MAE 2250

Water Pump Project

Dual-Piston Scotch-Yoke Water Pump

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Table of Contents

A Dual-Piston Scotch-Yoke Water Pump

A.1 Description & Mode of Action

We designed a dual piston scotch yoke water pump. The scotch yoke mechanism translates rotational energy from the drive shaft into the linear motion of the piston arms which are used to pump water. As it is dual acting, one piston intakes water as the other simultaneously outputs water into the reservoir.

Our design is grouped into three assemblies, composed of two cylinder assemblies and the scotch yoke assembly. The scotch yoke assembly is made of a rectangular arm and ¼-20 screw as a pin, interfacing with a rectangular slot bar. The cylinder assembly includes the piston head attached to a threaded rod, which is then connected to the slot bar. The two cylinder assemblies are fastened to a supporting face plate with two mount blocks.

B Design Process

B.1 Preliminary Designs

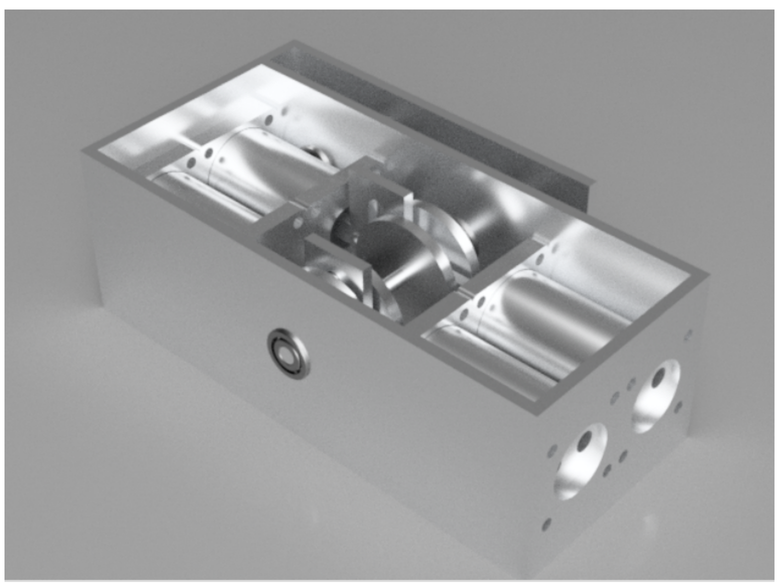


Figure B.1: *Design 1: Callen & Isaac*

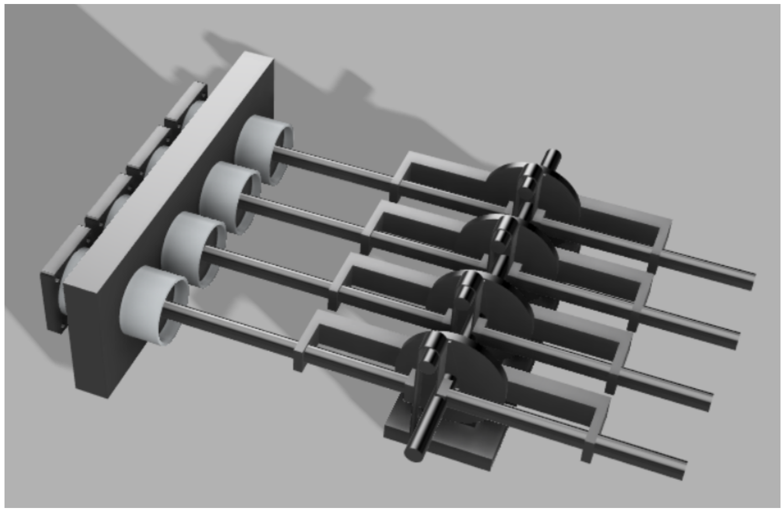


Figure B.2: *Design 2: Sean & Julianna*

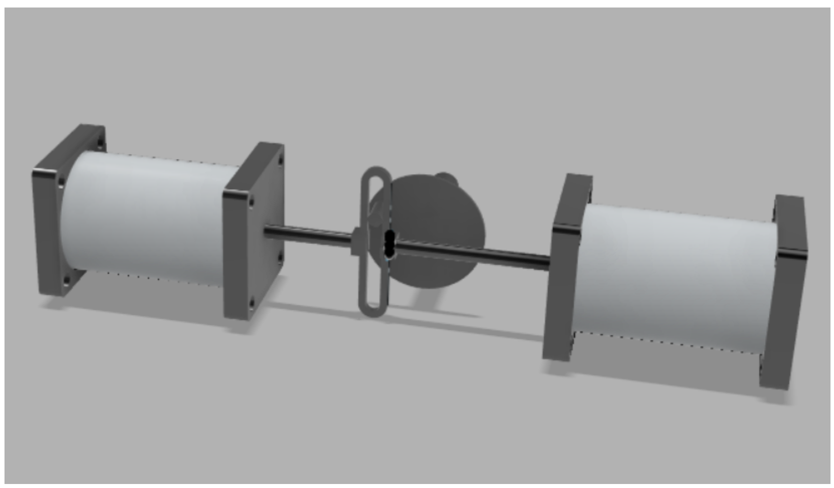


Figure B.3: *Design 3: Maia & Stephanie*

Analysis

During our first meeting as a group, we discussed our preliminary designs. We noticed that we all had incorporated a scotch yoke to convert rotational motion to linear motion, so we decided to include this mechanism in our final design. We elaborate on this decision in ***Sec. III*** as it relates to our final pump design—though the line of reasoning was the same for our preliminary designs. There were other similarities among the initial designs: Designs 1 and 3 both had the pistons in parallel, and designs 1 and 2 maximized the number of input and output tubes (*see:* ***Fig B.1–B.3***) (4). After discussing the possibilities, we decided to pick a simple design similar to Design 3. This design involved the most reasonable machining expectations due to its low part count, considering