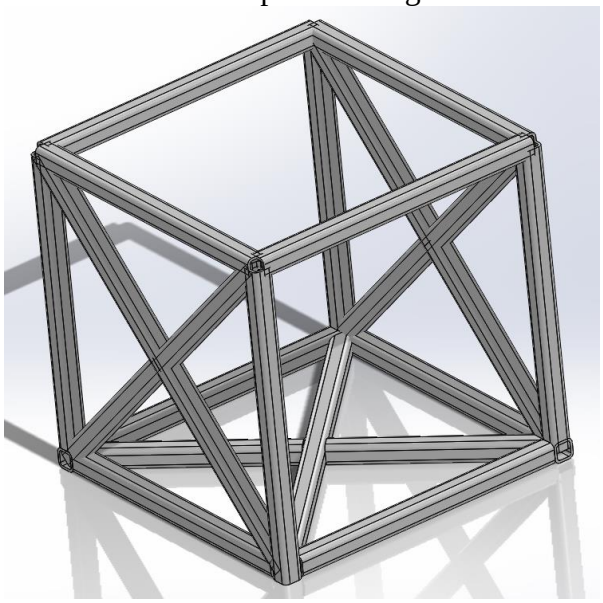


Figure 18-Concept 3

Concept 4 and Final selection

Below in figure 19 shows the final shell design.

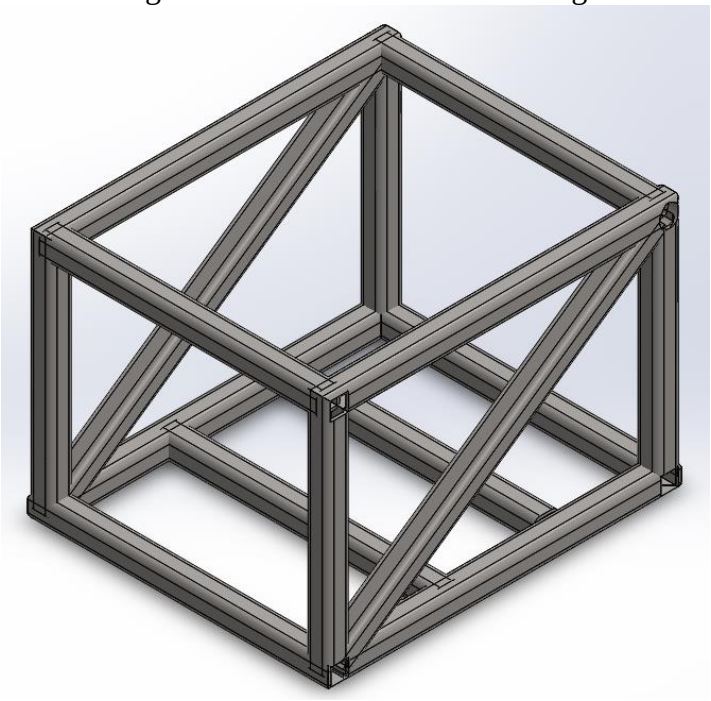


Figure 19- final design

This design further reduced the material used by removing two of the four support beams in addition to having the engine support traveling the width of the shell which was the shortest distance.

Loading Conditions

The first loading conditions considered were the engine support beams. The engine has a weight of 60 lbs when it is full of gas as well as oil. Below in figure 20 shows force diagram for one of the engine support beams.

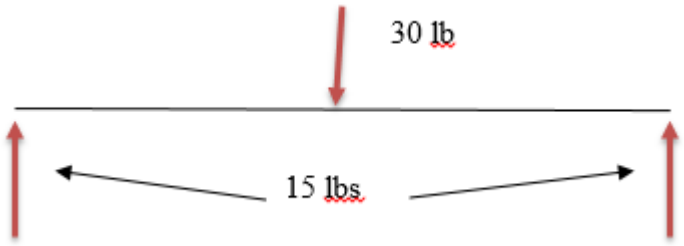


Figure 20- Force on engine supports

Force Analysis

Initially there was difficulty finding the proper way to determine the force applied on a body caused by 6ft. fall without making unreasonable assumptions. For example if using the equation below

$$F_{average} = m\,\frac{\Delta v}{\Delta t}$$

To find the force one would have to assume the time of impact. This was too great of an assumption to accurately determine the applied force on the system. Initially I used the equation to find potential energy which is $PE = mgh$ where *m* is mass, *g* is gravity (32.2 ft/s^2) and *h* is the height of the fall this was determined to be incorrect. Instead research on g forces had to be done. Research done by Matthew Huang (11) while researching the mechanics of vehicle crashes was able to determine the effects of the acceleration of an unharnessed body. With this study he crashed a vehicle at a solid wall at 8, 14 and 21 mph during all of these test the dummy had an accelerometer on its center of mass and was unrestrained. Since the only force acting on the body was gravity from the time the vehicle stopped and the time the dummy stopped it could be treated as a falling object. The next step was to find the velocity of an object dropped from 6 ft. below are the equations to determine this.

$$v = \sqrt{2gh}$$
$$v = \sqrt{2 * 32.2 * 6ft}$$
$$v = 19.6571 ft/s$$
$$v = 20\frac{ft}{s}$$
$$v = 13.64\ mph = 14\ mph$$

13.64 mph= 20.00 ft/s

From his research Huang determined that a dummy traveling at 14 mph will still be traveling with an acceleration of -3g's w.r.t the stopped vehicle **[10]**. With this information a better system analysis was able to be attained. After the acceleration was attained the force was found. To find the force the total weight was needed on the next page in table 6 will show the breakdown of the weight with a final total.

Table 6- All weights including total

Part	weight
Exoskeleton (solidworks)	56 lbs
Tires (4)	19 lbs
Power transmission system	74 lbs
Engine (fluid included)	60 lbs
Total	**206 lbs**

After the total weight was found the final force could be calculated. Below is the equations for force analysis.

$$F = ma$$
$$F = m3a\ (With\ Huang's\ research)$$

$$F = \frac{206}{32.2} * 3 * 32.2$$

$$F = 618\ lb$$

After this a safety factor of 5 was applied due to the fact that this assembly will be under some shock loading from engine start. With this applied safety factor it brings the final applied force to 3090 lbf.

Design Analysis

During the final design analysis a safety factor of **5** was used since this system would see impact loading. This safety factor brought the total applied force to 3090 lbf. The FEA analysis from solid works is shown below in figure 21

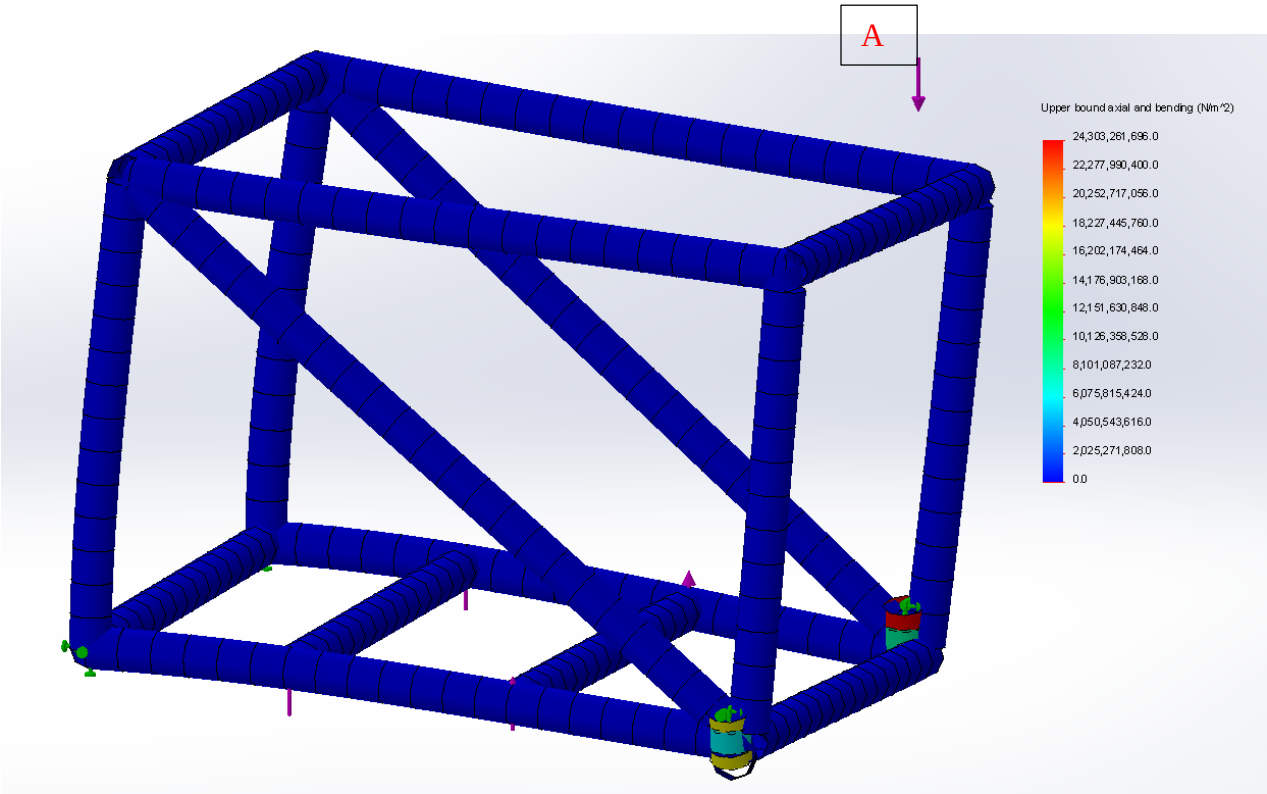
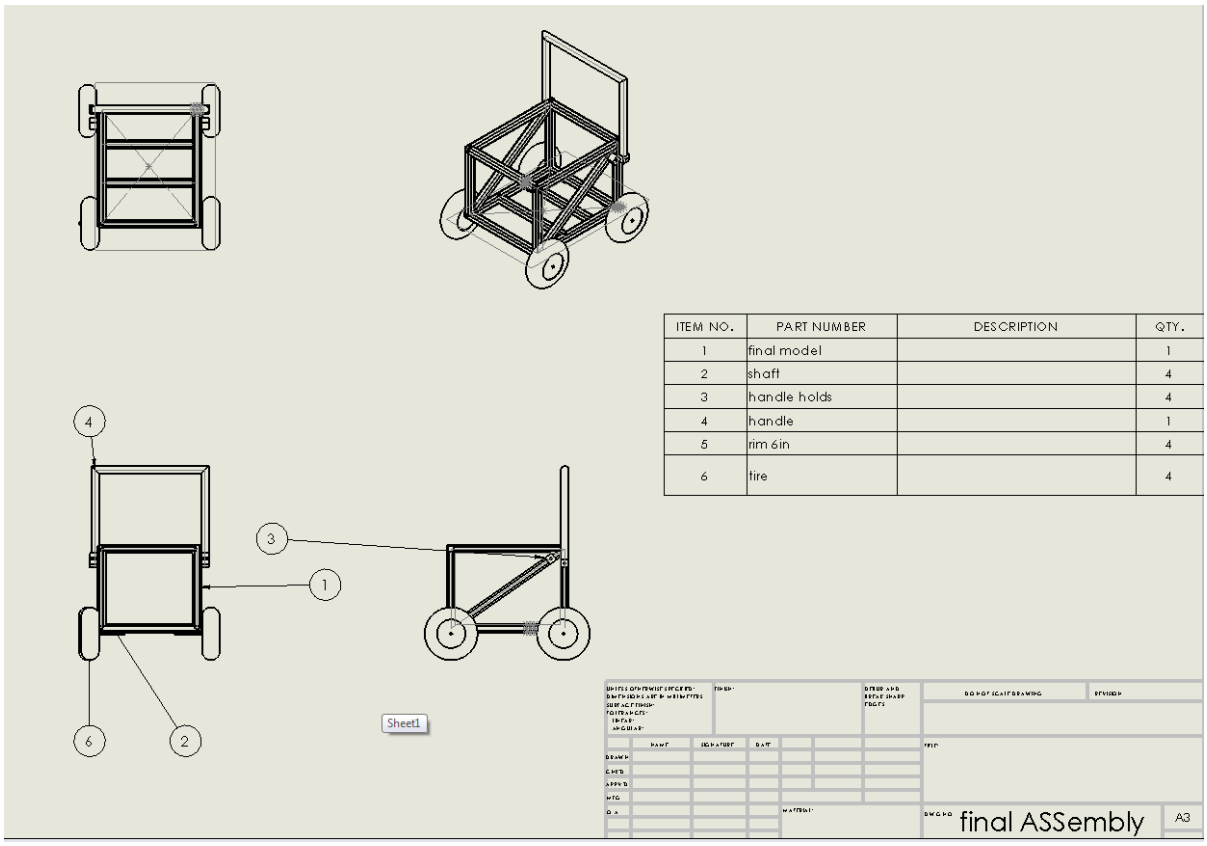


Figure 21-FEA Analysis

The worst case scenario would be for the system to fall on the top corner where the wheels were unable to take any of the force, this point is documented as A in figure 21. The arrows from the bottom represent the weight of the engine as it falls. From the above figure it is shown that the weld in the bottom corner will fail but the engine will be protected which is the primary concern.

Bill of Materials



ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	final model		1
2	shaft		4
3	handle holds		4
4	handle		1
5	rim 6in		4
6	tire		4

FABRICATION AND ASSEMBLY

To begin the fabrication process the minimum clearances were determined using wood block and tubing the pictures below show the process.

Figure 22- top

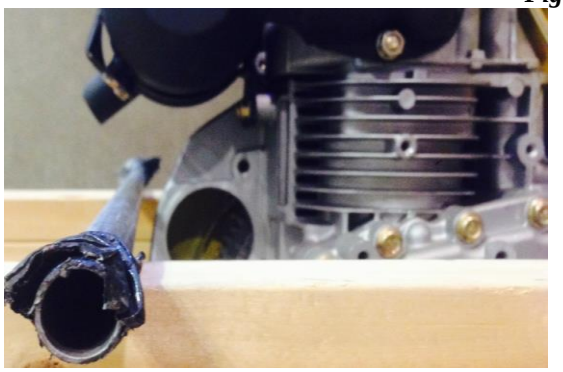
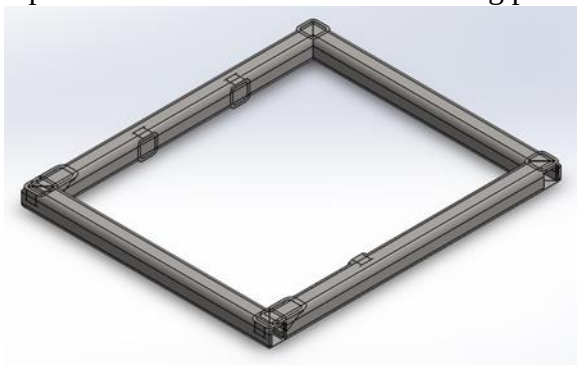
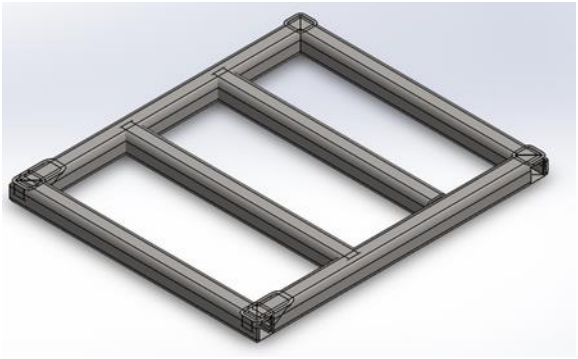


Figure 23- side

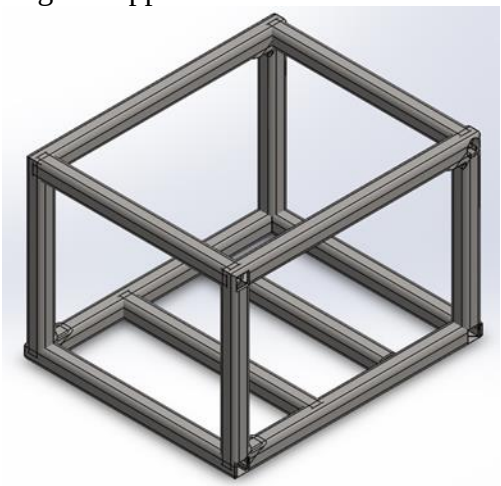
After the minimum clearances were determined the square tubing was cut to length and the top and bottom welded. The following pictures will take you thru the fabrication process.



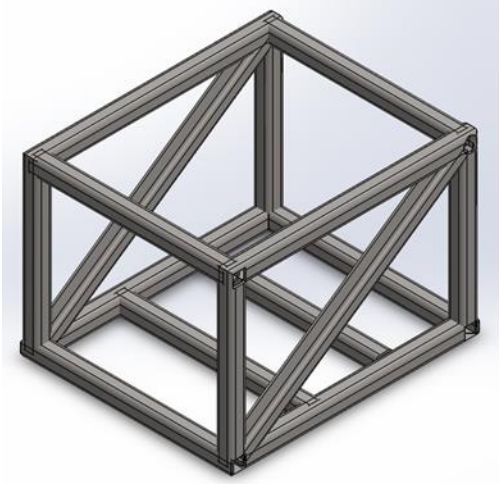
Bottom and top welded into rectangles



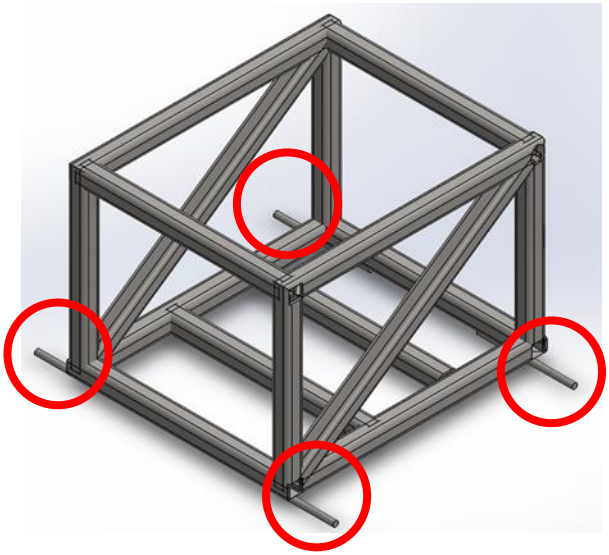
Engine support members welded into the bottom



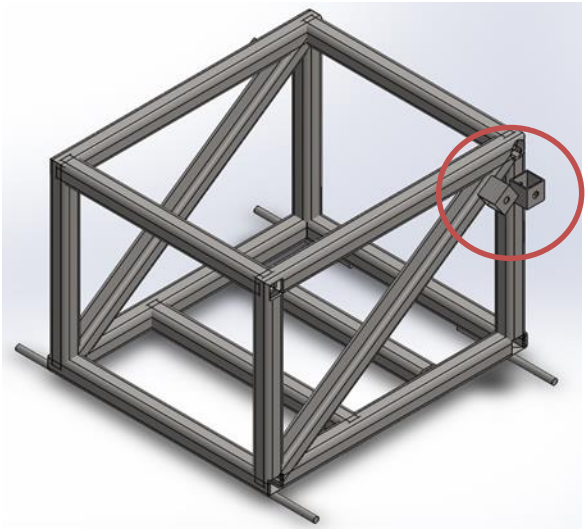
Vertical support welded into place securing the top and the bottom



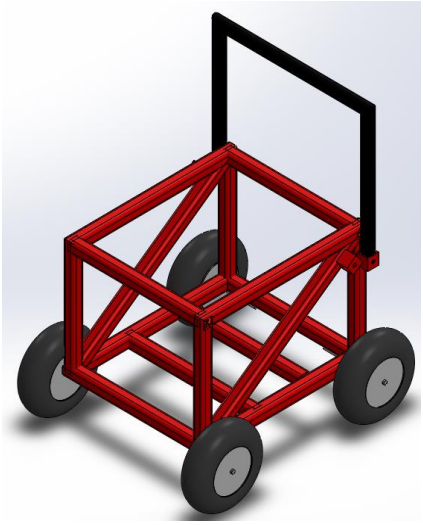
Cross members cut and welded in after bearing mounting holes were drilled



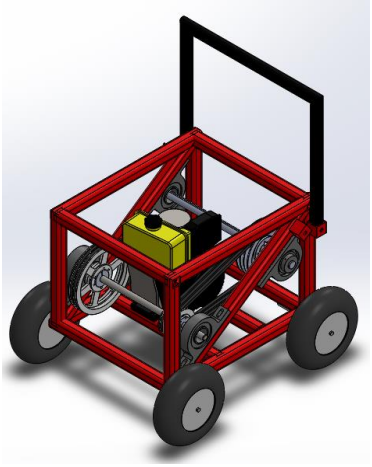
Axels for the wheels welded to the bottom of the assembly



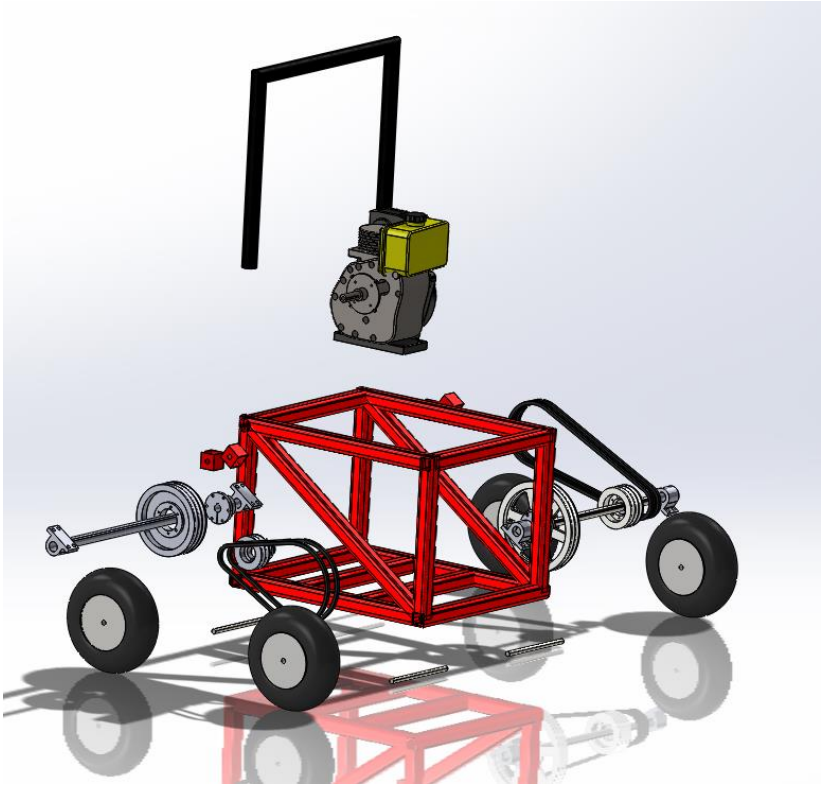
Handle holders welded on after holes drilled



Wheels attached with cotter pins and handle installed with bolts



Engine mounted, sheaves, bearings, shafts, and belts installed.



Exploded view of the model

TESTING AND PROOF OF DESIGN

To test the assembly it was taken to the sand volleyball court on the University of Cincinnati's campus. This was determined to be the best testing scenario since the terrain in Senegal is primarily sand. The assemly was then manuvered around the court to test the overall mobility of the system in a similar terrain. Below is the testing process taken place.

Figure 24-Testing

Figure 25- More Testing

From the above pictures it is seen that the tires dig into the sand instead of ride on top. This beacme a big concern if this is to be implemented in senegal, there needs to be changes.

Results

The initial testing went poorly, the system was extremely difficult to manuver in the sand. It was barely able to be moved by one individual. It was determined to be impractical for the customer needs. It was easier to lift the overall weight and carry the assembly compared to attempting to push the sinking tires thru the sand. Next the tire pressure was lowered in the hopes of increaasing the contact surface area with the sand to increase mobility howerer this was determind to be insufficient in the aid of mobility.

PROJECT MANAGEMENT
Budget, Proposed/Actual

Proposed

- **Materials**

	Price	Qty.
Total Cost		
Square Tubing		
8 Feet of 14 gauge 1-1/2" w/.083" wall	$24.96	2
$49.92		
Shipping	$13.59	1
$13.59		
Sub-Total		
$63.51		
Components		
Tires		
15"X 6" max capacity 365 Lbs.	$19.75	4
$79. 00		
Total	**$142.51**	

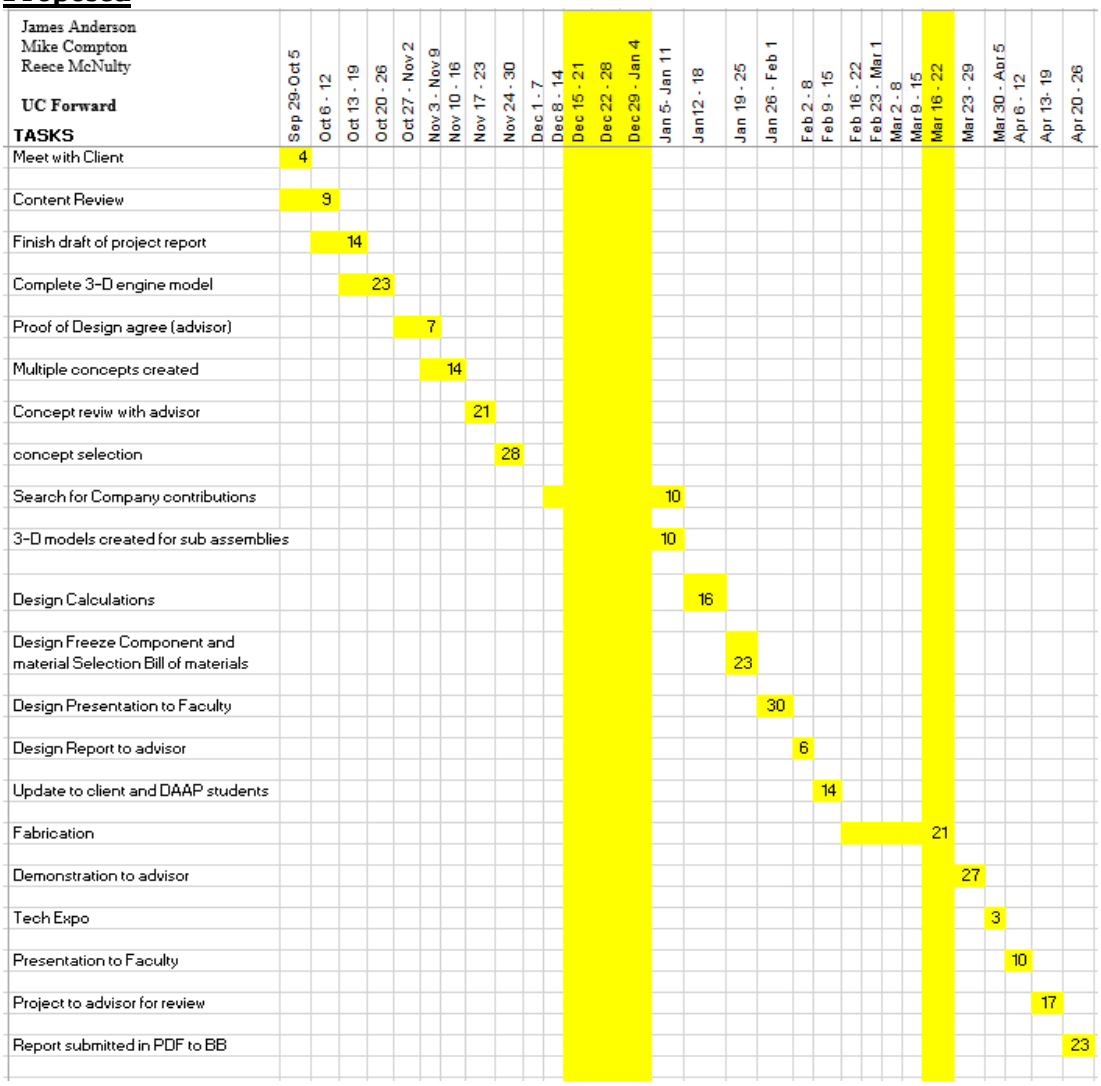
Actual

- 60 ft A- 500 tubing= $182.67
- Tires $13.95 X4 = $55.80
- Hardware = $5.62
- Total cost = $244.09

The price difference came from the underestimated amount of material needed.

Schedule, Proposed/Actual

Proposed



Actual

We deviated from the tech expo by not having the testing completed. Fabrication took longer due the limited lab time as well as only having Professor Conrad to help with the setup of machines.

CONCLUSION AND OPTIMIZATION IDEAS

This project began with very little input information from the end users, due partly because of the over 4,000 mile difference and in addition the project contact was difficult to reach. For the exoskeleton there was over 40 hours spent in the lab cutting the A500 steel to length then welding the members into place.

There was discovered many ways to optimize this system, beginning with the material selection. The 2 inch square tubing was selected due to the bearing selection. The bearings were 2 inches thick so they needed that much for a mounting surface. The bearings were selected from the required torque for the oil press and the sheave sizing.

To better optimize this assembly the exact amount of force to cruch the Jathropa seed needed to be known. There did not need to be 177 N*m of force on these seeds it was over designed which in turn drove up the sizes of the rest of the assembly. If the output force was lowered the pulley sizes and weights would be decreased and become smaller in overall size. With these small sizes a smaller cross section of material could be used, decreasing the weight tremendously. Instead of a 2" square cross section possibly a 1" square cross section could be used as long as the bearings fit. It was determined with FEA analysis that a 1" cross section would have taken the force but was unusable due to the bearing selection. If all these components were downsized the testing with the current tires would have yielded better results due to the fact that the overall weight would be greatly reduced. This scenario would also be better for the user since weight is a major concern due to the high levels of travel that this will see.

POWER TRANSMISSION

RESEARCH, TECHNOLOGY, AND EXISTING PRODUCTS

The use of a reduction system allows for the increase or decrease of output shaft speed, by changing the ratio of the pulley effective diameters. When an engine is equipped with a sheave and belt system, a multitude of output shaft speeds and torques are available for the user to power their application. For this design application, a reduction system utilizing sheaves and belts is desired to satisfy the end user in terms of price, ease of manufacturing/assembly, durability, and efficiency. Research data is listed in the *Appendices* section and includes interviews, reduction system examples, patented systems, customer surveys, and other helpful information that aided in the design.

Dan Bode, the project leader of UC Forward, was interviewed to get details on the current operating conditions of the Senegalese workers. He shared that the workers are currently using pulleys to reduce the engine speed, but are doing so without proper ratios and design elements, causing inefficient engine operation and undesirable engine wear (5). Designing a system that would allow the proper ratios to be implemented would help increase engine efficiency, reduce maintenance costs, and reduce unnecessary engine wear. Through further meetings with the *UC Forward Group,* more details that would affect the Senegalese were brought to attention such as operating conditions, safety issues, and current work area problems. (6)

Existing Technology

Following the preliminary interviews, research regarding reduction systems began. Although it was stated that a belt system is desired, other options for the reduction system exist including gear drives, chain drives and belt drives, all of which need to be evaluated before settling on a design. It was decided to contact *Gates Corporation,* one of the leaders in power transmission, to get a better idea of the options available. It was suggested to read *Designing Drives for a Competitive Edge* to spark some ideas and choose a preliminary path for design. (7) The literature provided pros and cons on each individual drive train system. The most helpful piece of data taken from my reading was that the operating conditions of the machinery was going to the most detrimental effect to the systems longevity. This led to the decision of continuing the design as a sheave and belt system, which is more resilient to dust, arid air content, high UV exposure, and dry environments.

I searched for existing technology and discovered that the most similar piece equipment that our design would represent was that of a power take off system, commonly seen in machine shops, in which a single motor powers a belt system which provides power to particular “take-off” areas where belts can be attached to power applications. These systems commonly use flat belts, where as our system is to use v-belts due to the high amount of torque to be transferred through the belts.

Other similar systems include reduction systems that can be found on lathes and other machine systems, in which localized reductions happen through changes in sheave and belt sizes, to change the shaft speed and alter the torque. These systems are very similar to what is to be designed, however, the main difference is that our engine is not immediately sized to our application, therefore requiring a v-belt and sheave system which is capable of handling the torque and belt tensions present in the system.

Customer Needs, Survey, QFD

The voice of the customer is one of the most critical tools in designing the complete system. Using surveys, product objectives were determined and analyzed, allowing rated engineering characteristics to be developed, helping shape the overall design to cater the customer's needs.

The results shown below reflect the results from the surveys, which can be seen in the *Appendix*. The second column of the surveys show the desired average rating of the end user for different features such as safety, durability, ease of operation, price, ease of manufacture, efficiency, multi-output speeds, size and other capabilities that the designs will include. The third column of the table (not available for *Table 4*) shows the current average rating for the applications being currently used. Using these results and other design criteria with the quality function deployment tool, adjusted weighted results for design criteria were created and then administered for each project segment to two leading members of AYWA International, as well as an end user in Malika, Senegal in the form of surveys.

Table 7- Survey feedback

Survey Feedback	
Desired System Design Features	**Weight**
Safety	2.66
Durability	5
Ease of Operation	5
Price	5
Ease of Manufacture	4.66
Efficiency	5
Multi-Output Speeds	4.66
Size	2.33
Current Operating Conditions	**Weight**
Safety	4.33
Durability	2.33
Ease of Operation	3
Price	1.33
Ease of Manufacture	1.66
Efficiency	4
Multi-Output Speeds	N/A
Size	2.66

Table 8- Customer Needs

Customer Needs
Design for multiple loading conditions (13%)
Three output speeds to maximize engine operation (12%)
High grade materials (11%)
Color coded operating system (8%)
Adjustable shaft position (7%)
Quick release for attachment coupler (7%)
Shields (6%)
Maintenance Schedule (6%)
Single bolt design (6%)
Built on single platform (6%)
Mounted to engine carriage (5%)
Design for user manufacturing/assembly (4%)
15:1 maximum speed reduction (3%)
Adjustable output shaft orientation (3%)
Bolt torque specifications (2%)

Taking the customer results from the quality function deployment (see below), all aspects of customer satisfaction were touched upon via adjusting the designer multiplier and providing an estimated customer satisfaction for the finished product. In order to get these desired results, the designer multiplier for "Ease of Operation" and "Efficiency" was changed from 1 to 1.1.

Table 9 - QFD

James Anderson **UC Forward Senegal** **Pulley Reduction System** 9 = Strong 3 = Moderate 1 = Weak	Shields	Bolt torque specifications	Design for multiple loading conditions	Maintenance schedule	High grade materials	Mounted to engine carriage	Desgined for user manufacturing/assembly	Single bolt design	Color coded operating system	Adjustable shaft position	Built on single platform	Three output speeds to maximize engine operation	15:1 maximum speed reduction	Quick release for attachment coupler	Adjustable output shaft orientation		Customer importance	Designer's Multiplier	Current Satisfaction	Planned Satisfaction	Improvement ratio	Modified Importance	Relative weight	Relative weight %
Safety	9	3	9	1	3	1			3		3		1	3	1		2.7	1	4.3	5	1.2	3.1	0.06	6%
Durability	3	3	9	9	9	3			3		3			3	1		5	1	2.7	4	1.5	7.4	0.13	13%
Ease of Operation				1		3		1	9	3	3	3		3	3		5	1.1	4.3	5	1.2	6.4	0.11	11%
Price	3		3		9	1		3		1	3	3	3	3	1		5	1	1.3	2	1.5	7.7	0.13	13%
Ease of Manufacture	3		3		3	3	9	9		3	3			3	1		4.7	1	1.7	3	1.8	8.3	0.15	15%
Efficiency		1	3	3	3				1	3		3	3	3			5	1.05	4	5	1.3	6.6	0.12	12%
Multi-Output Speeds			3						3	3		9					4.7	1	1	3	3.0	14.1	0.25	25%
Size			3							1	3	3	1		3		2.3	1	2.7	4	1.5	3.4	0.06	6%
Abs. importance	1.73	0.67	3.77	1.68	3.33	1.35	1.31	1.83	2.42	2.06	1.91	3.49	0.87	2.08	0.98	29.5						57.0	1.0	1.0
Rel. importance	0.06	0.02	0.13	0.06	0.11	0.05	0.04	0.06	0.08	0.07	0.06	0.12	0.03	0.07	0.03	1.0								

Product Objectives

Multiple output speeds (25%)

Multiple output speeds are required to operate different applications at their proper speed/torque values. The multiple speeds will aid in the longevity of the engine, efficiency, and safer operating environments.

a. Minimum of three output speeds

Ease of manufacturing (15%)

It is imperative that the system is capable of being produced by the workers, with the machinery available to the workers. Assembly is also at the forefront of the design.

a. Ability to manufacture parts on vertical mill, lathe, and simple grinding machinery

Durability (13%)

Having a durable system will allow the Senegalese people to increase their overall output by removing the downtime required to fix mal-designed fixtures. Durability will also decrease the price in the long run.

a. Replaceable shaft bearings
b. Mounted to engine carriage to keep off ground
c. Pulley material grade at/above recommended design level
d. Sealed bearings are *ok* for wet conditions
e. Maintenance schedule
f. Can withstand one-foot drop

Price (13%)

Price is to be kept as low as possible. This can be accomplished by substituting certain manufactured parts with purchased parts, leaning out the manufacturing process, and keeping the overall process simple.

a. Must be under $500

Efficiency (12%)

Increasing efficiency will aid in a higher output from the fuel produced, reducing the Senegalese dependence on crude oil, and helping them have a higher crop output. The pulley system will help the engine operate at the desired torque/speed settings for each application. Operating at the desired settings, will reduce fuel use and help maximize the application output.

a. 15:1 maximum output speed reduction
b. Cross with engine specifications for maximum torque values
c. Adjust for biofuel consumption

Ease of operation (11%)

The pulley system will use multiple belts and pulleys to achieve the desired speeds/torque values. Setting the system up properly is required to achieve these desired speeds, and this must be instructed in a clear and concise method to prevent confusion of the workers (some who are unskilled and illiterate). A system which incorporates symbols and colors would be ideal.

a. Design with color-coded system to help illiterate workers properly configure system

b. Switch belts in fewer than ten minutes

c. Operable on/off engine carriage

d. Quick release to adjust shaft position

Safety (6%)

Safety is of importance to the engineering designers, however, based on the feedback from the end-user, safety is of little importance.

a. Fully enclosed, removable case (pulley reduction)

b. Recommended torque specifications for all bolts

c. Safety factors for pulleys, shafts, and coupler

d. Design with shock loading involved

Size (6%)

Size is to correspond to engine exoskeleton size. Must be attached to engine exoskeleton.

a. Weight below 40lbs

b. Built within aluminum rectangular box frame

DESIGN

The design of the power transmission began with setting a hierarchy of failure modes, to determine which parts should fail before one another. It was decided that the order for the power transmission failure modes are as follows (preferred to fail (1) vs. no failure (5)): bolts and hardware, belts, bearings, sheaves, and shafts.

The shaft design was completed by analyzing the shaft operating conditions in four different loading situations. First the torque analysis was done to analyze what torsional forces would be present in the system purely from the engines input. The reductions in speed were based on the desired output speed of the mechanical oil press, and were adjusted by changing the sheave sizes.

The next step was to determine an approximate shaft size based on the torsional values collected from the engine specs and the material properties of a general steel (I chose to use a preliminary material of AISI 1020-1059 Hot-Rolled Steel). Using the formulas seen below in "design analysis", I was able to solve for suitable minimum shaft diameters (based on a solid cross section).

Design Alternatives and Selection

Some of the main goals of this project, as iterated through the interview with Dan Bode, are to make a working system with low overall cost that will be able to handle the extreme environment conditions, last for multiple years without major failure, and be designed for small-shop manufacturing. Choosing a belt system over a gear/chain system will aid in keeping the overall cost minimal, see *Table 10*.

Table 10- Price drivers for power transmission

Price drivers for system design
Short manufacturing time for sheave (versus gear)
Machinability of entire system on low-tech machinery
User training is minimal (currently use belt drives)
Low maintenance
Less wasted material (machining scrap)
Existing belts and pulleys can be reused
No lubrication required

The shafts must be supported by a frame to support the belt pull and torsional forces produced in the system. The first idea for the frame was to have the shafts be oriented similar to the style of the bike rollers as shown in *Figure 26*. Having the shafts oriented simply between two rails will allow us to reduce material, have a rigid frame, keep a low profile and operate on a flat surface fairly easily. After modeling began, it became evident that the forces produced by the system would cause this style of rail support to fail from torsion and opposing forces.

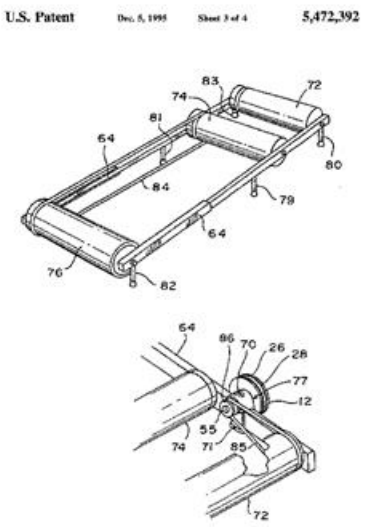


Figure 26- Bicycle rollers

Another housing option can be seen in *Figure 27*, which shows a lathe reduction system that utilizes a simply supported shaft. The high torque and operating conditions would prevent the simply beam supported style shaft from operating safely, so a fully supported bearing system must be used. Another feature seen in *Figure 27* is the stepped sheave, which allows for multiple speed ratios to be available from a single sheave, saving setup time and material costs.

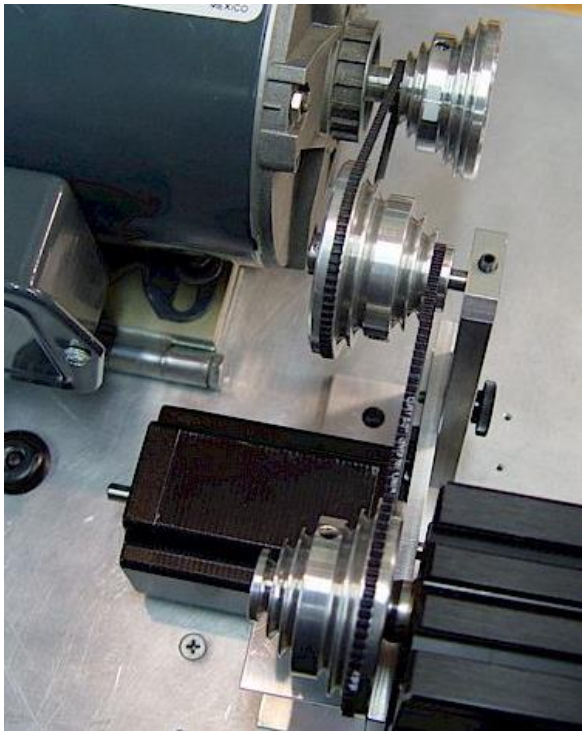


Figure 27-Lathe reduction

Figure 28 shows an industrial line-shaft system powering multiple applications from a single shaft. Line-shaft systems utilize a series of pulleys connected to a single, simply supported drive shaft, which then powers various applications. The bearing support style is simple, yet very effective, and allows for a mounting style which only requires two contact points by the bearings, saving space and material. *Figure 29* is a good example of a simply supported shaft powered by a direct-connect universal joint, which is the current method of powering the water pump.