our current skill levels and schedules, and it was also the most compact of the three. Having only two pistons in parallel configuration would ensure that our pump could fit within the specified dimensions easily, even given significant modifications to the design (*See:* ***Sec. III.3***). The other two designs would have had to adhere strictly to the CAD dimensions to stay within a fourteen-inch cube. Although it doesn't include the maximum allowable input and output tubes, we expected that Design 3 would perform reasonably well if we were able to focus our efforts to perfect the design instead of adding more piston-cylinders and reducing the available time for testing and assembly.

B.2 Design Process Rationale

We selected a scotch yoke design to convert rotational motion to linear motion. The scotch yoke's main benefit is its simplicity: it requires few machined parts and yields perfect sinusoidal motion. Its range of motion and relative torque output are similar to more-complex systems involving linkages or cams, but it requires no hinges or springs. We additionally chose to use two pistons for our pump, keeping in mind that we would have an inlet and outlet on each of the endcaps, both only allowing flow in one direction. With one piston on either side of the scotch yoke, we would be able to accomplish a nearly-continuous flow with simultaneous intake and output from the pistons—as one extends, the other retracts. Moreover, the two-piston design helped us avoid problems with off-axis forces, because they would be distributed between two supports instead of creating a moment about one of them. This would be an easy adaptation from a single piston and would still be fairly simple to construct. To hold the pin in the scotch yoke assembly, we ultimately decided to use a rectangular arm instead of a circular plate. We chose this shape because it would require less material and be easier to manufacture, and its flat surface would let us easily add a set screw to hold the D-shaped end of the drive shaft.