Figure 28- Industrial power take off

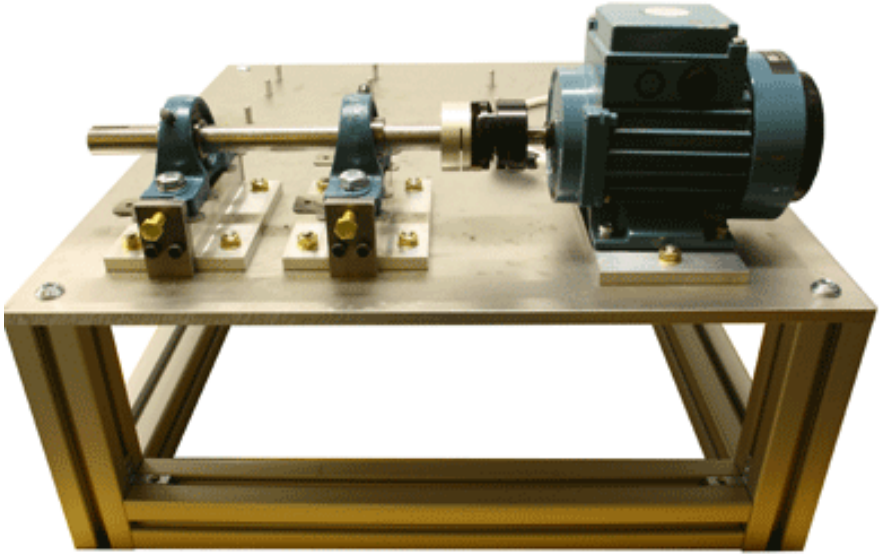


Figure 29- Simply supported shaft

It seems that the simply supported shaft will be the most ideal setup to use based on the ease of setup, maintenance, mounting face area, common part availability, and the presence of similar pre-existing systems. The sketches below represent the design alternatives that were not selected for the final design:

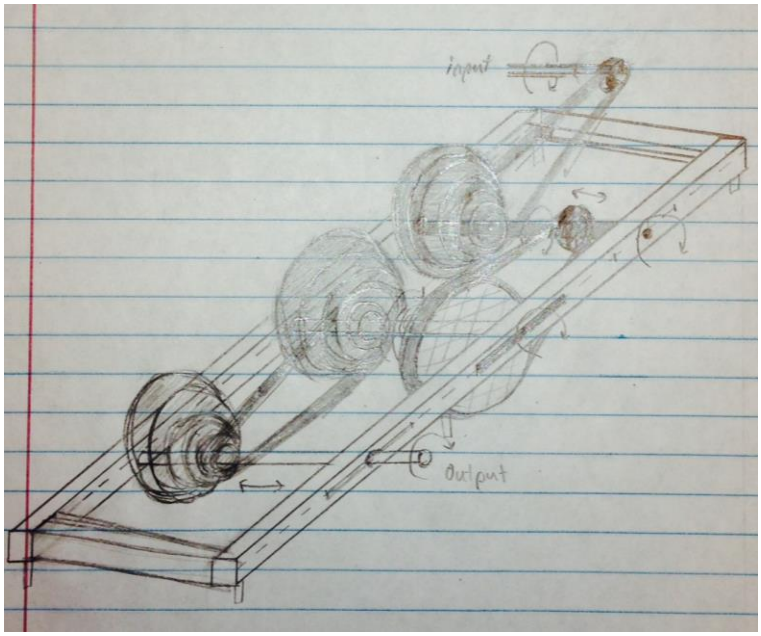


Figure 30 - Design alternative 1

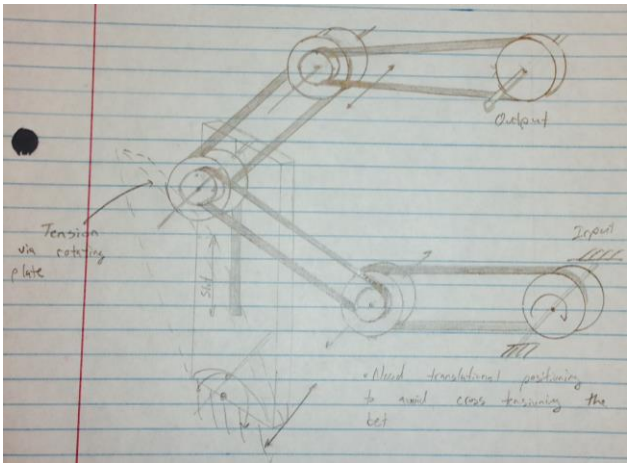


Figure 31 - Design alternative 2

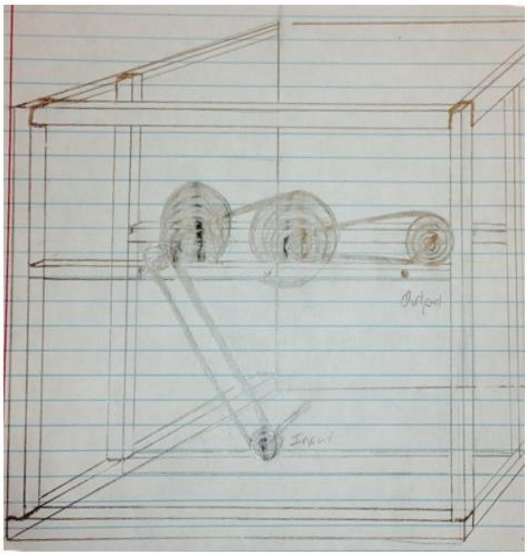


Figure 32 - Design alternative 3

Loading Conditions

Input 1 (T_o)	10.98	*N·m*
Input RPM	1500.00	*rpm*
HP at Input	2.31	*hp*
Power	1.73	*kilowatts*
Input 2 (T_1)	28.16	*N·m*
Input RPM	584.91	*rpm*
HP at Input	2.31	*hp*
Power	1.73	*Watts*
Input 3 (T_3)	65.20	*N·m*
Input RPM	252.65	*rpm*
HP at Input	2.31	*hp*
Power	1.73	*Watts*
Input 4 (T_4)	177.73	*N·m*
Input RPM	92.69	*rpm*
HP at Input	2.31	*hp*
Power	1.73	*Watts*

Sheave Diameters:		
Drive	78.74	*mm*
Driven 1	201.93	*mm*
Drive 2	92.71	*mm*
Driven 2	214.63	*mm*
Drive 3	78.74	*mm*
Driven 3	214.63	*mm*

Step Down One	584.91	rpm	61.22	rad/s
Torque	28.16	N·m		
Belt Pull	90-100	lbf		

Step Down Two	252.65	rpm	26.44	rad/s
Torque	65.20	N·m		
Belt Pull	130-140	lbf		

Step Down Three	92.69	rpm	9.70	rad/s
Torque	177.73	N·m		

Design Analysis

Taking the loading conditions, the shafts were designed for four different cases: nominal shear, simple bending, combined torsion and bending, and reversed bending with steady torsion. Analyzing the loading conditions for four different cases, would allow for the choice of the worst case scenario to be designed for, providing the optimal shaft diameter. All equations were taken from the *Design of Power-Transmitting Shafts* (8) and Gates V-belt Design Manual (9).

Case 1. it can be seen that the forces (τ design) are well below that of the materials acceptable shear stress, meaning that the material and chosen diameter are suitable for the time being.

τ_{nom-0}	τ_{nom-1}	τ_{nom-2}	
3.41	8.75	6.00	N/mm^2
$\tau_{design-0}$	$\tau_{design-1}$	$\tau_{design-2}$	
6.83	17.51	9.61	N/mm^2
$\tau_{acceptable}$	$\tau_{acceptable}$	$\tau_{acceptable}$	
588.54	588.54	588.54	N/mm^2

Input 1 (T_o)	10.98	*N·m*		τ_{design}	6.83	N/mm^2
Input RPM	1500.00	*rpm*		D_{ideal}	25.40	*mm*
HP at Input	2.31	*hp*		K_t	2.00	
Power	1.73	*kilowatts*		-		
Input 2 (T_1)	28.16	*N·m*		τ_{design}	17.51	N/mm^2
Input RPM	584.91	*rpm*		D_{ideal}	38.10	*mm*
HP at Input	2.31	*hp*		K_t	2.00	
Power	1.73	*Watts*		-		
Input 3 (T_3)	65.20	*N·m*		τ_{design}	9.61	N/mm^2
Input RPM	252.65	*rpm*		D_{ideal}	34.93	*mm*
HP at Input	2.31	*hp*		K_t	1.60	
Power	1.73	*Watts*		-		

$$T = 9550 * (\frac{InputPower(kw)}{rpm})$$

$$\tau_{nom} = 16T/\pi d_0^3$$

$$\tau_{design} = K_t 16T/\pi d_0^3$$

$$\tau_{acceptable} = 0.577 * \sigma_y$$

Case 2 takes into account the bending forces induced. The bending of the shaft in the x and y plane would be caused by the weight of the sheaves, belts, and hardware acting on the shaft, as well as the directional belt pull. The forces caused by belt pull were found using *Gates Corporation' Design Flex Pro,* a software program that takes the engine input, input style, operating conditions, sheave style/size, belt style/size, shaft diameter, bushing style/size, and operating specs to choose safe hardware options.

Other equations used to solve for the forces caused by bending are shown below. The individual forces were solved in both the x and y plane, to ensure that the max forces encountered were solved for in the correct directions. This was also helpful in choosing a bearing that was not oversized, by breaking down the total forces into two directions. The angles between the respective shafts, were solved using simple geometry theorems and algebraic strategies, and then applied to the overall belt pull forces to find the respective individual forces in both planes. Once the x and y plane forces were solved, the reaction forces at the bearing supports were found by solving for a zero moment about one of the reaction points and using the provided forces to solve for the unknowns.

Next, the max moment encountered by each shaft was found and the equation for the effective stress was applied to find the stresses encountered due to bending.

Input 1 (T_o)	10.98	*N·m*		τ_{design}	6.83	N/mm^2				
Input RPM	1500.00	*rpm*		D_{ideal}	25.40	*mm*				
HP at Input	2.31	*hp*		K_t	2.00					
Power	1.73	*kilowatts*		-						
Input 2 (T_1)	28.16	*N·m*		τ_{design}	17.51	N/mm^2		M_{max}	89.9959	*N·m*
Input RPM	584.91	*rpm*		D_{ideal}	38.10	*mm*				
HP at Input	2.31	*hp*		K_t	2.00					
Power	1.73	*Watts*		-						
Input 3 (T_3)	65.20	*N·m*		τ_{design}	9.61	N/mm^3		M_{max}	1084.918	*N·m*
Input RPM	252.65	*rpm*		D_{ideal}	34.93	*mm*				
HP at Input	2.31	*hp*		K_t	1.60					
Power	1.73	*Watts*		-						

$$\sigma_{nom-1} = (32M)/(\pi d_0^3)$$

$$\sigma_{design-1} = ((32M)K_t)/(\pi d_0^3)$$

$$\Sigma M_{R1} = 0$$

$$\Sigma F_y = 0$$

$$\Sigma F_x = 0$$

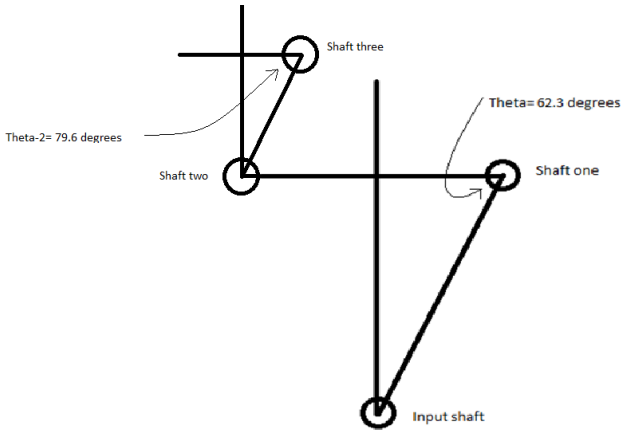


Table 11-Shaft one design

Forces from input to shaft 1

F_x	323.88	*N*
F_y	-209.99	*N*

Forces from shaft one to shaft two

F_x	600.00	*N*
F_y	0.00	*N*

$F1 + F2 = Belt\ Pull$

X reactions

$\Sigma M_{R1-x} = 0$

R_2	-464.62

$\Sigma F_X = 0$

R_1	-459.26

Y reactions

$\Sigma M_{R1-y} = 0$

R_2	-29.21

$\Sigma F_Y = 0$

R_1	239.20

Load Forces in X-plane

R_1	-459.26	*N*
F_1	323.88	*N*
F_2	600.00	*N*
R_2	-464.62	*N*

Shear Forces in X-plane

R_1	-459.26	*N*
D_1	-135.38	*N*
D_2	464.62	*N*
D_3	0.00	*N*

Moments in X-plane

R_1	0.00	*N·m*
D_1	-53.54	*N·m*
D_2	-134.15	*N·m*
R_2	-75.51	*N·m*

Load Forces in Y-plane

R_1	239.20	*N*
F_1	-209.99	*N*
F_2	0.00	*N*
R_2	-29.21	*N*

Shear Forces in Y-plane

R_1	239.20	*N*
D_1	29.21	*N*
D_2	29.21	*N*
R_2	0.00	*N*

Moments in Y-plane

R_1	0.00	*N·m*
D_1	27.88	*N·m*
D_2	45.27	*N·m*
R_2	48.96	*N·m*

Max Moment

$M_t = \sqrt{[(R_{2-x})^2 + (R_{2-y})^2]}$

M_t	89.9959	*N·m*

Table 12-Shaft two design

Forces from Shaft One to Shaft Two

F_x	563.71	*N*
F_y	-205.49	*N*

Forces from Shaft Two to Shaft Three

F_x	577.85	*N*
F_y	935.99	*N*

$F1 + F2 = Belt\ Pull$

X reactions

$\Sigma M_{R1-x} = 0$

R_2	569.25

$\Sigma F_X = 0$

R_1	-1710.82

Y reactions

$\Sigma M_{R1-y} = 0$

R_2	-795.07

$\Sigma F_Y = 0$

R_1	-140.92

Load Forces in X-plane

R_1	-1710.82	*N*
F_1	563.71	*N*
F_2	577.85	*N*
R_2	569.25	*N*

Shear Forces in X-plane

R_1	-1710.82	*N*
D_1	-1132.97	*N*
D_2	-569.25	*N*
D_3	0.00	*N*

Moments in X-plane

R_1	0.00	*N·m*
D_1	-199.44	*N·m*
D_2	-874.04	*N·m*
R_2	-945.88	*N·m*

Load Forces in Y-plane

R_1	-140.92	*N*
F_1	935.99	*N*
F_2	-205.49	*N*
R_2	-795.07	*N*

Shear Forces in Y-plane

R_1	-140.92	*N*
D_1	795.07	*N*
D_2	589.58	*N*
R_2	-205.49	*N*

Moments in Y-plane

R_1	0.00	*N·m*
D_1	-16.43	*N·m*
D_2	456.98	*N·m*
R_2	531.38	*N·m*

Max Moment

$M_t = \sqrt{[(R_{2-x})^2 + (R_{2-y})^2]}$

M_t	1084.918	*N·m*

Case 3 takes the bending and torsional values and combines them within a single equation to give the effective stresses. This equation allows us to see the main forces that will cause the system to fail, and choose a suitable material that will have a higher yield value than $\sigma_{effective}$.

$$\sigma_{effective} = \frac{32M}{\pi d_0^3} * [M^2 + \frac{3}{4}T^2]^{1/2}$$

Input 1 (T_o)	10.98	*N·m*		$\mathcal{T}_{design}$	6.83	*N/mm²*				
Input RPM	1500.00	*rpm*		D_{ideal}	25.40	*mm*				
HP at Input	2.31	*hp*		K_t	2.00					
Power	1.73	*kilowatts*		-						
Input 2 (T_1)	28.16	*N·m*		$\mathcal{T}_{design}$	17.51	*N/mm²*		M_{max}	89.9959	*N·m*
Input RPM	584.91	*rpm*		D_{ideal}	38.10	*mm*				
HP at Input	2.31	*hp*		K_t	2.00					
Power	1.73	*Watts*		-						
Input 3 (T_3)	65.20	*N·m*		$\mathcal{T}_{design}$	9.61	*N/mm³*		M_{max}	1084.918	*N·m*
Input RPM	252.65	*rpm*		D_{ideal}	34.93	*mm*				
HP at Input	2.31	*hp*		K_t	1.60					
Power	1.73	*Watts*		-						

Shaft One Analysis

$\sigma_{effective-1}$	2.24	*M·Pa*

Shaft Two Analysis

$\sigma_{effective-1}$	343.75	*M·Pa*

Material Selection	**SAE 1144 - As Forged**	
σ_u	1034	*M·pa*
σ_y	1020	*M·pa*
σ_f'	1586	*M·pa*
b	0.09	*M·pa*
σ_f	454	*M·pa*

Case 4 solves for reversed bending with steady torsion using the following equation:

$$d_0^3 = \frac{32(Fs)}{\pi} * (\left(\frac{M}{\sigma_{eff}}\right) + \left(\frac{3}{4}\right) * \left(\frac{T_m}{\sigma_y}\right)^2)^{1/2}$$

Shaft One Analysis

$d_{0\text{-}1}$	9.08	*mm*

Shaft Two Analysis

$d_{0\text{-}2}$	25.21	*mm*

Bill of Materials

Analyzing the results from the four different cases led me to choose the following components:

Table 13- Bill of materials

Bill of Materials
Shaft One
Shaft One – 25.4mm (SAE 1144) Fully Keyed
1” Pillow Block Mounted Bearing (2)
1” Split Collar Lock Ring (2)
Gates SDS 1. Bushing
Gates JA 1. Bushing
Gates SH 1. Bushing
Gates 3VX366 Belt (2)
Gates QD2/3V3.15 Sheave
Gates QD2/3V8.00 Sheave
Gates QD3A3.4/B3.8 Sheave
Shaft Two
Shaft Two – 31.8mm (SAE 1144) Fully Keyed
1.25” Pillow Block Mounted Bearing (2)
1.25” Split Collar Lock Ring (2)
Gates SK 1.25 Bushing
Gates AP52 (3)
Gates QD3A8.2/B8.6 Sheave
Other

¼” x ¼” Keystock
Hardware (Bolts, Nuts, Washers, Loctite)

FABRICATION AND ASSEMBLY

The section below will cover the fabrication and assembly methods undergone to make the power transmission. Additional information can be found in the appendices.

Fabrication and Assembly Process

1. Align engine
    A. Drill through holes (3/8” diameter)
    B. Mount engine with provided bolts and nuts
        i. Rotate bolt tension order to avoid uneven tensions

Figure 33-Engine alignment

2. Cut cross beams to length
3. Shim and c-clamp cross beams

Figure 34- Shaft alignment non-drive side

4. C-clamp shaft assembly to cross beams
    A. Confirm center to center distance of shafts on either side
    B. Confirm shaft clearance for engine and exoskeleton
    C. Mark bearing location for mounting through holes

Figure 35- Shaft alignment drive side

5. Drill through holes on cross beams
    A. 1/2” diameter through hole
6. Weld cross beams in place (MIG)
7. Install bearings and spacers for tensioning
    A. Micro alignment to be performed utilizing the bearing slots
    B. Fill pillow block bearings with grease
8. Determine bushing location on shaft and install keys
    A. Ensure clearance for belts and sheaves
9. Install sheave/bushing assembly
    A. Align sheaves using straight edges along both faces of sheaves
    B. Torque sheave/bushing assembly to specified values (manufacturer spec)

Figure 36- Sheave/bushing assembly orientation

10. Install belts and tension
    A. Tension belts to requirement based on deflection length of belt compared to top of sheave plane
    B. Belts can be tensioned by adding washers to appropriate bearing bolts or by backing out the nuts to increase center to center distance.

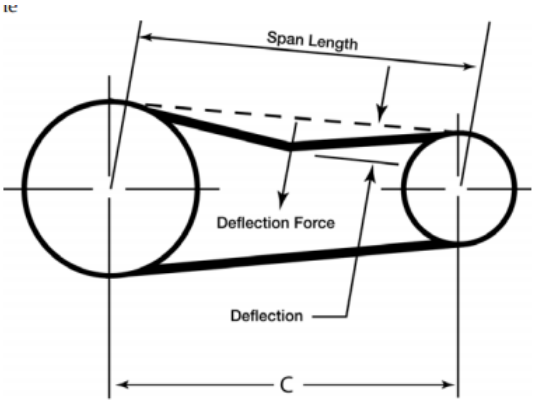


Figure 37- Belt tension check

TESTING AND PROOF OF DESIGN

Testing Methods and Results

Testing for the power transmission included running the system through variable input speeds and ensuring there was no slippage occurring in the belts when under load (via the mechanical oil press). The system was considered to be properly working if the belts held tension after running through the oil expeller test period, without gaining significant deflection distance.

The optimal testing method would be to run the engine on a dynamometer, mark the operating speeds on the throttle adjustment, then record the operating conditions off the final output shaft and comparing the data to the calculated output speeds and torque values. Unfortunately, our local dynamometer was unable to be connected to our output shaft due to the location of the output shaft and the lack of direct connection coupling to attach to the dynamometer, leaving us with a less accurate test method. However, for our application in Senegal, if the belts are sufficiently powering the press without slippage or alignment issues, it will be considered a success.

The initial deflection for belt one fell outside the max deflection distance of 0.14", so spacers were added and the tension was rechecked. When re-tensioned, the belts were found to have a deflection of 0.12" which was acceptable. Once in tension, the system was ran and checked for problems including: poor belt alignment, non-concentric rotation in sheaves, slipping of belts, difficult starting of engine, and poor engine operating characteristics. Figures 38 and 39 show the tension requirements for each belt as calculated using Gates DesignFlex Pro. The full document can be found in the appendices.