B.3 Functional Decomposition

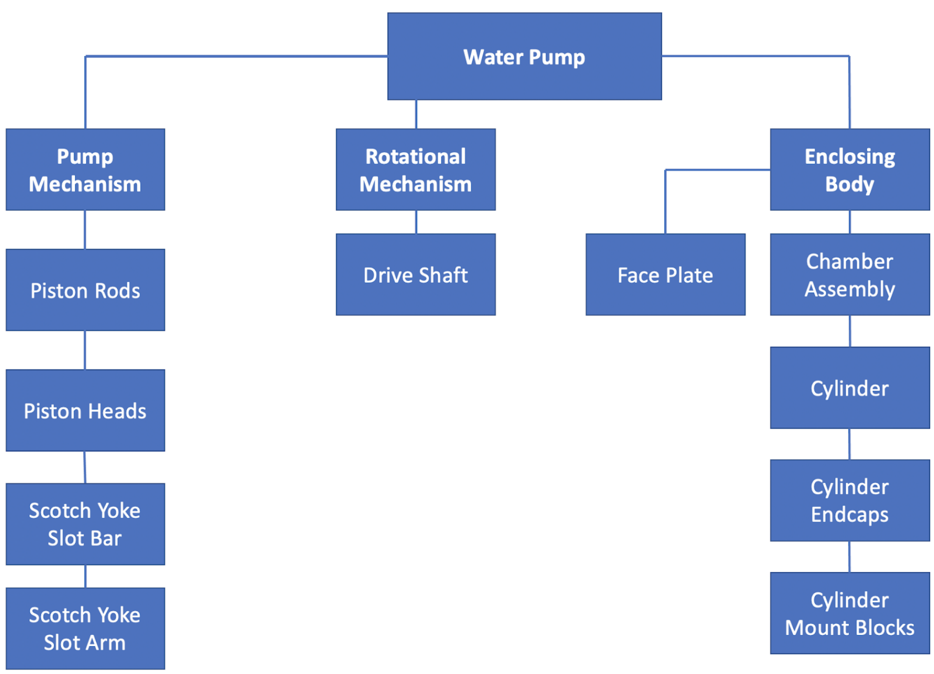


Figure B.4: *Functional Decomposition*

B.4 Morphological Chart

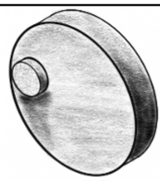
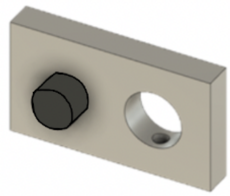
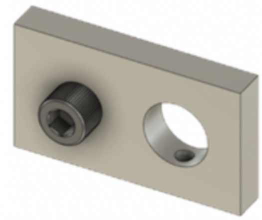
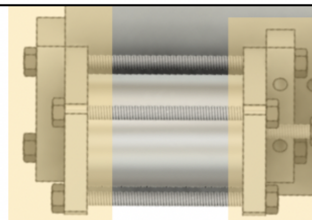
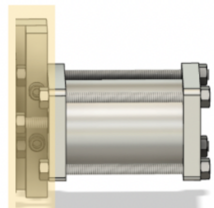
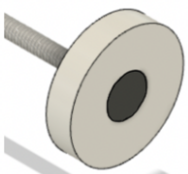
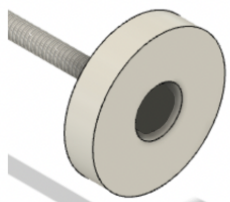
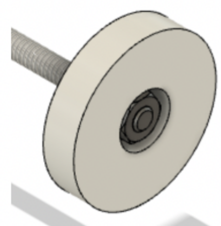
Scotch Yoke Mechanism	 Circle with machined pin	 Arm and machined pin	 Arm and screw as pin
Mount Blocks	 Inner and outer mount blocks	 Inner mount block only	
Piston Head Attachment	 All the way through with epoxy	 Halfway through with epoxy	 Through hole (not threaded) with one nut on each end

Figure B.5: *Morphological Chart*

B.5 Design Development

Once we chose our design, we modeled the parts and assembly in CAD using Fusion 360. Once we had a general idea of how we wanted our parts to work, we decided to model the scotch yoke portion of our design using a cardboard prototype (*see:* ***Fig B.6***). This tangible model helped us visualize the translation of the drive shaft's rotation into linear motion for the pistons. This prototype was particularly useful because it confirmed the mechanism would work before we finalized our CAD and began manufacturing.

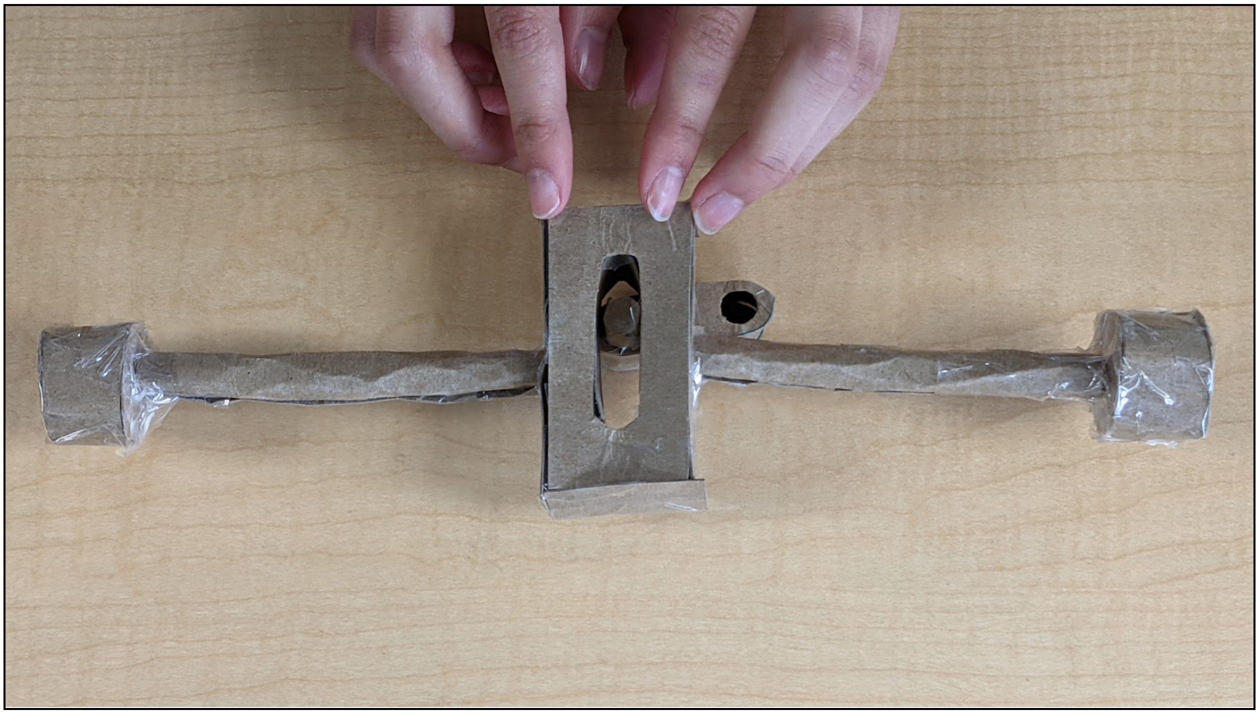


Figure B.6: *Rotational-to-Linear Motion, Cardboard Prototype*

One aspect of our design that was developed throughout our process was the scotch yoke arm (See: ***Fig B.5***). We originally planned a circular piece with a pin because it was the most typical shape we saw during our initial research. However, we decided that we could change the shape to a rectangular one because it would be easier to machine and would also integrate better with our D-shaped drive shaft. We also realized we could use a screw as the pin instead of trying to machine a cylinder and press fit it into the arm. We also went through a similar iterative process with the piston head (*See:* ***Fig B.5***). We wanted the bottom surface to be as flat as possible to ensure our piston could travel as far as possible without interference. We considered epoxy, but eventually decided to use a threaded rod. This would require us to tap the piston itself and then thread the rod through the hole. Since we would have to secure the rod with a nut, we chose to counter-bore the piston head to ensure the nut would be flush with the PVC.

One of the key challenges faced in our design process was determining how to best mount the cylinders onto the face plate without interfering with the 1.5 in. clearance for the provided mounting boss. This was the driving force behind many design changes. We initially used extra supports at both ends of each piston, reasoning that the entire piston sub-assembly could be mounted to the faceplate with enough distance normal to the surface of the plate such that it would not enter the clearance region. However, we found that the faceplate itself would be obstructing the clearance region regardless of where the cylinders were mounted. Furthermore, we would have had to spend almost all of our McMaster budget on a face plate long enough to hold the outer edges of both cylinders.

Moving forward, the design was then changed such that the far-end supports could mount into the face plate by inserting threaded rods normal to the surface facing the piston, and having the threaded rods holding the piston sub-assembly together interface with the support block to constrain its position relative to the faceplate. Upon measuring the clearance leftover from this design iteration, we once again found that it would interfere with the 1.5 in. region around the boss.

We iterated our support-block design a third time to have the far-end support blocks fastened to the ends of the faceplate (*See:* ***Fig B.5***), using screws parallel to the line of action of the pistons. This allowed for the faceplate length to be reduced by about one and a half inches in total. Still, we found that the clearance wasn't met unless the entire assembly was fastened to the lab-provided mounting plate at an angle of more than 35 degrees to avoid the boss entirely.

The far-end support block idea was subsequently discarded, and instead two separate support blocks of similar size was proposed, alongside a rectangular indent which could be milled into the faceplate to allow for extra clearance. While this design did finally provide enough clearance, after some close inquiry into similar prior pump designs, we determined that fastening the cylinder to the ends of the face plates was not strictly necessary to maintain the structural integrity of the assembly. Consequently, we removed the far-end supports entirely from the design and halved the length of the faceplate, saving about $15. This final decision solved another problem: our faceplate stock would have otherwise been unruly to machine.

Another problem which required significant time to solve was determining how to transmit the torque from our drive shaft into the scotch-yoke. Initially, we had chosen the drive shaft to be a D-shaft which interfaces with a corresponding low-tolerance D-shaped hole in our scotch-yoke. However, upon further discussion and inquiry, we found that the D-shaped hole would have been excessively difficult to machine given the time and resources available during the fabrication phase of the project. We instead opted to use a steel set screw which could be inserted normal to the flat of the D-shaft, threaded into the scotch yoke arm. The two circular holes would be much simpler to machine, allowing for much more time to assemble and test the device.

Further along in the manufacturing process, we expected that there would be additional difficulties regarding water-proofing the piston assembly. This would have made the pump more efficient and increased the output flow rate, but would have required significant modifications to the design in order to seal it. Mainly, it would have required epoxies to be used on the interfaces between piston parts and fasteners, but it would have also required gaskets or O-rings to be affixed to the piston heads. In particular, using an O-ring on the pistonhead would have required machining of a channel for the O-ring and precise tolerancing for the piston to still function afterwards.