TENSION

	New Belt	Used Belt
Static Tension (Per rib/strand):	49 to 52 lbf	42 to 45 lbf
Static Belt Pull:	94 to 100 lbf	80 to 87 lbf
Rib/Strand Deflection Distance:	0.14 in	0.14 in
Rib/Strand Deflection Force:	3.3 to 3.5 lbf	2.8 to 3.1 lbf
Sonic Tension Meter:	216 to 231 N	185 to 200 N
Belt Frequency:	141 to 146 Hz	131 to 136 Hz
505C/507C Model STM Settings:	Mass 52.99g/m,Width: 1 mm/#R, Span: 226 mm	
Powerband Multiplier:	1.0044 to 1.0047	1.0038 to 1.0041

Figure 38-Tension data for belt one

TENSION	New Belt	Used Belt
Static Tension (Per rib/strand):	65 to 70 lbf	56 to 61 lbf
Static Belt Pull:	130 to 139 lbf	111 to 120 lbf
Rib/Strand Deflection Distance:	0.27 in	0.27 in
Rib/Strand Deflection Force:	5.4 to 5.7 lbf	4.8 to 5.1 lbf
Sonic Tension Meter:	291 to 312 N	250 to 270 N
Belt Frequency:	58 to 60 Hz	54 to 56 Hz
505C/507C Model STM Settings:	Mass 114.18g/m,Width: 1 mm/#R, Span: 437 mm	
Powerband Multiplier:	1.0031 to 1.0033	1.0027 to 1.0029

Figure 39-Tension data for belt two

PROJECT MANAGEMENT

Budget, Proposed/Actual

Proposed Budget

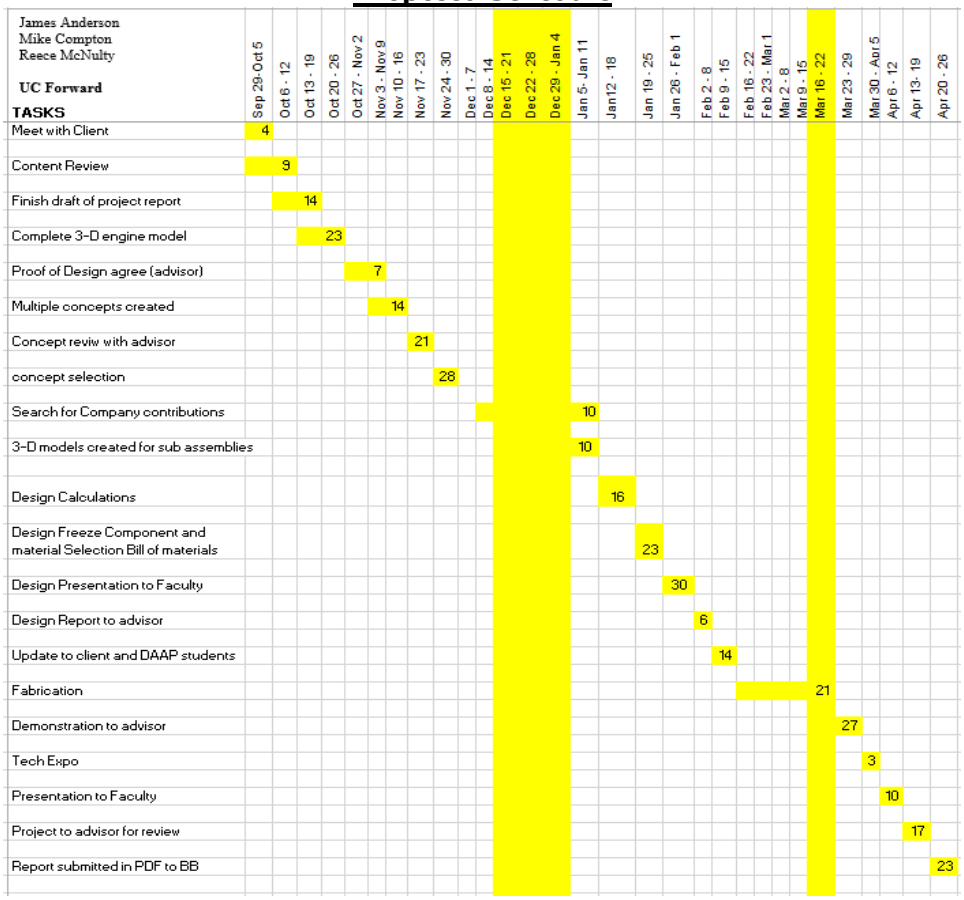
Structural Materials	$150
Guard Material	$50
Pulleys	$100
Bearings	$50
Assembly Parts	$25
Belts	$25
Labor	$50
Shipping	$50
Total	$500

Actual Budget

Belts, Sheaves, Bushings	$331.02
Bearings and Shafts	$89.98
Hardware and Labor	$50
Total	$471

Schedule, Proposed/Actual

Proposed Schedule



Actual Schedule

- Schedule was followed up to the fabrication task
  - Fabrication time was delayed because of :
    - Parts availability
    - Alternate class projects
    - Limited lab/tool access
  - Testing was delayed due to:
    - Individual completion of subprojects
    - Lab availability
    - Mutual free time of all group members

Plan to Finish

The project progress from plan to finish took many turns that effected the completion time of the project. The most difficult aspect was working with a group leader who was located thousands of miles from Cincinnati. This forced us to have limited contact, forcing us to make design changes every couple of weeks, when our project status was reviewed. Once we had a design freeze, we were able to begin manufacturing and testing, and the frustration from design changes was removed from the picture. The manufacturing and testing was more time consuming than expected, due to limited tool and lab access, forcing us to work around the lab schedule. The small problems that arose with the sub projects, proved to add up quickly and cause setbacks to the completion of the overall project. Working as a team, we were able to help each other out and deliver a finished prototype.

CONCLUSION

The power transmission design, fabrication, and assembly proved to be a challenging and fulfilling project. From a blank slate to and working prototype, our group went through many unforeseen challenges that forced us to adapt on the spot and go through rapid problem solving. The most challenging parts of this project were:

- Maintaining a schedule
- Finding mutual meeting times for all group members
- Working on a limited lab schedule for manufacturing
- Testing methods

Even though there were many hard times during this project, I still consider it to be a great success in that a working prototype was developed, which was fabricated based on a set design plan to meet the needs of the Senegalese people. The transmission has successfully stepped down the input rotational speed and increased the torque output, allowing the oil press to have the required shaft speed and excess torque needed to produce oil. The sheaves are capable of being quickly removed and switched if desired (thanks to a quick detachable bushing), and currently available v-belts can be used to cut price. Although the desired testing methods were unavailable, the working system shows that our design is at least reducing the input speed of the shaft and theoretically increasing the torque output.

Our project could be improved by:

- Advanced machine tools to reduce fabrication time and increase quality of machining
- Belt system attachment to run dynamometer
- 24/7 machine lab access
- Synchronized schedules to guarantee mutual free time
- Reduced design time requirements (1^{st} semester)

To improve the multidisciplinary aspect of the project, I would suggest having a combined lab, in which all the individual colleges are forced to meet and adhere to a schedule. This would have removed some of the communication problems and increased the group effort by allowing different parts of the project to be passed off.

APPENDICES

APPENDIX A - PULLEY RESEARCH AND DATA

Industrial Belt Design - Drive Detail Report

Design Flex® Pro by the Gates Corporation

Designed For:

Provided By: James Anderson
University of Cincinnati
2433 West Clifton Avenue
NULL
CINCINNATI, Ohio 45219
United States
ander2j2@mail.uc.edu
3309803640 Phone

Application: **Design #2**

INPUT

Drive Information

Speed Ratio:	2.34 Down
Input Load:	3.08 hp, Efficiency: 100.00 %
Service Factor:	1.6
Design Power:	4.93 hp
Center Distance:	18 in +/-6%

	DriveR	DriveN
RPM:	584.0	250.0 +/-4%
Shaft Diameter:	1.0 in	1.0 in
Bushings Checked:	QD, No MPB	
Belts Checked:	Super HC, Predator	

Single Belts

SELECTED DRIVE

Belt Type: **Predator - AP**

Speed Ratio: **2.32 Down**
dN RPM: **252.1**
Rated Load: **5.21 hp, ODR: 1.06**
Belt Pull: **290 lbf**
Center Distance: 17.37 in
Install/Take-Up Range: 16.62 in to 18.87 in

	Belt	DriveR	DriveN
Total # of Strands/Ribs:	3		
Part No:	3-AP52	QD3A3.4/B3.8	QD3A8.2/B8.6
Product No:	9185-0052	7831-3034	7831-3082
Top Width:	–	2.50 in	2.50 in
Weight:	1.2 lbf	4.0 lbf	22 lbf
Rim/Belt Speed:	558 ft/min	617 ft/min	581 ft/min
RPM:	125.6	584.0	252.1
Bushing Part No:	–	SH 1	SK 1
Bushing Product No:	–	7838-1100	7838-4100
Bore:	–	1.0 in	1.0 in
Pitch Diameter:	–	3.65 in	8.45 in

TENSION

	New Belt	Used Belt
Static Tension (Per rib/strand):	65 to 70 lbf	56 to 61 lbf
Static Belt Pull:	130 to 139 lbf	111 to 120 lbf
Rib/Strand Deflection Distance:	0.27 in	0.27 in
Rib/Strand Deflection Force:	5.4 to 5.7 lbf	4.8 to 5.1 lbf
Sonic Tension Meter:	291 to 312 N	250 to 270 N
Belt Frequency:	58 to 60 Hz	54 to 56 Hz
505C/507C Model STM Settings:	Mass 114.18g/m,Width: 1 mm/#R, Span: 437 mm	
Powerband Multiplier:	1.0031 to 1.0033	1.0027 to 1.0029

NOTES

- Matched Belts are required and must be special ordered. Call Customer Service for help.
- Installation allowance falls out of the specified centre distance range.
- User requested non-PowerBand belts.
- stock pulleys and bushings are designed to carry the high torque loads of this application.
- Pricing Scheme Used: Default

- Design Flex Drive Solutions assume Gates products and are not applicable to non-Gates products.
- products are not intended for use in any application where the failure of the product to perform could cause injury or death. This includes use on aircraft propeller or rotor drive systems or other in-flight systems necessary for safe flight.

Drive Price

Belt	$16.05
DriveR	
Pulley	$67.52
Bushing	$21.22
DriveN	
Pulley	$135.05
Bushing	$38.17
Total	**$310.11**

Industrial Belt Design - Drive Detail Report

Design Flex® Pro by the Gates Corporation

Designed For:

Provided By:
James Anderson
University of Cincinnati
2433 West Clifton Avenue
NULL
CINCINNATI, Ohio 45219
United States
ander2j2@mail.uc.edu
3309803640 Phone

Application: **Design #2**

INPUT

Drive Information

Speed Ratio:	2.57 Down
Input Load:	3.08 hp, Efficiency: 100.00 %
Service Factor:	1.6
Design Power:	4.93 hp
Center Distance:	9.75 in +/-8%

	DriveR	DriveN
RPM:	1500.0	584.0 +/-4%
Shaft Diameter:	1.0 in	1.0 in
Bushings Checked:	QD, No MPB	
Belts Checked:	Super HC, Predator	

Single Belts

SELECTED DRIVE

Belt Type: **Super HC - 3VX**

		Belt	DriveR	DriveN
	Total # of Strands/Ribs:	2		
Speed Ratio: **2.56 Down**	Part No:	2-3VX366	QD2/3V3.15	QD2/3V8.00
dN RPM: **584.9**	Product No:	9412-0366	7872-2031	7872-2080
Rated Load: **4.74 hp, ODR: 0.96**	Top Width:	–	1.09 in	1.09 in
Belt Pull: **140 lbf**	Weight:	0.25 lbf	1.2 lbf	10 lbf
Center Distance: 9.22 in	Rim/Belt Speed:	1217 ft/min	1237 ft/min	1225 ft/min
Install/Take-Up Range: **8.72 in to 10.22 in**	RPM:	399.1	1500.0	584.9
	Bushing Part No:	–	JA 1	SDS 1
	Bushing Product No:	–	7838-0100	7838-2100
	Bore:	–	1.0 in	1.0 in
	Pitch Diameter:	–	3.10 in	7.95 in

TENSION

	New Belt	Used Belt
Static Tension (Per rib/strand):	49 to 52 lbf	42 to 45 lbf
Static Belt Pull:	94 to 100 lbf	80 to 87 lbf
Rib/Strand Deflection Distance:	0.14 in	0.14 in
Rib/Strand Deflection Force:	3.3 to 3.5 lbf	2.8 to 3.1 lbf
Sonic Tension Meter:	216 to 231 N	185 to 200 N
Belt Frequency:	141 to 146 Hz	131 to 136 Hz
505C/507C Model STM Settings:	Mass 52.99g/m,Width: 1 mm/#R, Span: 226 mm	
Powerband Multiplier:	1.0044 to 1.0047	1.0038 to 1.0041

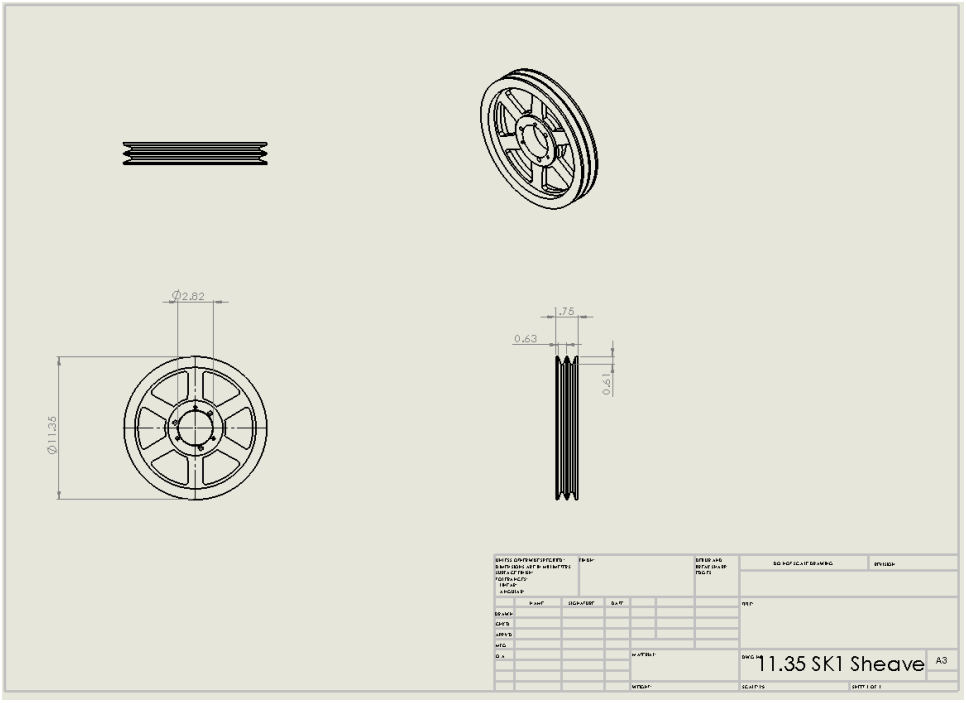
NOTES

- The drive power rating is slightly less than the requested design power. Review design requirements.
- Installation allowance falls out of the specified centre distance range.
- User requested non-PowerBand belts.
- Pricing Scheme Used: Default

- Design Flex Drive Solutions assume Gates products and are not applicable to non-Gates products.
- products are not intended for use in any application where the failure of the product to perform could cause injury or death. This includes use on aircraft propeller or rotor drive systems or other in-flight systems necessary for safe flight.

Drive Price

Belt	$15.37
DriveR	
Pulley	$47.83
Bushing	$15.52
DriveN	
Pulley	$101.28
Bushing	$24.64
Total	**$220.01**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0.61
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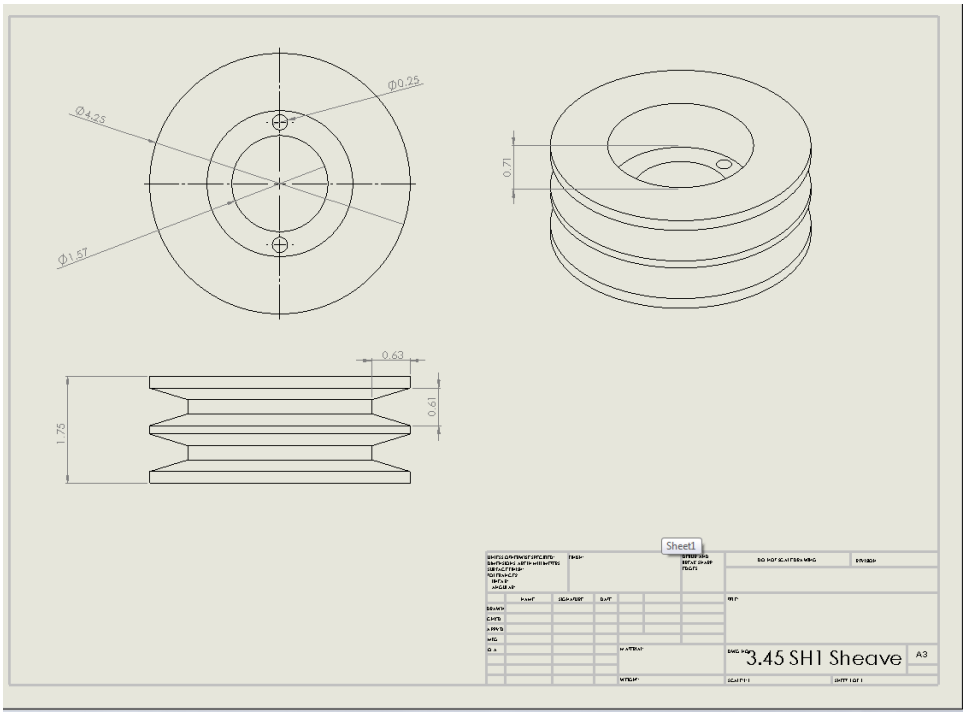
A3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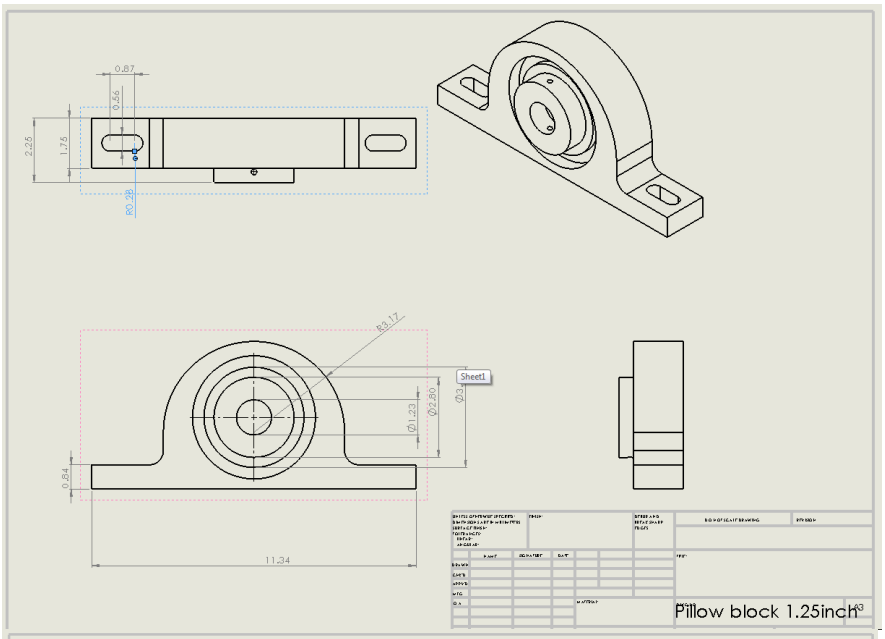
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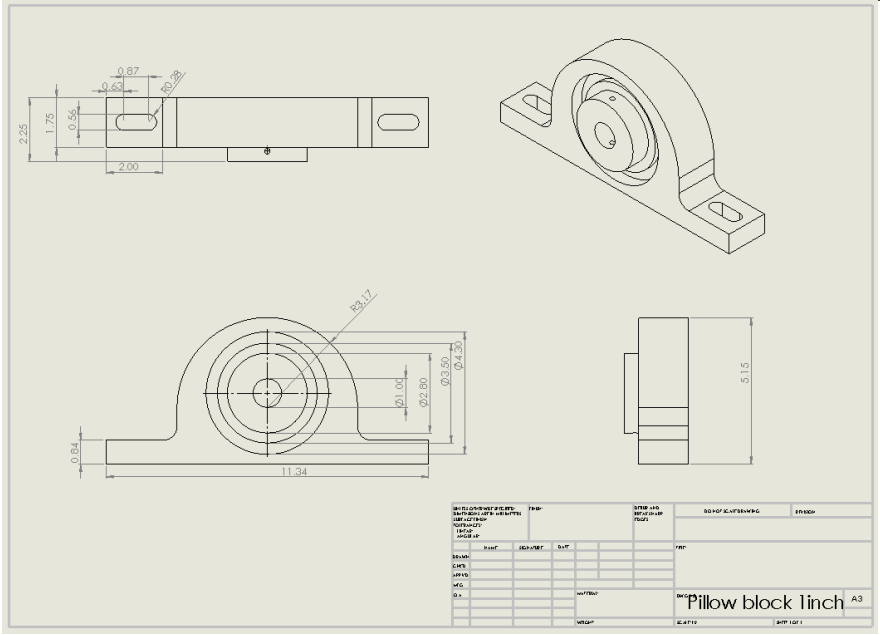
11.34
Pillow block 1.25inch
A3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2.00
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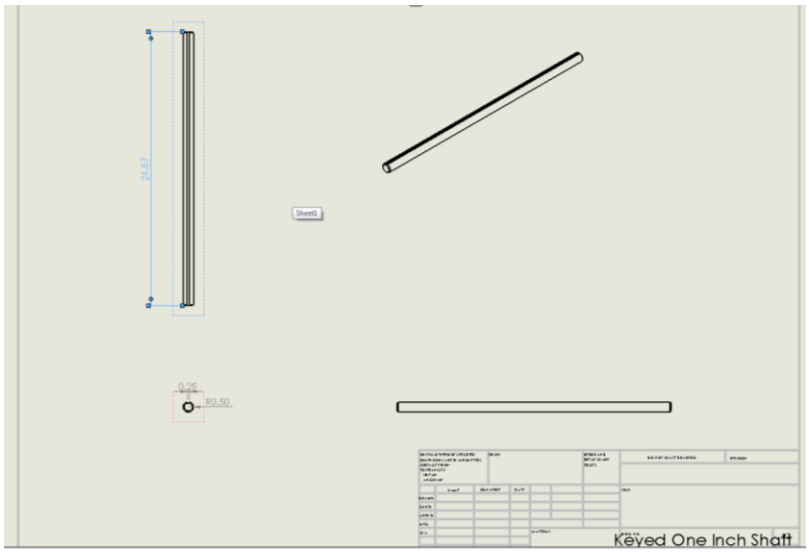
5.15
0.64
11.34
Pillow block 1inch
A3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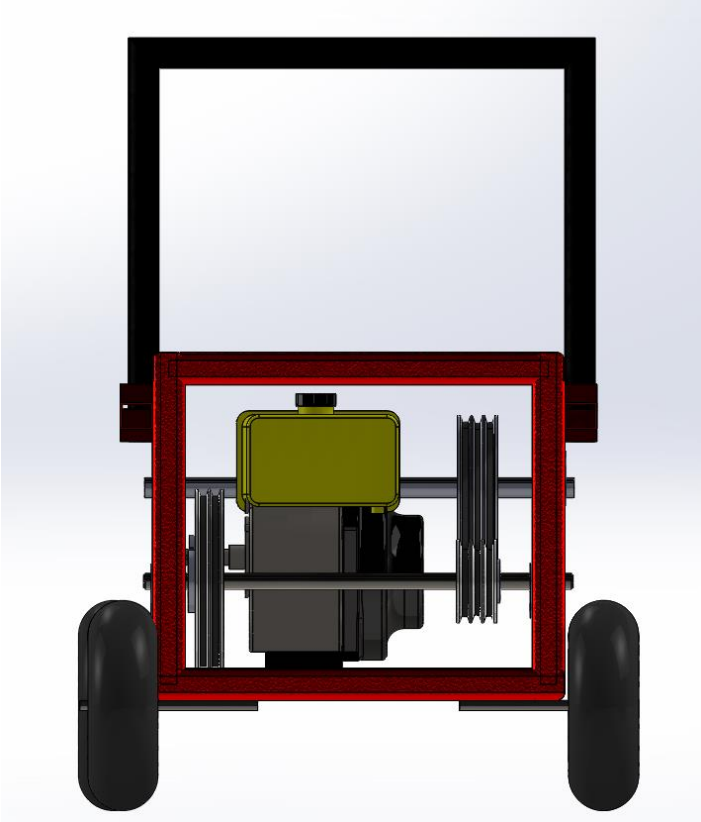
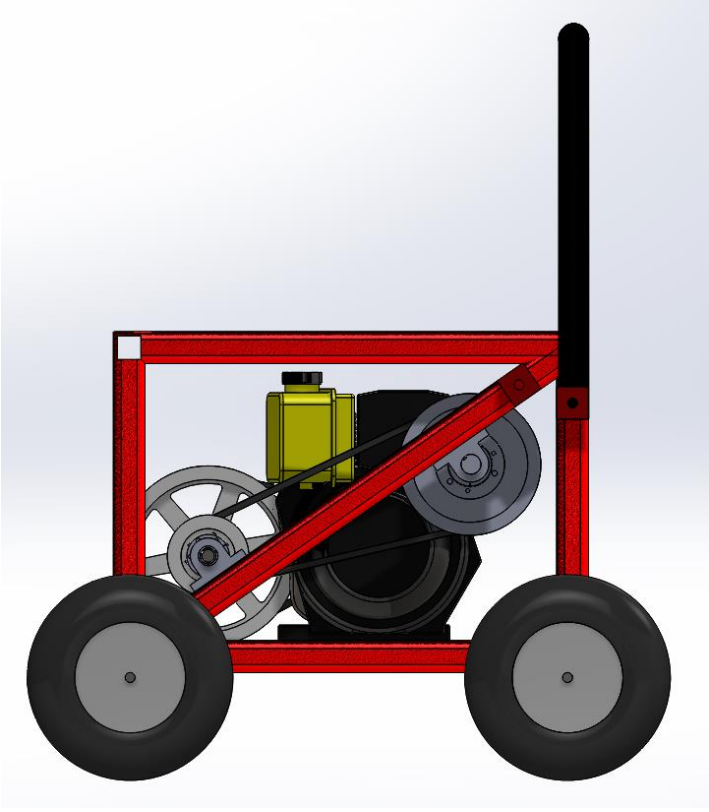
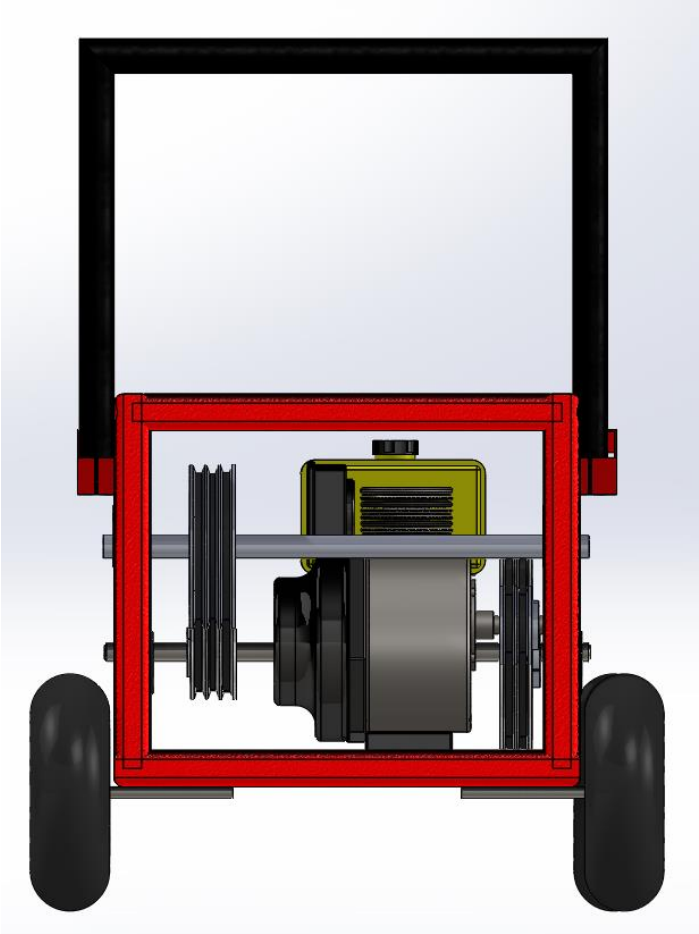
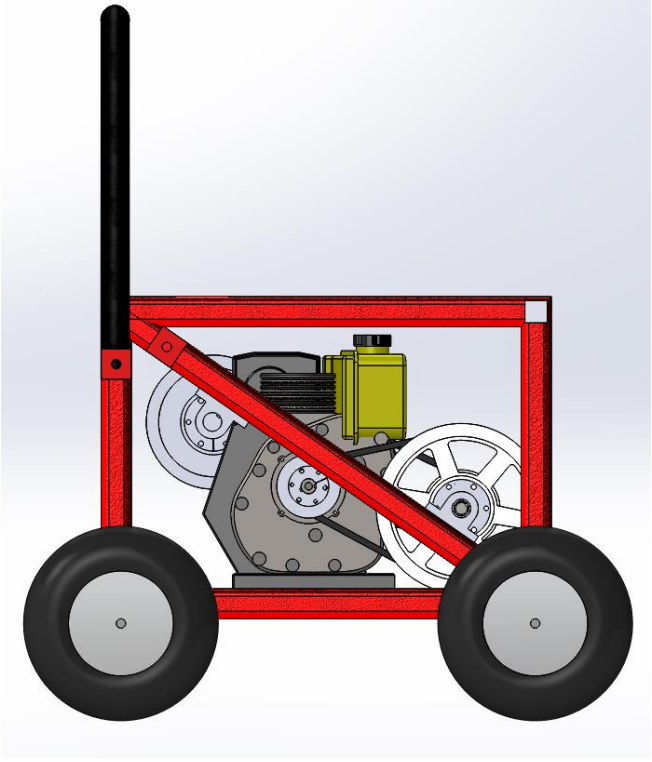
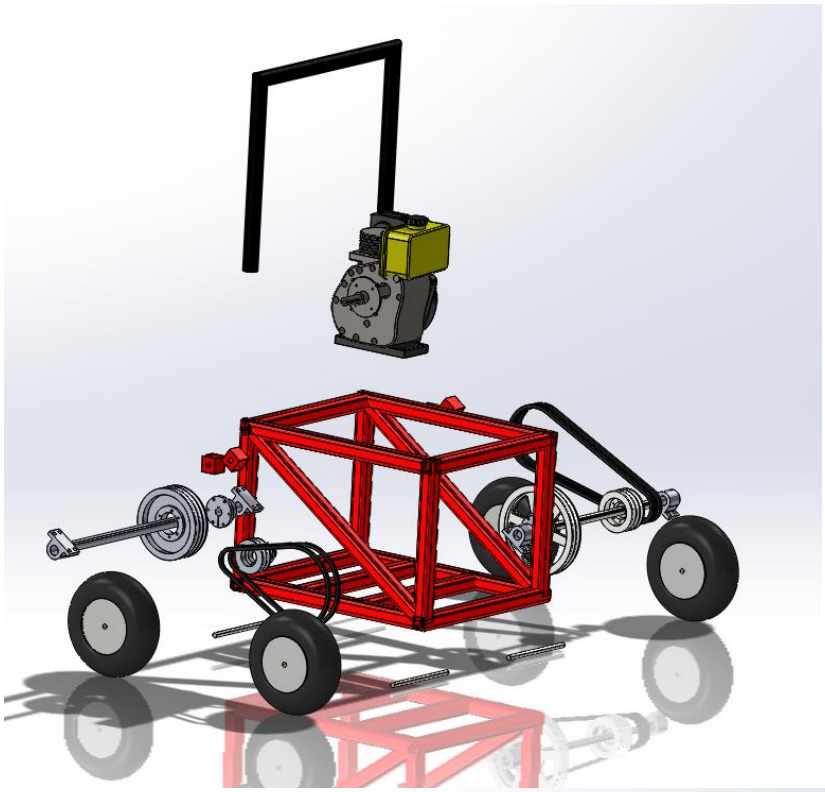
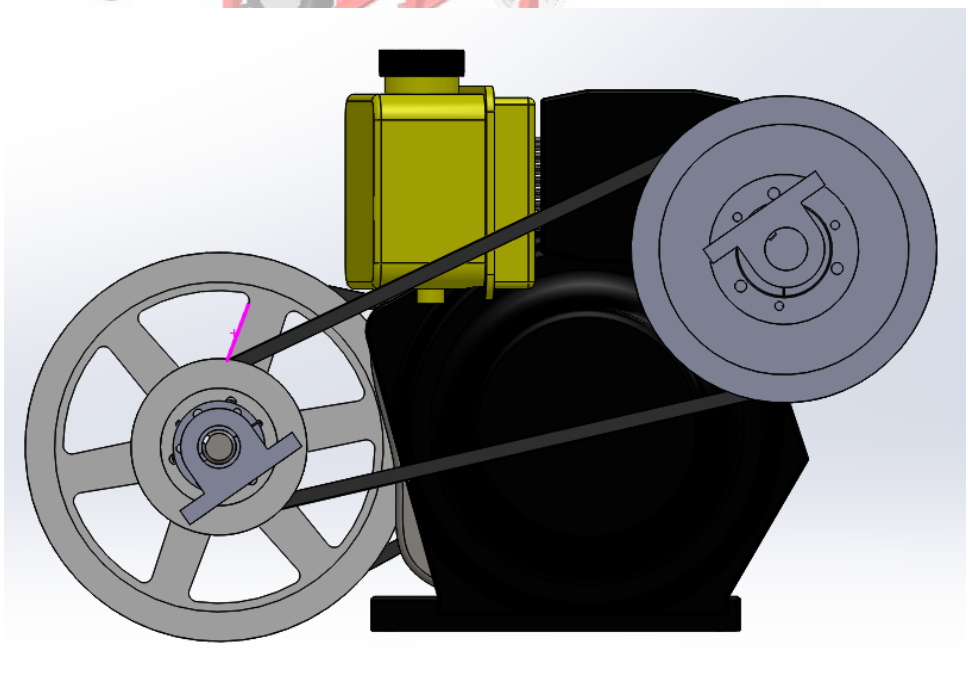


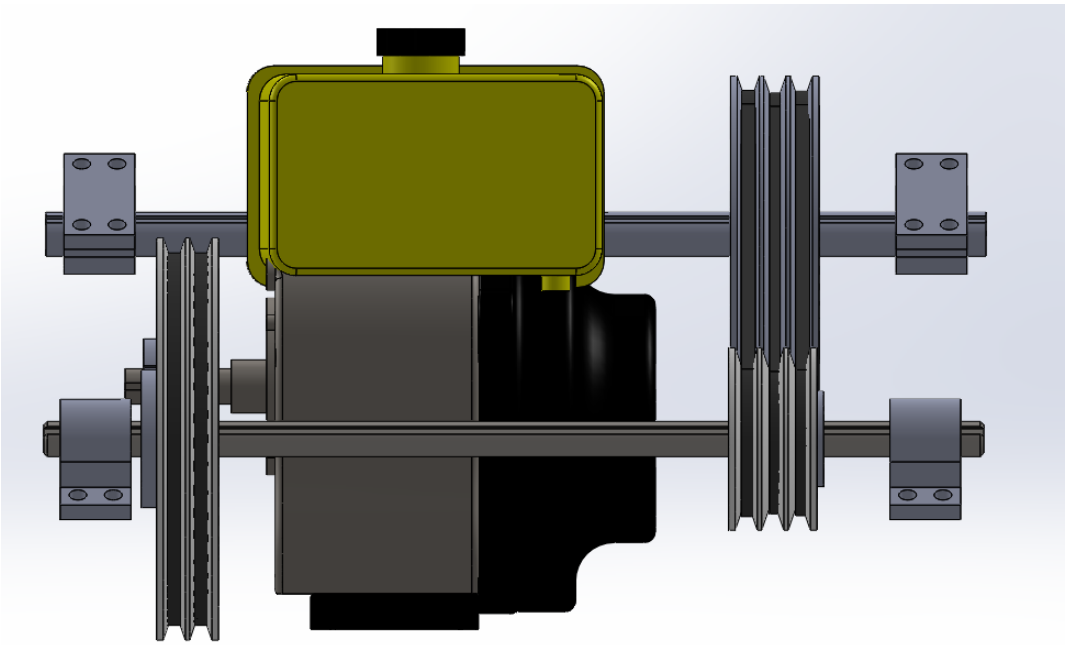


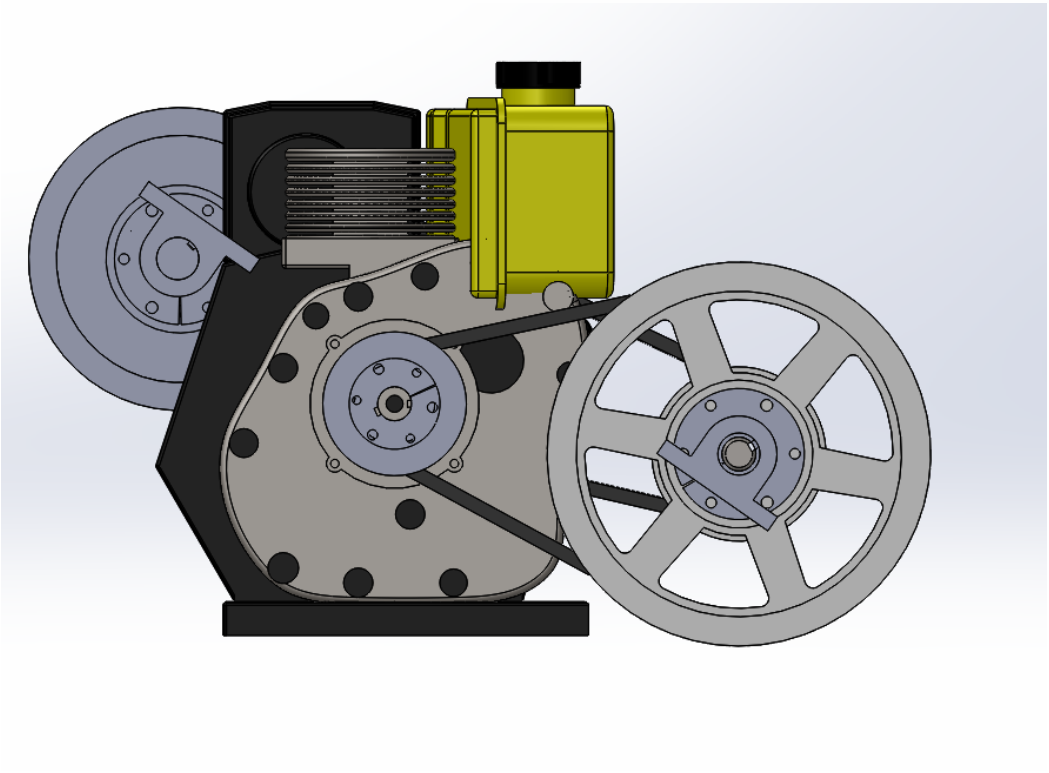


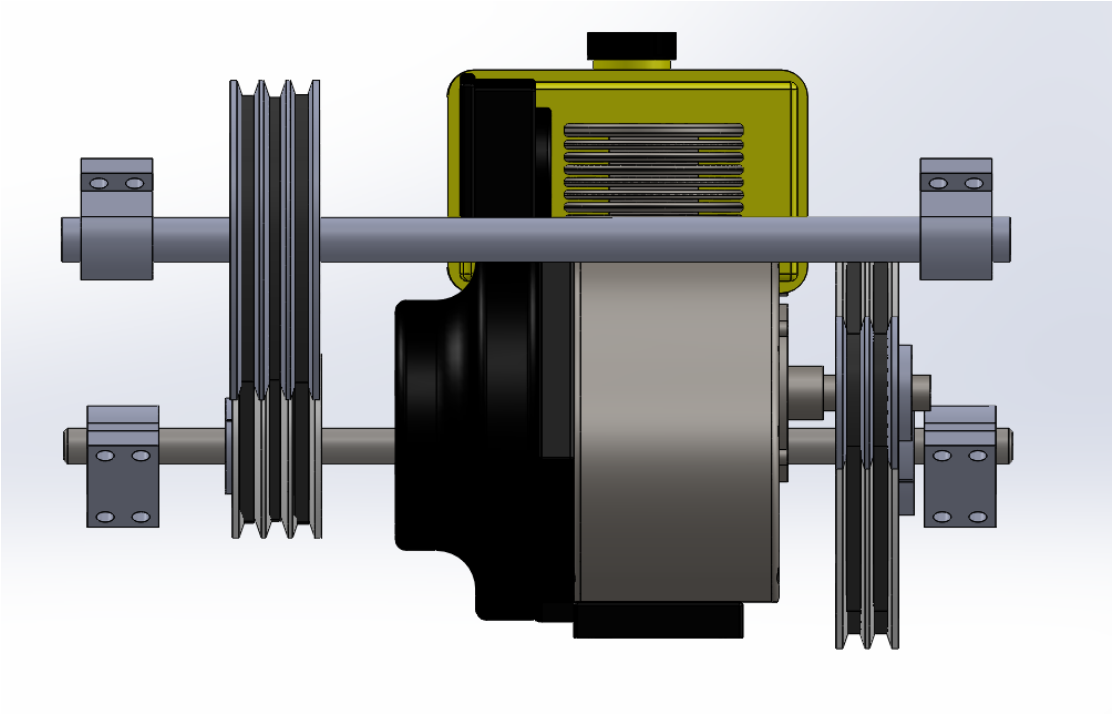


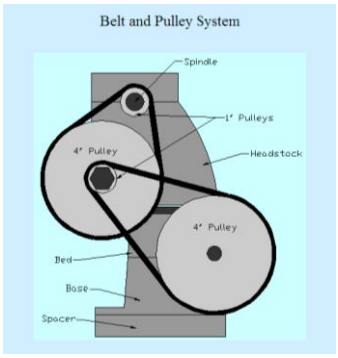


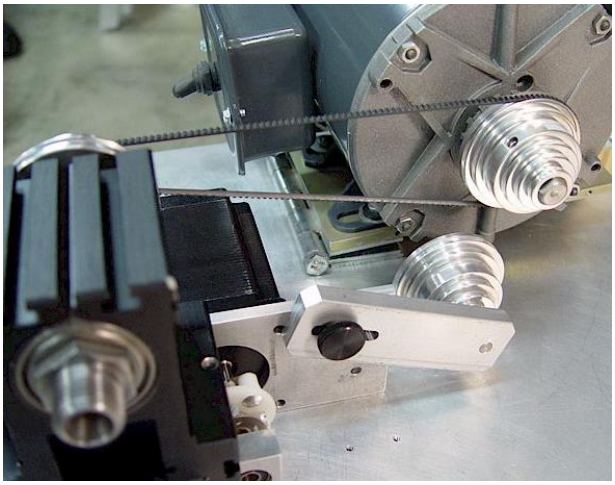
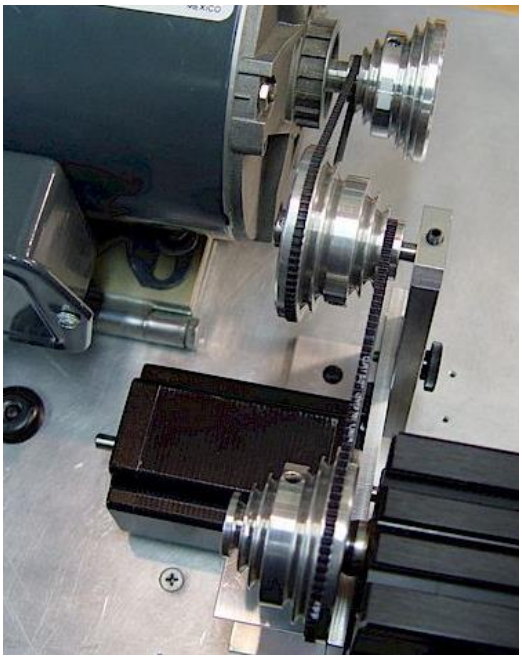


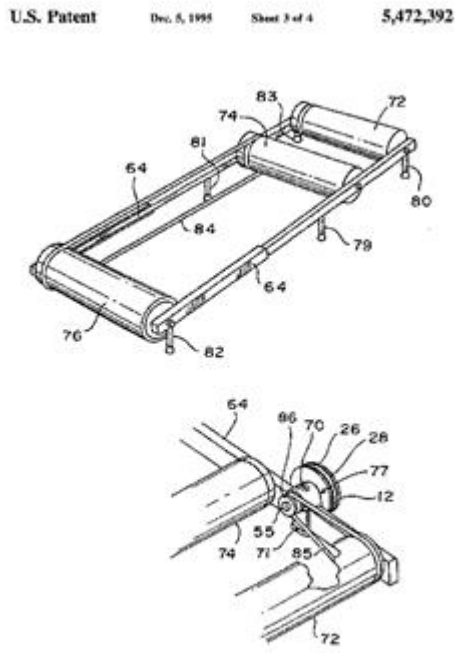






Lathe Pulley Reduction Example  ***Figure 40 Step pulley reduction on a lathe***  ***Figure 41 Offset pulley placement***	-High reduction with two belt system -Stepped pulleys increase options -Low tech -Post supports with bearings -Relatively compact -Price varies

Stepped Pulley Example  ***Figure 42 Adjustable stepped pulley orientation***  ***Figure 43 Three stepped pulley***	-Example of stepped pulley -Quick change capable -Set screws are easily accessed -Lateral tension adjustment for belt tensioning -Compact system -Allowable range for multiple pulley ratios -Needs single mounting fixture -Double post construction for increased stability -Drive pulley should be geared and not tension driven -Quick release system

Pulley Mounting Example  ***Figure 44 Bicycle roller showing shaft orientation***	-Cylinders mounted between aluminum standing brackets -Lightweight -Lateral adjustment for belt tensioning -Easily manufactured -Simply structured