B.6 Performance Calculations

The first design analysis we did was a power calculation for the system, which allowed us to then get an initial estimate of the flow rate for the dual piston setup. This analysis uses a significant number of simplifying assumptions, but nonetheless gives us a back-of-the-envelope solution for how our pump will perform prior to actually testing it. The main assumptions are as follows:

1. Frictional forces are negligible
2. Each part is modeled as a rigid body
3. The fluid mechanical system can be modeled as hydrostatic

The calculations are below:

Givens:

$$\rho_w = 1000\ kg/m^3$$
$$g = 9.81\ m/s^2$$
$$h = 1.5\ m$$
$$D = 1.77\ in. = 0.044938\ m$$
$$\ell = 0.875\ in. = 0.0222\ m$$
$$\omega = (900 RPM)\left(\frac{9}{70}\right)\left(\frac{2\pi\ rad}{rev}\right)\left(\frac{1\ min}{60\ s}\right) = 12.1\ rad/s$$

Calculations:

$$P = \rho_w g h = \left(1000\ \frac{kg}{m^3}\right)\left(9.81\ \frac{m}{s^2}\right)(1.5\ m) = 14715 \frac{N}{m^2}$$
$$A = \frac{\pi D^2}{4}$$
$$F_w = PA = \frac{\pi g h D^2}{4}$$
$$P_{max} = \omega_{max}\tau_{max}$$

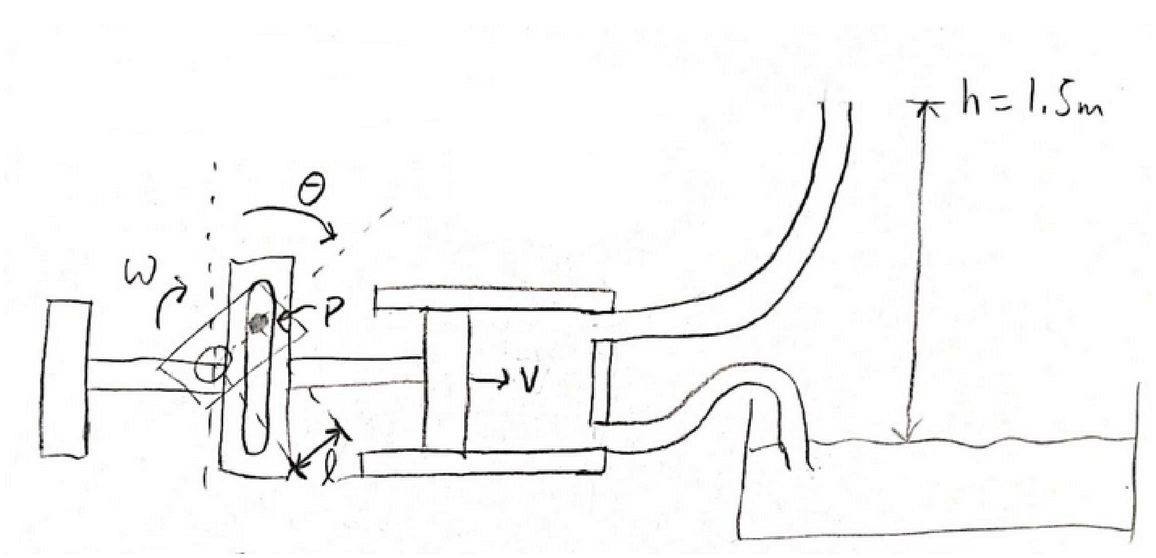
Free Body:

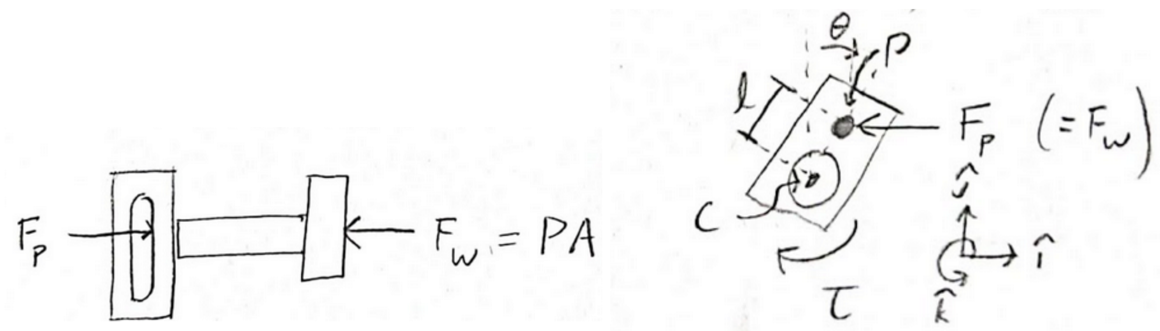
$$\Sigma \vec{F} = \cancel{m}^0\, \vec{a}$$
$$F_p\hat{i} - F_w\hat{i} = 0 \qquad \rightarrow \qquad F_p = F_w$$
$$\Sigma \vec{M}_c = 0$$
$$-\tau\hat{k} + \ell(\cos\theta\,\hat{j}) \times (-F_p\hat{i}) = 0$$
$$\left(-\tau\hat{k} + \ell(\cos\theta\,\hat{j}) \times (-F_p\hat{i}) = 0\right)\cdot\hat{k}$$
$$\tau = F_w\,\ell\,\cos\theta$$
$$\tau_{max} = F_w\,\ell = \pi\rho_w g h \ell\omega D^2/4$$
$$P_{max,1} = \pi\rho_w g h \ell D^2/4$$
$$P_{max,2} = P_{max,1}$$

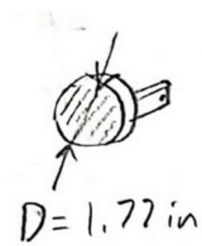
Right piston and left piston have τ_{max} occur at $\theta = 0^\circ$ and $\theta = 180^\circ$ $\therefore$ the two have the same magnitude and are out of phase

Plugging in givens:

$$\boxed{P_{max} = 6.30\ W = 0.00844 HP}$$







B.7 Stress Analysis

Finite Element Analysis (FEA): Harmonic Response Stress
To determine if there were any stresses requiring thicker cross sections in our final design, we utilized a harmonic response stress analysis within ANSYS. The design was modeled with supports in face plate holes and used a rotating force of 3600 N at the drive shaft—corresponding to the full-load torque on the motor (*see:* ***Eq. 2***, ***Appendix D***). The frequency was then modeled to rise from 0 to 5 Hz—corresponding to nearly double the max RPM on the motor output shaft. All the contacts were then readjusted to meet realistic expectations for the pump and a material assignment was applied to each part. Observing the resulting stress contours (*see:* ***Fig B.7***), we can see that the max stress does not exceed 8 MPa.

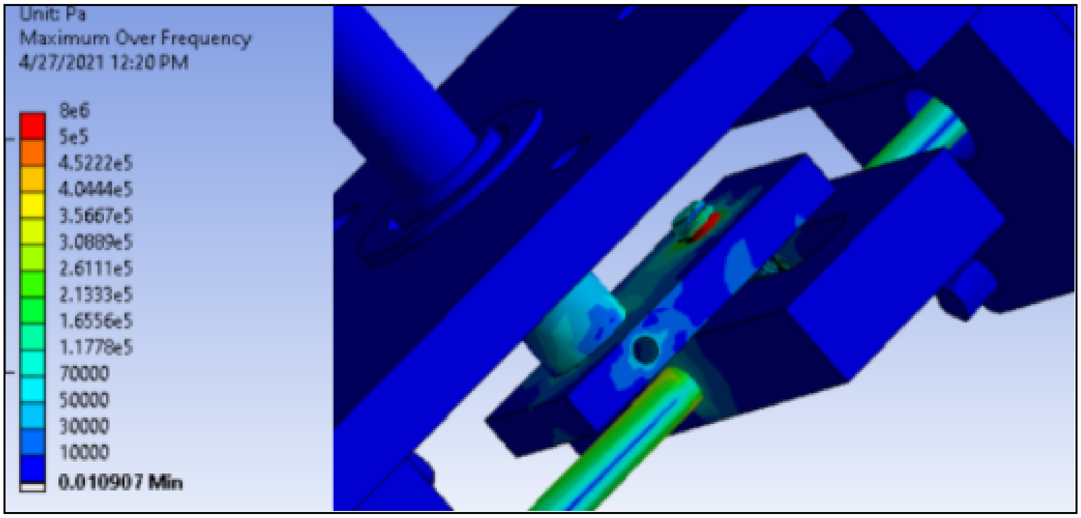


Figure B.7: *Max Stress Contours Over Frequency Range*

Since the yield strengths of aluminum 6061 T6, 1012 steel, and PVC are approximately 276 MPa, 310 MPa, and 42.6 MPa respectively, our materials are all an order of magnitude stronger than the max predicted stress anywhere in the design [1][2][3]. This therefore corroborates our second assumption for the power calculation, as we can expect only marginal elastic deformation. Moreover, because we observe that the max stresses (in red) only significantly appear within the scotch yoke pin (*see: **Fig. B.7-B.8***), and since this pin is made of steel, we conclude that the stresses in this design will not appreciably affect the water pump's performance. Therefore, no edits to the pump geometry are needed.