APPENDIX B – ENGINE EXOSKELETON RESEARCH AND DRAWINGS

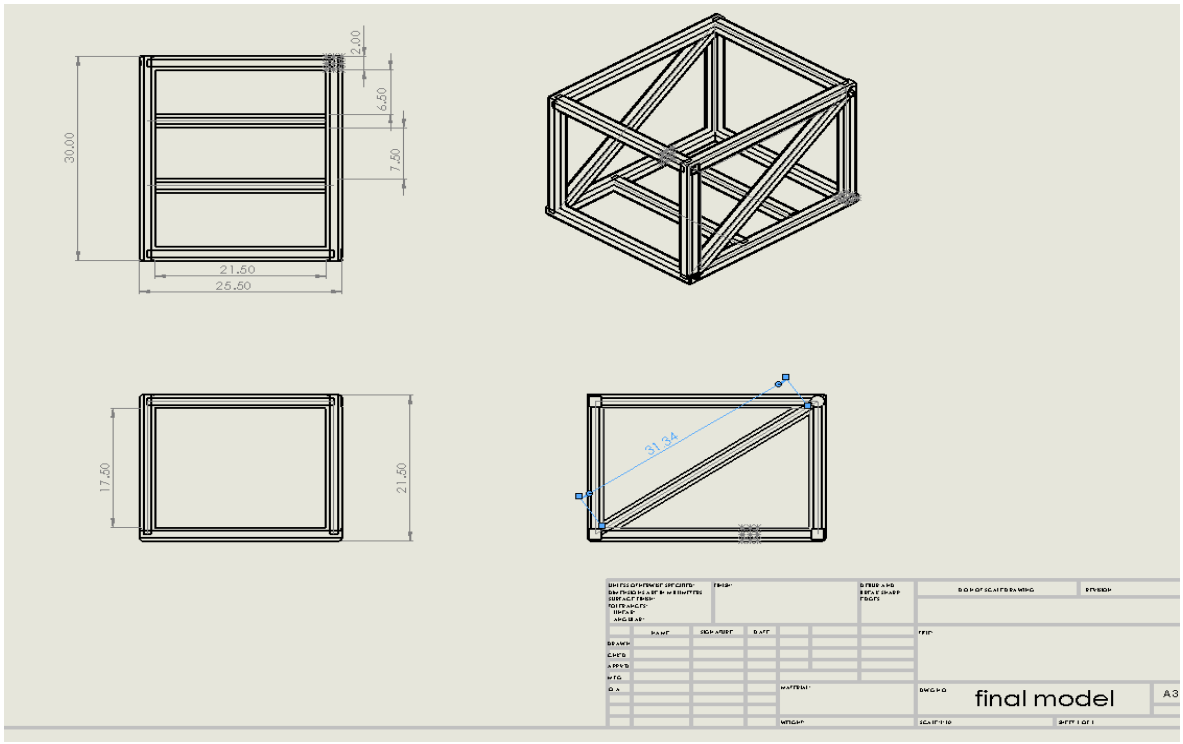
Figure 45 Wheeled engine housing

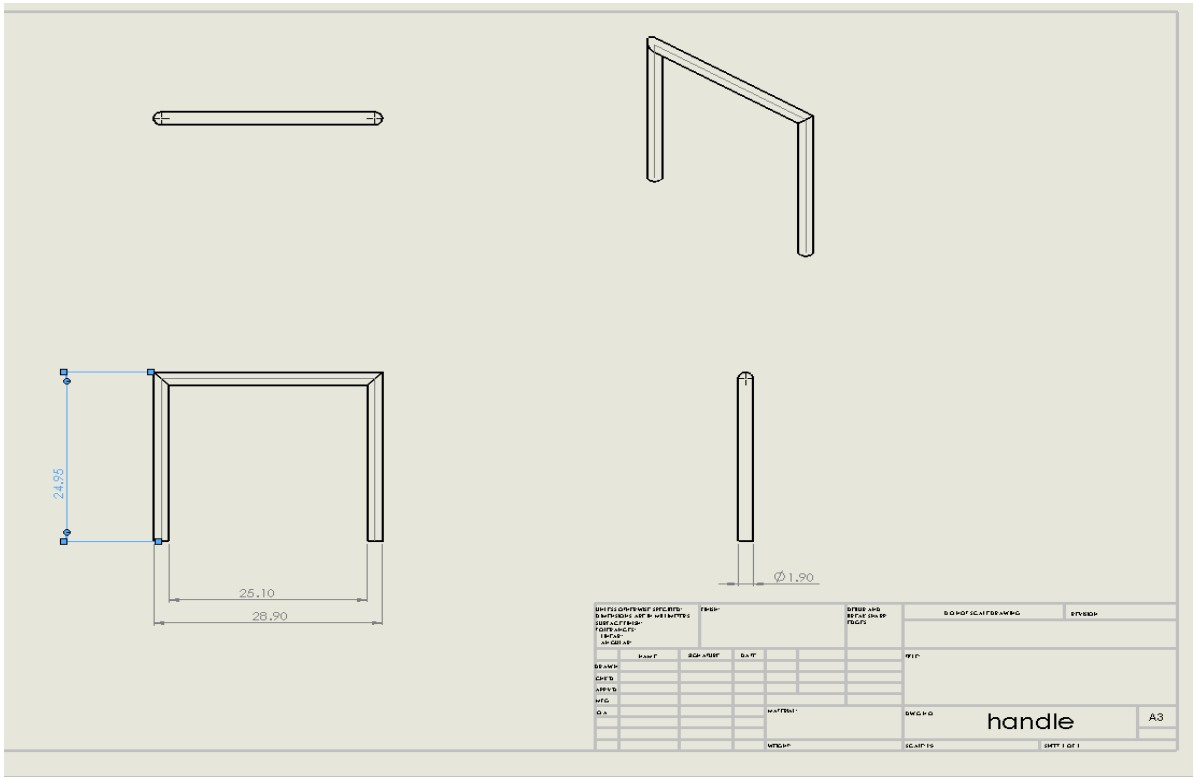
10000 WATT PRO SERIES™ PORTABLE GENERATOR

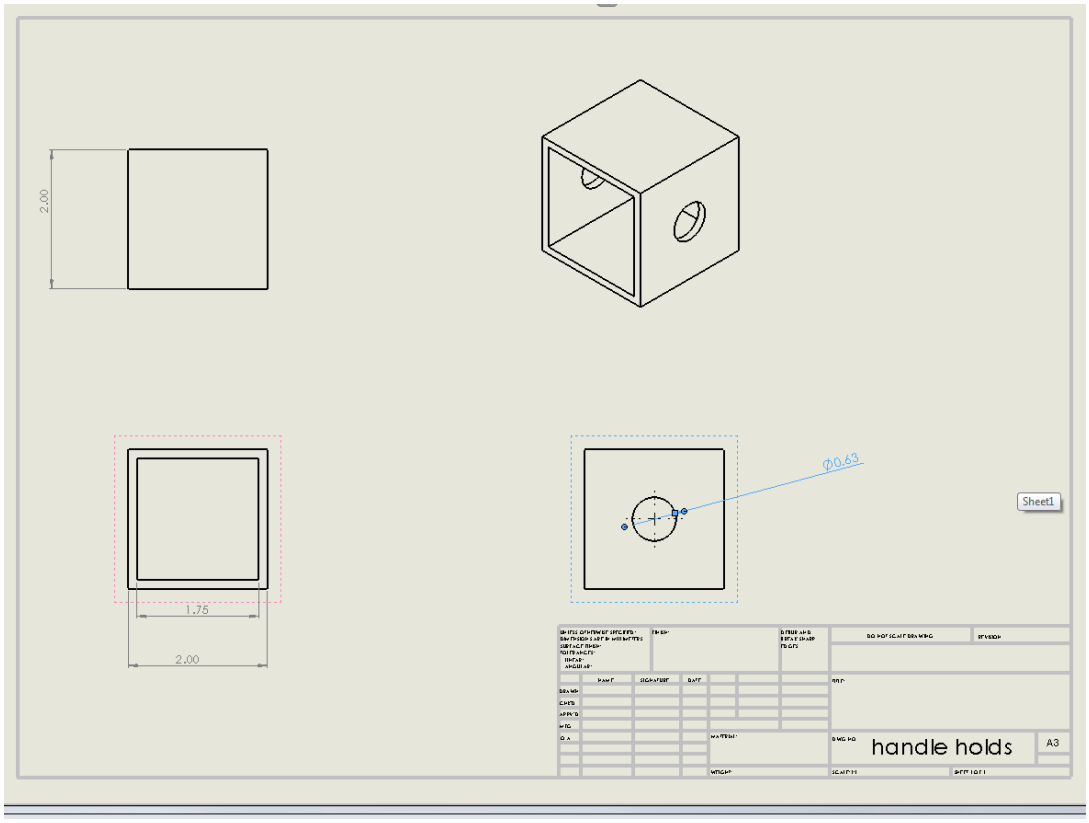
Engine Brand	Briggs & Stratton Vanguard
Engine Series	Vanguard
Alternator	PowerSurge
Start Type	Pull/Electric
Wheel Diameter (Inches)	10
Weight (lbs)	295
Length (in)	30
Width (in)	27
Height (in)	32

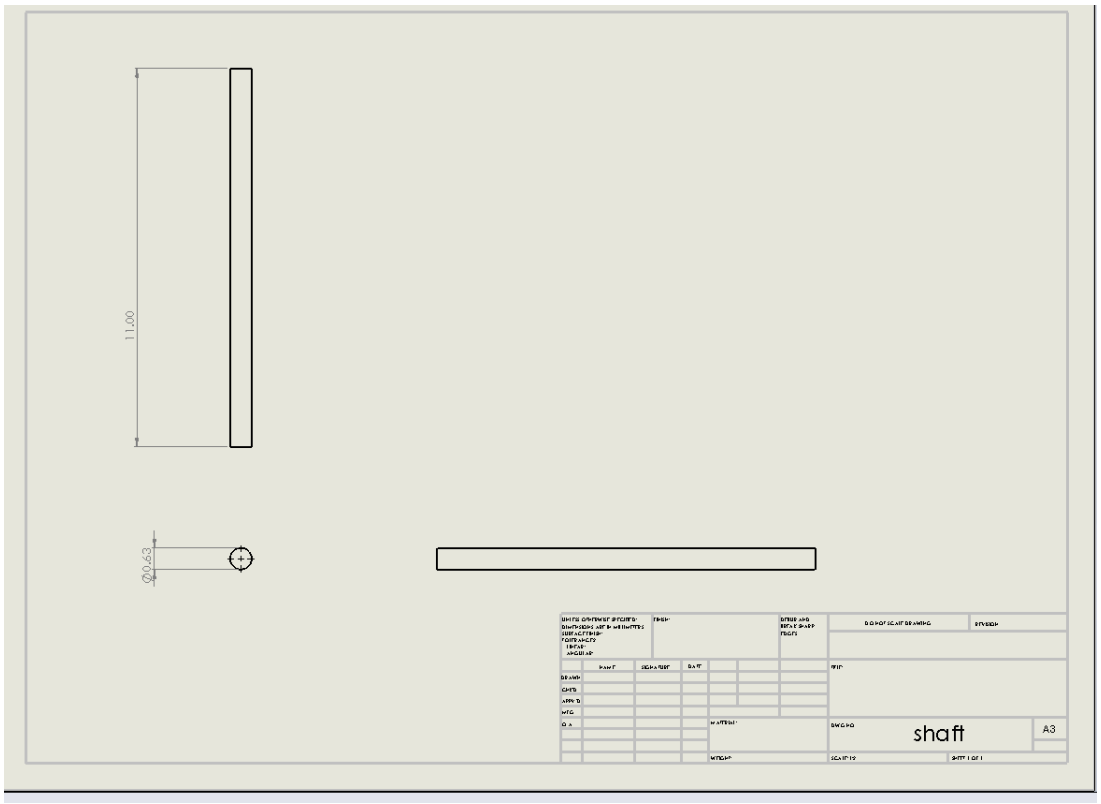
Pros:
- 4 wheels
- Roll cage for engine
-Engine Acess

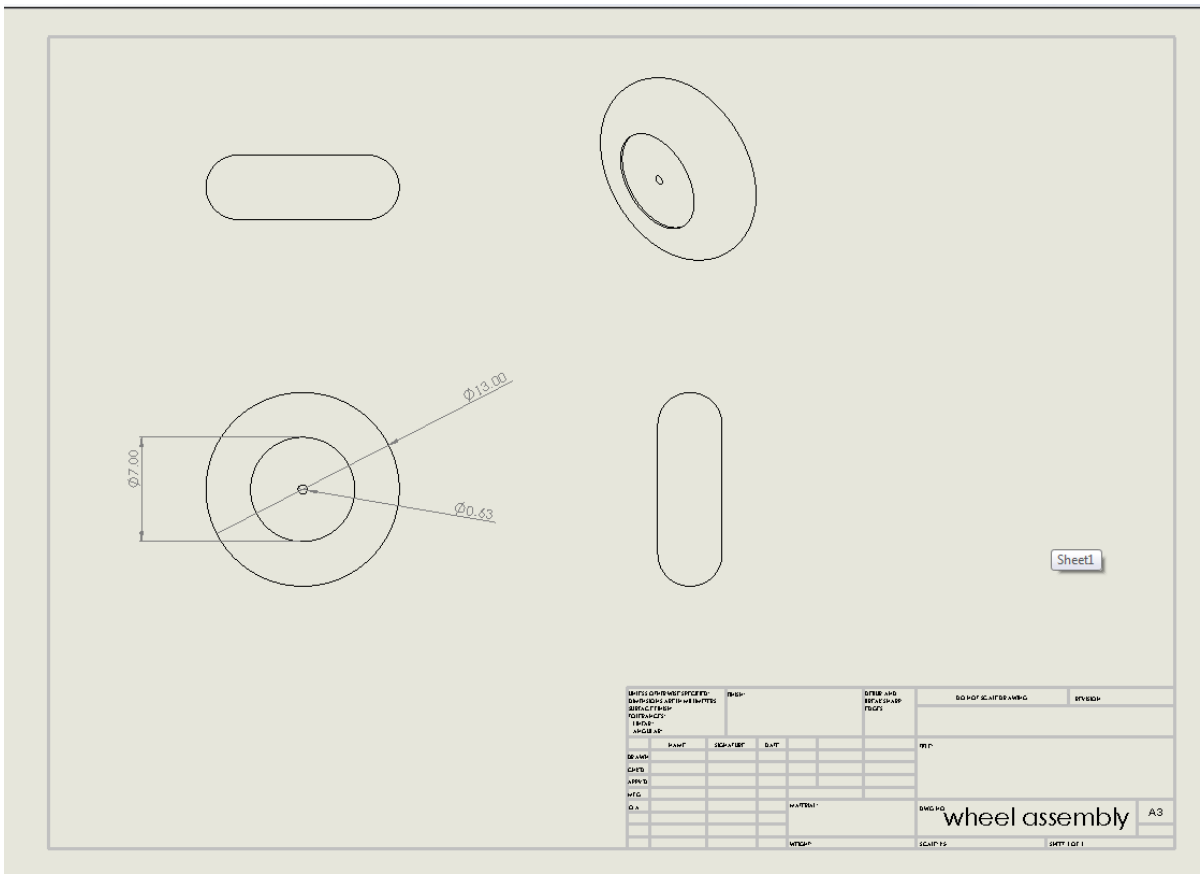
Cons:
-Heavy
-Not robust enough
-Minimal Wheel diameter
-No brakes
-$1,499


2.00
6.50
7.50
30.00
21.50
25.50
17.50
21.50
31.34
final model
A3


24.95
25.10
28.90
Ø1.90
handle
A3

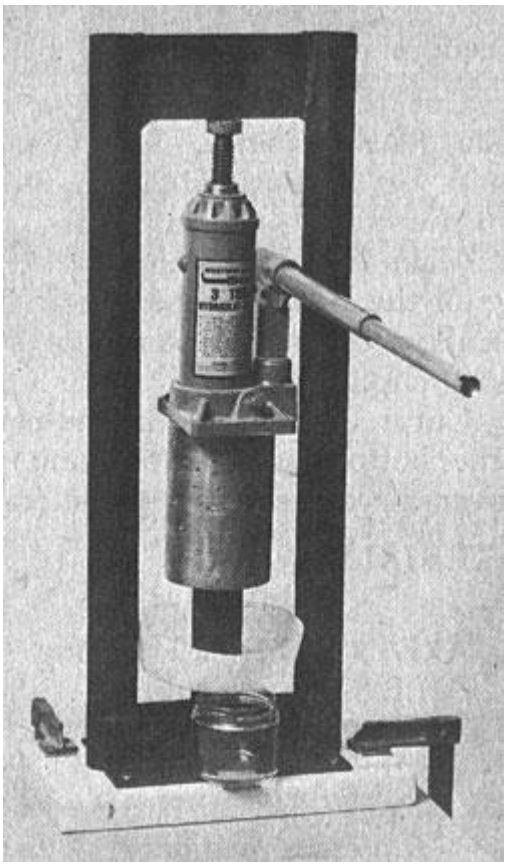

2.00
1.75
2.00
Ø0.63
Sheet1
handle holds
A3


11.00
Ø0.63
shaft
A3


Ø13.00
Ø7.00
Ø0.63
Sheet1
wheel assembly
A3

APPENDIX C – OIL PRESS RESEARCH AND DRAWINGS

Manual Oil Press  ***Figure 46 Screw-driven oil press*** • Manually operated • Universal: for all seeds and nuts with over 25% oil • Continuous pressing • Processing up to 5 kg seed per hour • Small size and low weight: 2.3 Kilo (5 pounds) • Original Piteba design • Holland hand made tool	-Manual Drive could be hooked up to an engine for a higher output. -Lower output than piston driven model -$149.99

Piston Driven Oil Press  ***Figure 47 Hydraulicly driven oil press*** -Manually Operated -Easy to produce with basic manufacturing resources(mill, drill press, arc welder, hack saw) -Most materials used(except hydraulic cylinder) can be improivsed from locally sourced scrap.	-Seeds must be ground prior to pressing -Hydraulic Cylinders must be purchased -Seeds must be heated separately -Very low output -Labor Intensive -Material Cost: $20-$40