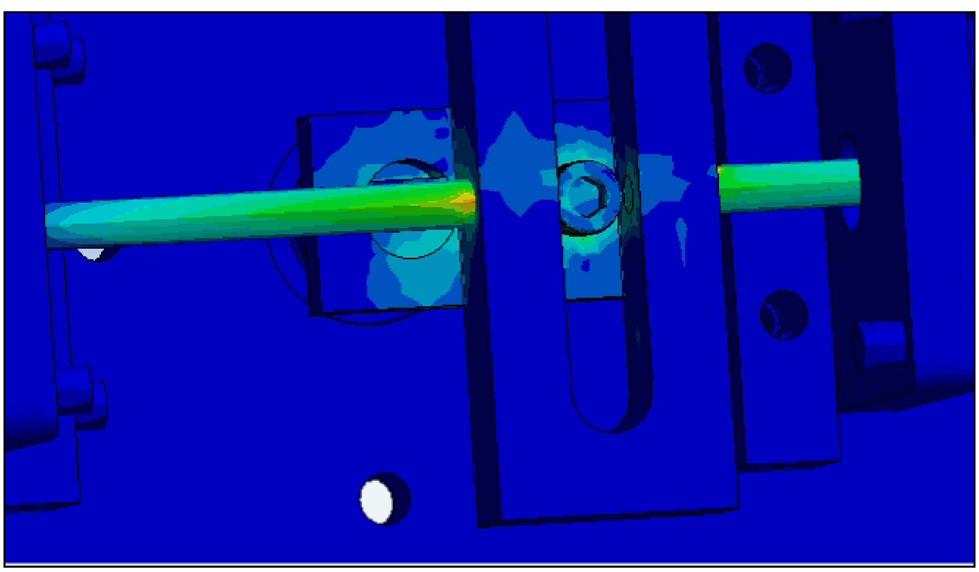


Figure B.8: *Stress Contours With Increasing RPM, Animation*

C CAD Images & Renders

C.1 Full Assembly

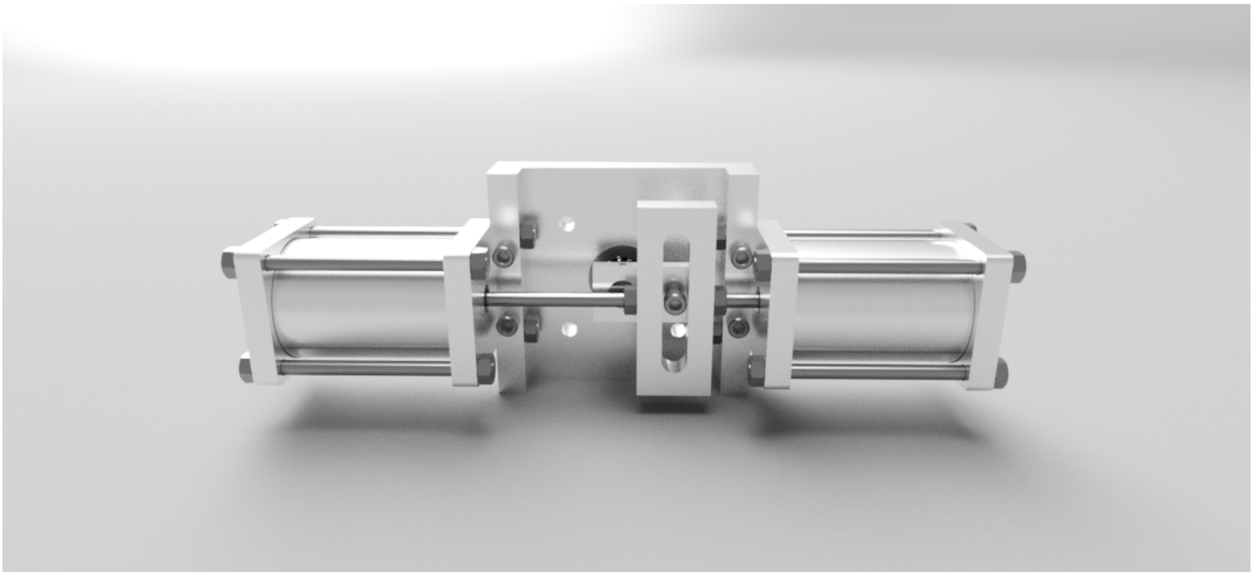


Figure C.1: Rendered CAD Model of The Full Assembly

C.2 Exploded Assembly

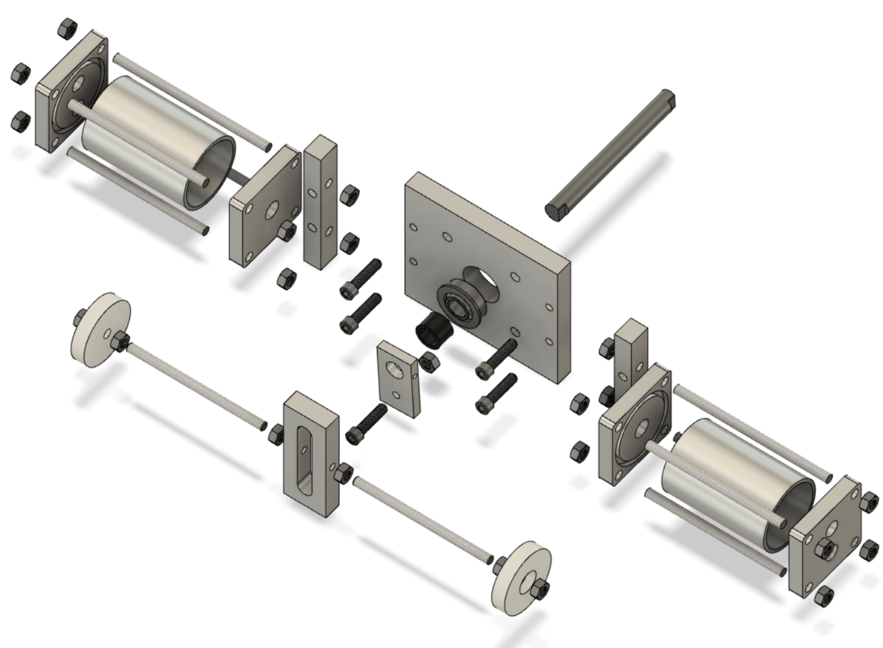


Figure C.2: Exploded View of The Full Assembly

C.3 Tolerancing

The tolerances on our part drawings were determined based on maximizing ease of manufacturability while still ensuring that the final pump design would have its full intended functionality. There were generally only two different scenarios in which we decided for a dimension to have a low tolerances (± 0.001 or ± 0.005):

1. *The feature interfaced tightly with another component*; for example, the central hole on the face plate interfacing tightly with a bearing for our drive shaft.
2. *The feature required a precise position relative to another feature*; for example, the two sides of the central slot on the scotch-yoke slot bar must have a precise width such that it can fit the scotch-yoke pin without leaving excessive clearance during actuation.

The drawings were developed such that each dimension was made with respect to a common datum, avoiding tolerance stack up when machining each part. For example, the datum for the drive shaft is the left face of the shaft, while the datum for the face plate is the bottom left corner of the face. Additionally, in cases where a