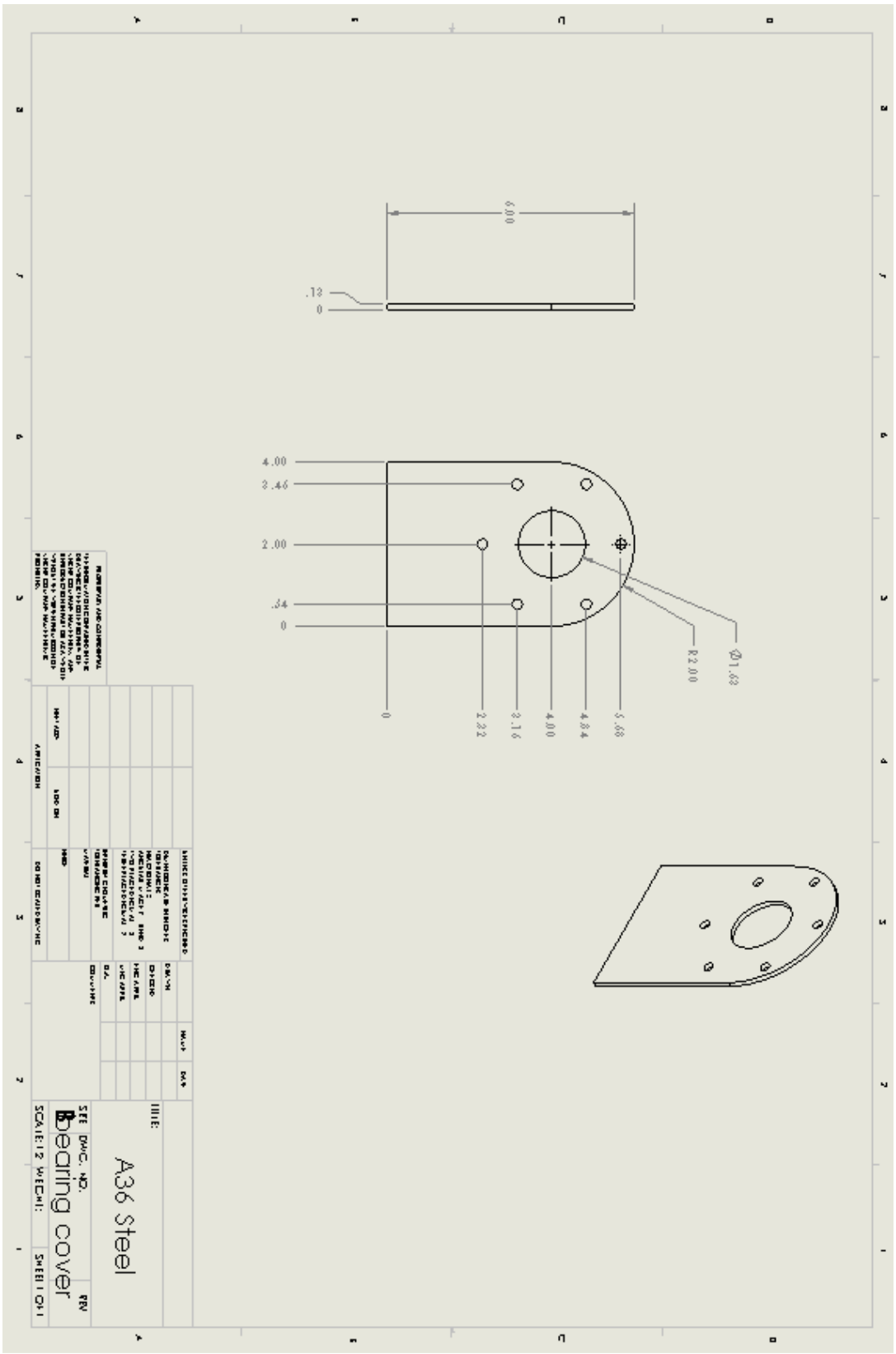


Figure 48 - Bearing Cover

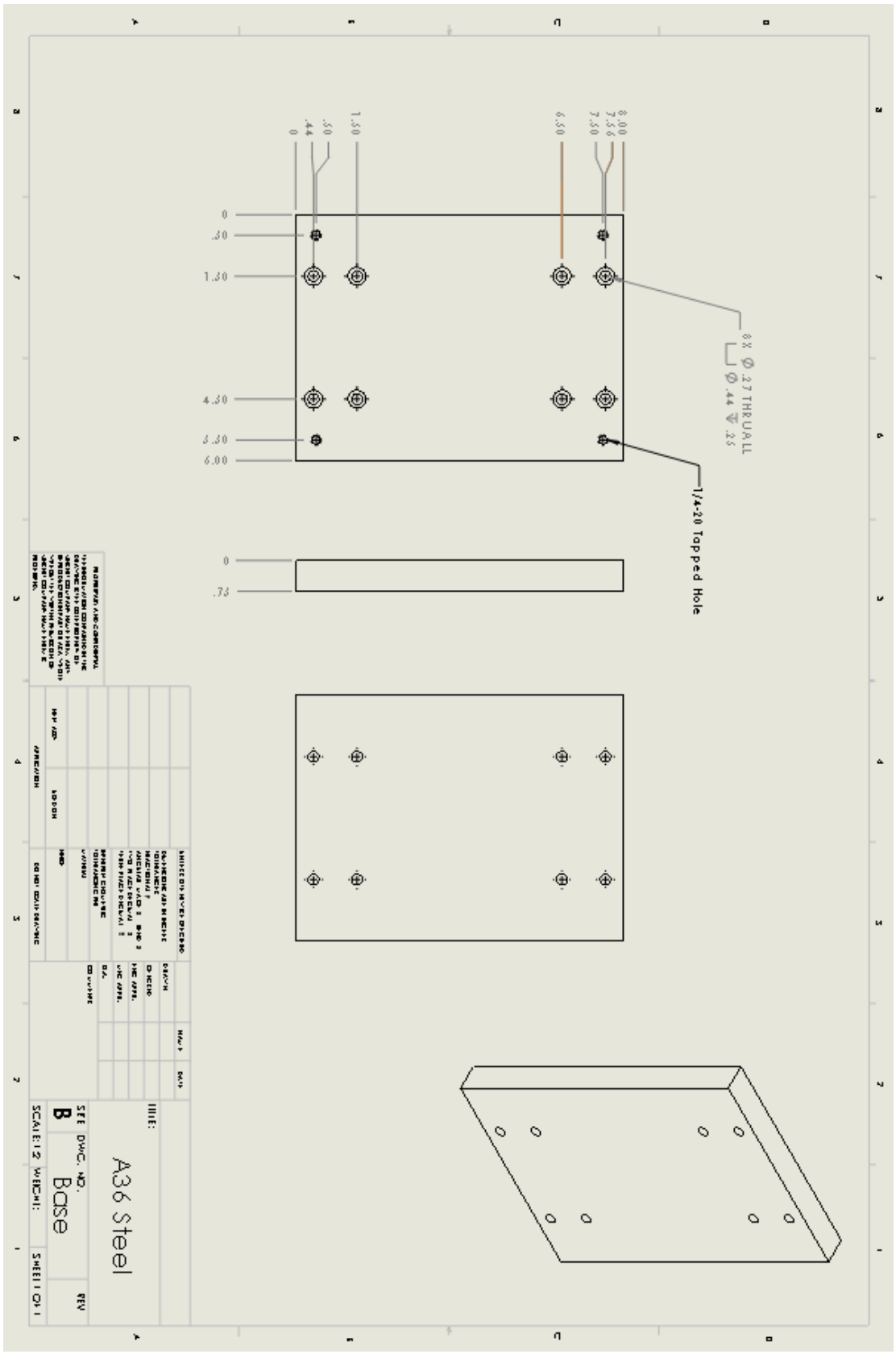


Figure 49 - Base Plate

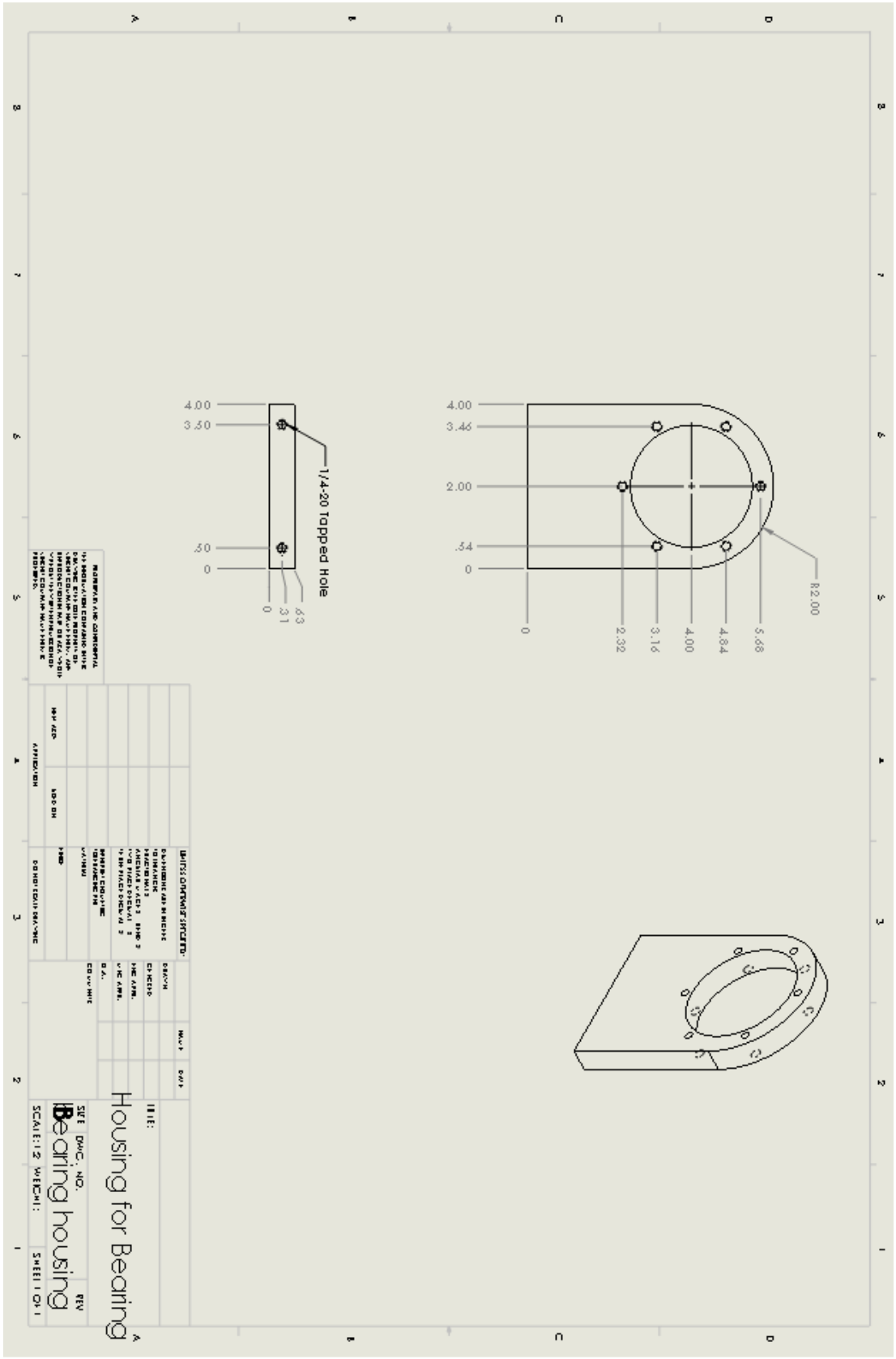


Figure 50 - Bearing Housing

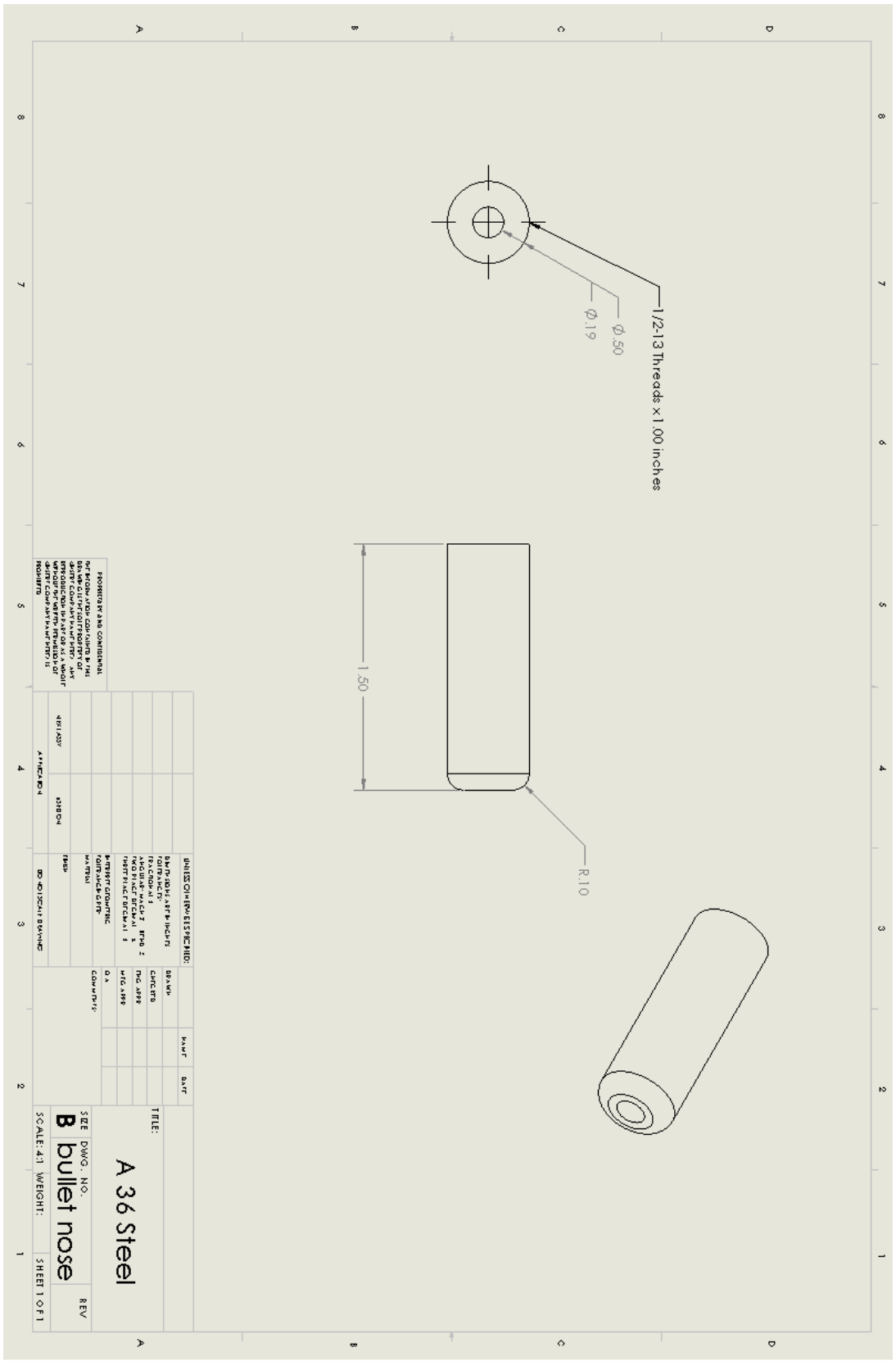


Figure 51 - Outlet Fitting

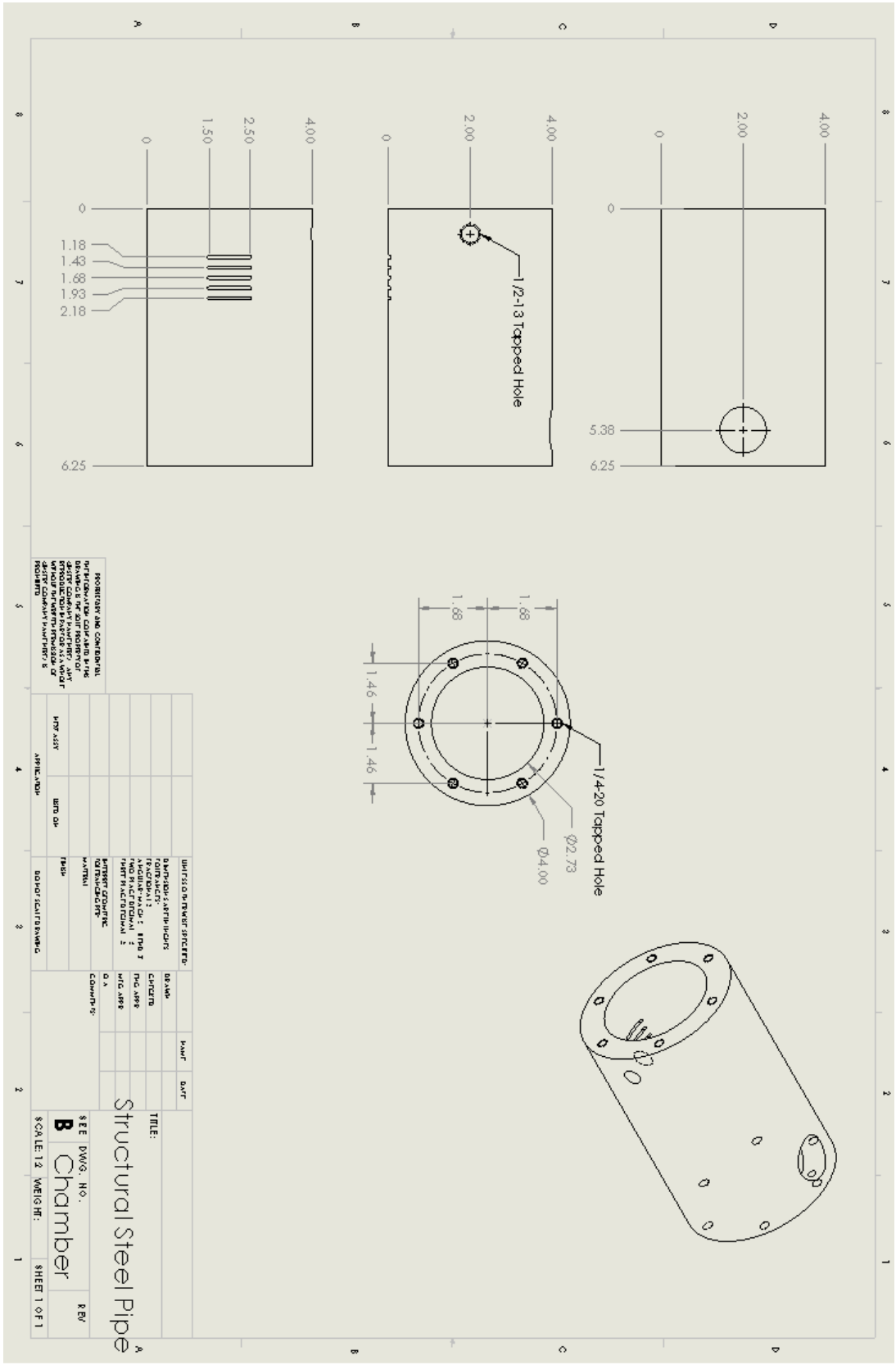


Figure 52 - Chamber

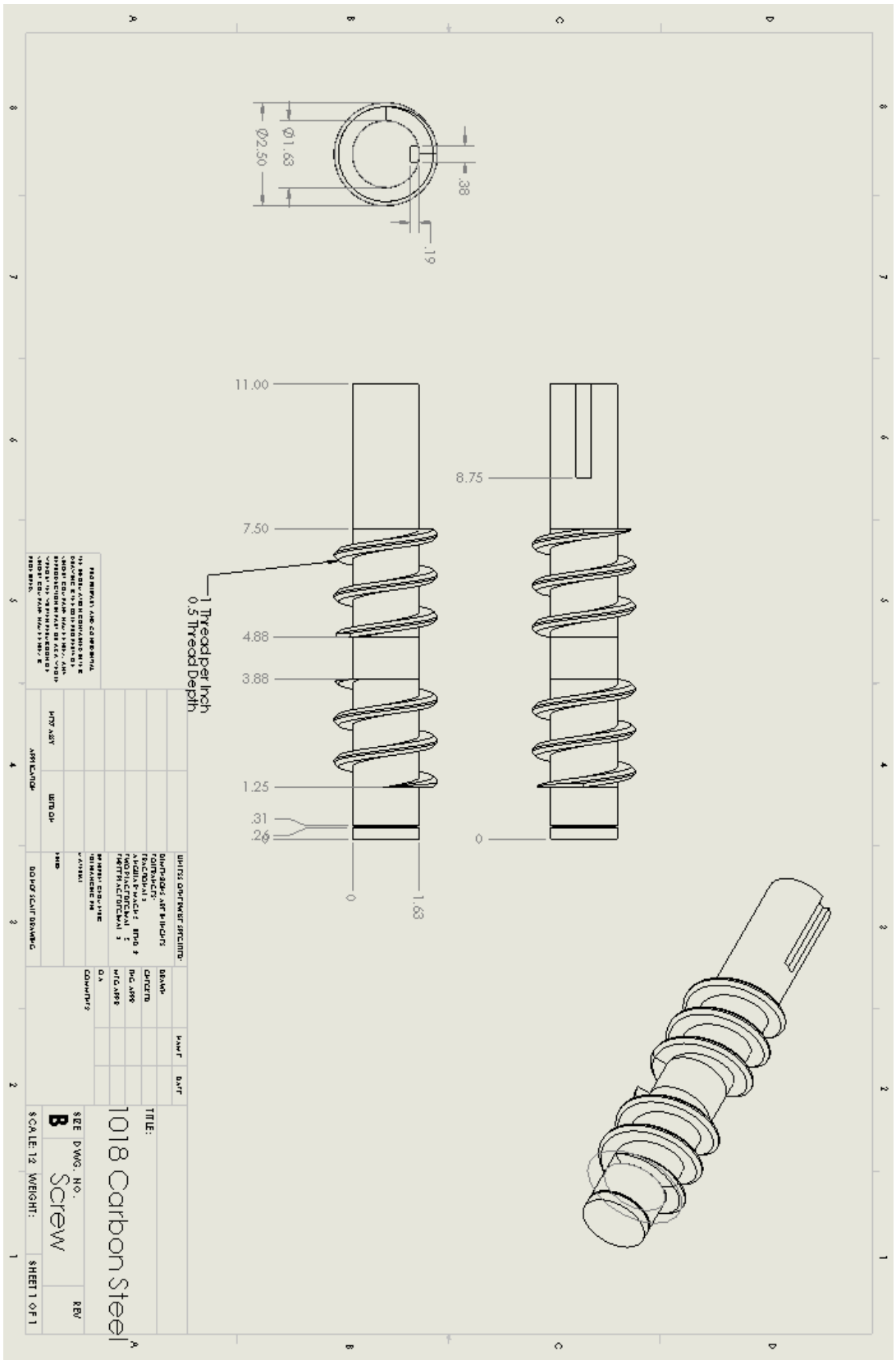


Figure 53 - Screw

APPENDIX D – INTERVIEWS

Interviews provided the needed detailed information about the overall project and helped narrow down the engineering designs to encompass the goals and desires of UC Forward. Beyond the initial interviews, the engineering group has attended weekly UC Forward group meetings to keep everyone informed and up to date with the newest information going into the engineering research and design.

Interview with Dale Murray conducted on 8/27/13: (513-608-8663) UC Industrial Design Department, Project Coordinator for non-engineering students.

-Confirmed our engineering goals while giving positive feedback with potential alternative routes to take.

-Suggested research opportunities and more contact information.

Interview with Dan Bode conducted on 8/28/13: (970-812-2186) Aywa International responsible for coordinating UC forward through Aywa.

-He gave specifics on engineering needs and requirements.
-Project leader
-Engineering wise:
-Pulley reduction instead of gears
-Attachment to tools
-Ease of use
-Oil press with refinement
-Better means of motor transport from site to site
-He will be available in person next month when he visits UC

Weekly meetings with UC Forward Group (beginning 9/13/13)

-Keep full group up to date with engineering progress, goals, and ideas
-Gives our team a better idea of what the whole group is striving to accomplish
-Dan Bode available via "Skype" to answer any detailed questions

In-person Interview with Dan Bode conducted on 10/4/13

-Details on Jatropha plant
-Talked about the workers skills and available machinery
-Talked about the environment which engine operates in
- Received new engine
- Discussed ideas for each individual design
- Summarized interviews

Interview with Brian Dobbs from D.A.A.P. regarding UC Forward trip to Senegal conducted on January 20, 2014

-Discussed UC Forwards trip in detail
-Details on the operating environment
-Work ethics of the Senegalese
-Poverty level was emphasized

APPENDIX E – SURVEYS

PULLEY REDUCTION SYSTEM
CUSTOMER SURVEY

The purpose of this survey is to determine the features that will best aid in the design of a pulley reduction system for a small utility diesel engine for the UC Forward Senegal Project. The feedback from this survey will weigh in on the product design and allow for an improved system that suits the customer best.

How important is each feature to you for the design of a pulley reduction system?

Please circle the appropriate answer. 1 = low importance 5 = high importance

							Average
Safety	1	2(1)	3(2)	4	5	N/A	2.66
Durability	1	2	3	4	5(3)	N/A	5
Ease of Operation	1	2	3	4	5(3)	N/A	5
Price	1	2	3	4	5(3)	N/A	5
Ease of Manufacture	1	2	3	4(1)	5(2)	N/A	4.66
Efficiency	1	2	3	4	5(3)	N/A	5
Multi-Output Speeds	1	2	3	4(1)	5(2)	N/A	4.66
Size	1	2(2)	3(1)	4	5	N/A	2.33

How satisfied are you with the current mode of engine operation?

Please circle the appropriate answer. 1 = very UNsatisfied 5 = very satisfied

							Average
Safety	1	2	3(1)	4	5(2)	N/A	4.33
Durability	1	2(1)	3(2)	4	5	N/A	2.33
Ease of Operation	1	2	3	4(2)	5(1)	N/A	3
Price	1(2)	2(1)	3	4	5	N/A	1.33
Ease of Manufacture	1(1)	2(2)	3	4	5	N/A	1.66
Efficiency	1	2	3	4(3)	5	N/A	4
Multi-Output Speeds	1	2	3	4	5	N/A(3)	N/A
Size	1	2(2)	3	4(1)	5	N/A	2.66

How much would you be willing to fund for this project? $50 $100, $100-$200, $200-$500 $500-$1000, $1000-$2000

Thank you for the feedback!

Engine Exoskeleton and Transporter
Customer Survey

The purpose of this survey is to determine the features that are considered most important for the engine exoskeleton for a small utility diesel engine that is to be designed for the UC Forward Senegal Project. The results from this survey will be used to better our design and also be more vigilant to the needs of the customer.

Please rate the following features with a level of importance
Please circle a number 1=lowest importance, 5=highest importance

							Average
Safety	1	2	3(2)	4	5(1)	N/A	3.66
Durability	1	2	3	4	5(3)	N/A	5
Ease of Operation	1	2	3	4(1)	5(2)	N/A	4.66
Ease of Maintenance	1	2	3	4	5(3)	N/A	5
Lightweight	1	2	3	4(1)	5(2)	N/A	4.66
Overall height	1(1)	2(2)	3	4	5	N/A	1.66
Ability to handle rough terrain	1	2	3	4	5(3)	N/A	5
Lifting mechanism for Engine	1	2(1)	3(1)	4(1)	5	N/A	3
Stabilizing mechanism	1	2	3(2)	4(1)	5	N/A	3.33
Ease of disassembly	1	2	3	4	5(3)	N/A	5
Versatile (different engines)	1	2	3	4	5(3)	N/A	5

How much would you be willing to fund for this project?
$50-$100x, $100-$200, $200-$500

OIL PRESS SYSTEM
CUSTOMER SURVEY

The purpose of this survey is to determine the features that will best aid in the design of an oil press system for the UC Forward Senegal Project. The feedback from this survey will weigh on the product design and allow for an improved product.

How important is each feature to you for the design of an oil press?
Please circle the appropriate answer. 1 = low importance 5 = high importance

							Average
Safety	1(1)	2(1)	3	4(1)	5	N/A	2.33
Durability	1	2	3	4	5(3)	N/A	5
Ease of Operation	1	2	3	4	5(3)	N/A	5
Price	1	2	3	4	5(3)	N/A	5
Ease of Manufacture	1	2	3(2)	4	5(1)	N/A	3.66
Efficiency	1	2	3	4	5(3)	N/A	5
Oil Output	1	2	3	4	5(3)	N/A	5
Size	1	2(2)	3(1)	4	5	N/A	2.33

How satisfied are you with the current method of extracting biodiesel fuel from Jatropha seeds?
Please circle the appropriate answer. 1 = very UNsatisfied 5 = very satisfied

							Average
Safety	1	2(1)	3	4(1)	5(1)	N/A	9.66
Durability	1(1)	2(2)	3	4	5	N/A	1.66
Ease of Operation	1(1)	2(2)	3	4	5	N/A	1.66
Price	1(3)	2	3	4	5	N/A	1
Ease of Manufacture	1(1)	2	3(2)	4	5	N/A	2.33
Efficiency	1(1)	2	3(2)	4	5	N/A	2.33
Oil Output	1	2(1)	3(2)	4	5	N/A	2.66
Size	1	2	3(2)	4(1)	5	N/A	3.33

How much would you be willing to fund for this project? $50-$100x, $100-$200, $200-$500
$500-$1000, $1000-$2000

Thank you for the feedback!

APPENDIX F – QUALITY FUNCTION DEPLOYMENT

Table 14 Pulley System Quality Function Deployment

James Anderson **UC Forward Senegal** **Pulley Reduction System** 9 = Strong 3 = Moderate 1 = Weak	Shields	Bolt torque specifications	Design for multiple loading conditions	Maintenance schedule	High grade materials	Mounted to engine carriage	Desgined for user manufacturing/assembly	Single bolt design	Color coded operating system	Adjustable shaft position	Built on single platform	Three output speeds to maximize engine operation	15:1 maximum speed reduction	Quick release for attachment coupler	Adjustable output shaft orientation		Customer importance	Designer's Multiplier	Current Satisfaction	Planned Satisfaction	Improvement ratio	Modified Importance	Relative weight	Relative weight %
Safety	9	3	9	1	3	1			3		3		1	3	1		2.7	1	4.3	5	1.2	3.1	0.06	6%
Durability	3	3	9	9	9	3			3		3			3	1		5	1	2.7	4	1.5	7.4	0.13	13%
Ease of Operation				1		3		1	9	3	3	3		3	3		5	1.1	4.3	5	1.2	6.4	0.11	11%
Price	3		3		9	1		3		1	3	3	3	3	1		5	1	1.3	2	1.5	7.7	0.13	13%
Ease of Manufacture	3		3		3	3	9	9		3	3			3	1		4.7	1	1.7	3	1.8	8.3	0.15	15%
Efficiency		1	3	3	3				1	3		3	3	3			5	1.05	4	5	1.3	6.6	0.12	12%
Multi-Output Speeds			3						3	3		9					4.7	1	1	3	3.0	14.1	0.25	25%
Size			3							1	3	3	1		3		2.3	1	2.7	4	1.5	3.4	0.06	6%
Abs. importance	1.73	0.67	3.77	1.68	3.33	1.35	1.31	1.83	2.42	2.06	1.91	3.49	0.87	2.08	0.98	29.5						57.0	1.0	1.0
Rel. importance	0.06	0.02	0.13	0.06	0.11	0.05	0.04	0.06	0.08	0.07	0.06	0.12	0.03	0.07	0.03	1.0								

Table 15 Oil Press Quality Function Deploy

Mike Compton UC Forward Senegal Oil Press 9 = Strong 3 = Moderate 1 = Weak	Guards for grinding mechanism and heating element	Recommended torque specifications for all bolts	Emergency Release	Material Selection	Designed for multiple loading conditions	Quick Disconnect/Coupler	Color Coding System for Different Applications	Maintenance Access Panel	Designed for User Manufacturing	Heating Element for Seeds	Calculat Feedrate for Highest Oil Output	Ability to be Disassembled		Customer importance	Current Satisfaction	Planned Satisfaction	Improvement ratio	Modified Importance	Relative weight	Relative weight %
Safety	9	3	9	1	1		3	1						2.3	3.6	4	1.1	2.6	0.04	4%
Durability	3	3		9	3			3						5	1.6	3.5	2.2	10.9	0.19	19%
Ease of Operation			3			9	9	1			1	3		5	1.6	4	2.5	12.5	0.22	22%
Price	1		1	3					1					5	1	3	3.0	15.0	0.26	26%
Ease of Manufacturability	1		1	1		1			9			1		3.6	2.3	3.5	1.5	5.5	0.10	10%
Oil Output								1		9	9			5	2.6	4	1.5	7.7	0.13	13%
Size	1		1	1						3				2.3	3.3	4.5	1.4	3.1	0.05	5%
Abs. importance	1.39	0.71	1.47	2.70	0.62	2.06	2.10	0.97	1.12	1.37	1.43	0.75	16.7					57.3	1.0	1.0
Rel. importance	0.08	0.04	0.09	0.16	0.04	0.12	0.13	0.06	0.07	0.08	0.09	0.04	1.0							

Table 16 Exoskeleton Quality Function Deployment

Reece McNulty Engine Exoskeleton Senegal Moving Forward 9 = Strong 3 = Moderate 1 = Weak	wheel size	Lifting mechanism	Brake selection	machineability	accessable Materials	lifespan	access doors/geometry	designed for multiple loading conditions	go-no-go gauge for pully guard	ground clearence		Customer importance	Designer's Multiplier	Current Satisfaction	Planned Satisfaction	Improvement ratio	Modified importance	Relative weight	Relative weight %
Durability	9	9	3		3	9	9	9	1	1		5	1.1	1	4	4.0	25.0	0.25	25%
Safety	9	1			3		3	3	3	3		3.6	1	1	3.8	3.8	16.0	0.16	16%
Ease of Operation	9	9	1			9	3	9	3	3		4.66	1	1	4.8	4.8	21.0	0.21	21%
Ease of Maintenance		9		3	3	9	3					5	1	1	4	4.0	20.0	0.2	20%
Cost		1	1		3							2	1	1	5	5.0	11.0	0.11	11%
Engine Swap Capabilities	1			1								5	1	1	2	2.0	7.0	0.07	7%
Abs. importance	5.65	6.21	1.07	0.67	2.16	5.94	3.96	4.62	1.36	1.36	33.00						100.0	1	100%
Rel. importance	0.17	0.19	0.03	0.02	0.07	0.18	0.12	0.14	0.04	0.04									

As shown above I chose a design multiplier of 1.1 for the durability because from the customer surveys durability was the most important requirement for the consumer. I changed the value to 1.1 so the exoskeleton would be able to exceed the requirements and not be just acceptable.

APPENDIX G – PRODUCT OBJECTIVES

Product Objectives for Pulley Reduction System

Multiple output speeds (25%)

Multiple output speeds are required to operate different applications at their proper speed/torque values. The multiple speeds will aid in the longevity of the engine, efficiency, and safer operating environments.

a. Minimum of three output speeds

Ease of manufacturing (15%)

It is imperative that the system is capable of being produced by the workers, with the machinery available to the workers. Assembly is also at the forefront of the design.

a. Ability to manufacture parts on vertical mill, lathe, and simple grinding machinery

Durability (13%)

Having a durable system will allow the Senegalese people to increase their overall output by removing the downtime required to fix mal-designed fixtures. Durability will also decrease the price in the long run.

a. Replaceable shaft bearings
b. Mounted to engine carriage to keep off ground
c. Pulley material grade at/above recommended design level
d. Sealed bearings are *ok* for wet conditions
e. Maintenance schedule
f. Can withstand one-foot drop

Price (13%)

Price is to be kept as low as possible. This can be accomplished by substituting certain manufactured parts with purchased parts, leaning out the manufacturing process, and keeping the overall process simple.

a. Must be under $500

Efficiency (12%)

Increasing efficiency will aid in a higher output from the fuel produced, reducing the Senegalese dependence on crude oil, and helping them have a higher crop output. The pulley system will help the engine operate at the desired torque/speed settings for each application. Operating at the desired settings, will reduce fuel use and help maximize the application output.

a. 15:1 maximum output speed reduction
b. Cross with engine specifications for maximum torque values
c. Adjust for biofuel consumption

Ease of operation (11%)

The pulley system will use multiple belts and pulleys to achieve the desired speeds/torque values. Setting the system up properly is required to achieve these desired speeds, and this must be instructed in a clear and concise method to prevent confusion of the

workers (some who are unskilled and illiterate). A system which incorporates symbols and colors would be ideal.

a. Design with color-coded system to help illiterate workers properly configure system
b. Switch belts in fewer than ten minutes
c. Operable on/off engine carriage
d. Quick release to adjust shaft position

Safety (6%)

Safety is of importance to the engineering designers, however, based on the feedback from the end-user, safety is of little importance.

a. Fully enclosed, removable case (pulley reduction)
b. Recommended torque specifications for all bolts
c. Safety factors for pulleys, shafts, and coupler
d. Design with shock loading involved

Size (6%)

Size is to correspond to engine exoskeleton size. Must be attached to engine exoskeleton.

a. Weight below 40lbs
b. Built within aluminum rectangular box frame

Weighted engineering characteristics list

Design for multiple loading conditions (13%)
Three output speeds to maximize engine operation (12%)
High grade materials (11%)
Color coded operating system (8%)
Adjustable shaft position (7%)
Quick release for attachment coupler (7%)
Shields (6%)
Maintenance Schedule (6%)
Single bolt design (6%)
Built on single platform (6%)
Mounted to engine carriage (5%)
Design for user manufacturing/assembly (4%)
15:1 maximum speed reduction (3%)
Adjustable output shaft orientation (3%)
Bolt torque specifications (2%)

Taking the customer results from the quality function deployment, all aspects of customer satisfaction were touched upon via adjusting the designer multiplier and providing an estimated customer satisfaction for the finished product. In order to get these desired results, the designer multiplier for “Ease of Operation” and “Efficiency” was changed from 1 to 1.1.

Objectives Engine Exoskeleton

Durability (25%)

a. Designed with shock loading in mind
b. Can withstand a 3'-4' foot drop
c. Minimal maintenance
d. Design for operator manufacture and assembly

Ease of Operation (21%)

a. Able to be moved by one person from site to site
b. Easy connect and disconnect capabilities
c. Ability to handle rough terrain
d. Lifting mechanism (crank driven scissor jack)
e. Moved by one person either by pulling or pushing
f. Engine accessibility (starting and adjustments)

Ease of Maintenance (20%)

a. Disassembles quickly
b. Made of standard parts available to the Senegalese

Safety (16%)

a. Stabilizing mechanism
b. Brakes for going down a steep decline
c. Made of easily disposable materials

Cost (11%)

b. Cost less than $500 to manufacture

Engine Swap Capabilities (6%)

b. Designed to house another engine within 20% of current engine

Product Objectives for Oil Press

Price (26%)

a. Low cost materials
b. Little to no commercial off the shelf components
c. Simplified part designs for low machining time

Ease of Operation (22%)

a. Quick Disconnect between oil press and engine
b. Color coding/numbering system for operation
c. Easy maintenance access panel

Durability (19%)

a. Material selection
b. Grinding mechanism enclosed in a protective housing
c. Designed for shock loading operation

Oil Output (13%)

a. Heating Element for Seeds
b. Meet or exceed benchmarked presses
c. Calculate the most effective feed rate for the highest oil output

Ease of Manufacture (10%)

a. All parts designed for manual lathes and mills
b. Easy to work with materials
c. Simplified part designs with low to medium skill machinists in mind

Size (5%)

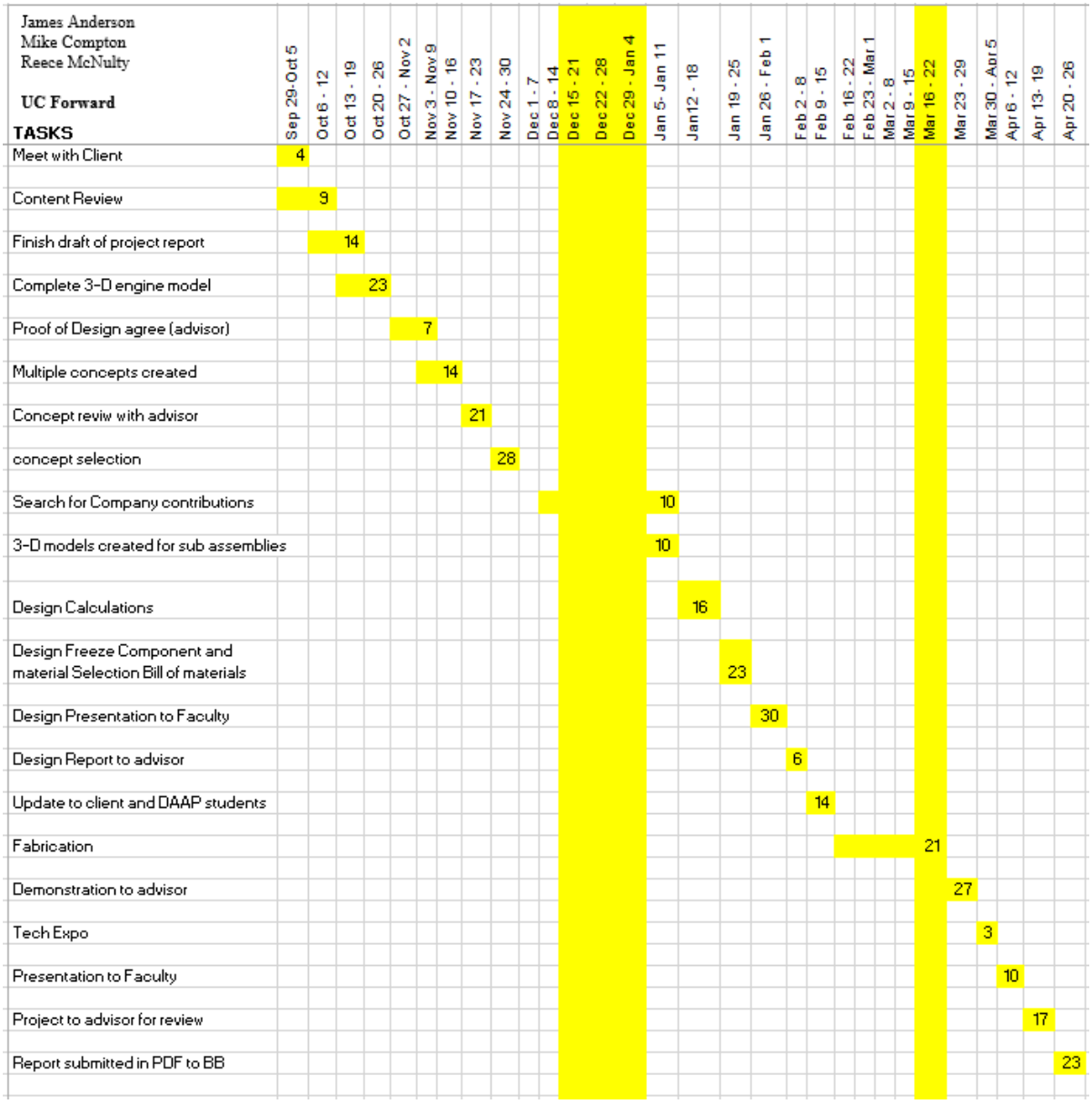
a. Keep design within 3 feet by 3 feet by three feet volume
b. Mobile
c. Total weight under 50 lbs

Safety (4%)

a. Guards for grinding mechanism and heating element
b. Safety factor for maximum press pressure
c. Recommended torque specifications for all bolts
d. Emergency release

APPENDIX H – SCHEDULE

Table 17 Schedule



APPENDIX I – BUDGET

Budget for Prototype Engine Exoskeleton

Materials

	Price	Qty.	Total
Cost			
Square Tubing			
8 Feet of 14 gauge 1-1/2" w/.083" wall	$24.96	2	
$49.92			
Shipping	$13.59	1	
$13.59			
Welding Rods			
Must coincide with material selection and forces	TBD		
Sub-Total			
$63.51			

Components

Brakes			
Alloy bike brake with all components	$16.37	1	
$16.37			
Tires			
15"X 6" max capacity 365 Lbs.	$19.75	4	
$79. 00			
Lifting screws			
Machining and material cost with infrastructure		**TBD**	

Grand Total without Welding rods & Lift Screws
$158.88

Budget for Power Transmission

	Proposed
Structural Materials	$150
Guard Material	$50
Pulleys	$100
Bearings	$50
Assembly Parts	$25
Belts	$25
Labor	$50
Shipping	$50
Total	$500

O R D E R A C K N O W L E D G M E N T
Online Order Status review available by registering at Applied.com

ORDER # 10544394 DATE: 03/03/14 PAGE 1
TIME: 12:42:28

BILL TO:
UNIVERSITY OF CINCINNATI
INVOICES - ACCTS PAY
PO BOX 212000
CINCINNATI, OH 45221-0001
TERMS: CASH
ATTENTION OF: JAMES ANDERSON
PO# CC-ANDERSON REQ#
SHIP VIA: UPS Ground By EOB -End of Business Day

SHIP TO:
JAMES ANDERSON -U.C.
2433 WEST CLIFTON AVENUE
CINCINNATI, OH 45219-1314

ACCT: 1056194
PHONE: (513) 556-6742

REQ 03/10/14

QUANTITY	MFG	PART	DESCRIPTION	UNIT PRICE	UM	EXTENDED PRICE	
1.00	SSMC	1.0"KEYEDX24"1144	MISC. FINISHED PARTS	26.940	EA.	26.94	*
1.00	SSMC	1.25"KEYEDX24"1144	MISC. FINISHED PARTS	37.320	EA.	37.32	*
2.00	CLIM	2C-100	SHAFT COLLAR	4.250	EA.	8.50	*
2.00	CLIM	2C-125	SHAFT COLLAR	5.040	EA.	10.08	*
1.00	PBP	14249	KEYSTOCK	1.090	EA.	1.09	*

MATERIAL SUBTOTAL: $83.93 ORDER IS NOT SUBJECT TO TAX

Shipping & handling charges apply 28.740

Thank you for your order which is accepted subject to the standard terms and conditions of sale available at www.Applied.com.
Returned product is subject to a restocking charge.
Material Safety Data Sheets (MSDS) are also available on the Applied.com Web site.

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10850 MILLINGTON CT.
BLUE ASH, OH 45242-4017
PHONE: (513) 242-1600
FAX : (513) 242-1719
Email-address: SC0105@APPLIED.COM
Please contact Dennis Daniel with any questions on your order

Biedler's Belts

Hello, Jamie Anderson

Thank you for your order from Biedler's Belts. Once your package ships we will send an email with a link to track your order. If you have any questions about your order please contact us at support@biedlers-belts.com or call us at (304) 263-9995 Monday - Friday, 8am - 5pm PST.

Your order confirmation is below. Thank you again for your business.

Your Order #200000669 (placed on March 19, 2014 8:14:13 PM EDT)

Billing Information:

Jamie Anderson
2433 West Clifton Avenue
Cincinnati, Ohio, 45219
United States
T: 3309803640

Payment Method:

PayPal Express Checkout
Payer Email:
ander2j2@mail.uc.edu

Shipping Information:

Jamie Anderson
2433 West Clifton Avenue
Cincinnati, Ohio, 45219
United States
T: 3309803640

Shipping Method:

United Parcel Service - Ground

Item	Sku	Qty	Subtotal
Gates SDS 1. Bushing	7838-2100	1	$15.12
Gates JA 1. Bushing	7838-0100	1	$9.52
Gates SH 1. Bushing	7838-1100	1	$13.02
Gates SK 1.1/4 Bushing	7838-4104	1	$23.42
Gates 3VX366 Belt	9412-0366	2	$15.76
Gates AP52 Belt	9185-0052	3	$25.77
Gates QD2/3V3.15 Sheave or Pulley	7872-2031	1	$31.07
Gates QD3A3.4/B3.8 Sheave or Pulley	7831-3034	1	$43.85
Gates QD3A8.2/B8.6 Sheave or Pulley	7831-3082	1	$87.71

Gates QD2/3V8.00 Sheave or Pulley	7872-2080	1	$65.78
		Subtotal	$331.02
		Shipping & Handling	$26.40
		Grand Total	**$357.42**

Thank you again, **Biedler's Belts**

Budget for Prototype Oil Press

Preliminary Budget

Table 16 Oil Press Preliminary Budget

Materials	$100
Labor	$75
Rubber Hose	$25
Fuel for heating mash	$20
Miscellaneous	$50
Shipping	$100
Total	$370

Final Budget

Drive System	$287.29
INA Thrust Bearings	$107.20
Raw Materials	$221.99
Assorted Fasteners	$25
Funnel	$15
Total	$756.48

Budget Comparison

Preliminary	$370
Final	$756.48
Difference	386.25

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