threaded rod would be fastened with hex nuts on either side of a face, it was necessary to ensure that the rod would fit through the through-hole. However, it was not required for the walls of the through-hole to make good contact with the rods—the hex nuts would ensure a good fit either way. Thus, in drawings such as the central hole of the piston head or the through-holes of the cylinder supports we gave loose upper limits (+0.01) and tight lower limits (-0.001 or -0.005).

D Fabrication Plan

D.1 Parts List

New Part #	Component	Assembly	Material	Stock Used	Supplier	Link	Unit Price		Units	Amt		Total Price	Need to machine?
001	1/4-20 Nuts	Main	Steel	1/4 - 20 hex nuts	Emerson	https://ca	$0.06	per	piece	22	piece	$1.32	☐
002	Screws to mount face plate to test setup	Main	Steel	Are these provided?	Emerson			per	piece		piece		☑
003	Screws to mount cylinder assemblies	Main	Steel	Screws: 1/4-20 by 1"	Emerson	https://ca	$0.17	per	piece	4	piece	$0.68	☐
101	Face Plate	Frame	Aluminum	1/2" x 4" aluminum bar	Emerson	https://ca	$1.18	per	in	5	in	$5.90	☑
102	Bearing	Frame	Steel	Flanged, Open with Ring, 1/2" Shaft, 1-1/16" Housi	McMaster	https://wv	$10.66	per	piece	1	piece	$10.66	☐
102	Bearing	Frame	Steel	Flanged, Open with Ring, 3/8" Shaft, 1-1/16" Housi	McMaster	https://wv	$7.31	per	piece	1	piece	$7.31	☐
201	Cylinder Endcap (One Hole)	Cylinder	Aluminum	Cyl Endcap 1-Hole	Emerson	https://ca	$1.00	per	piece	2	piece	$2.00	☐
202	Cylinder Endcap (Two Holes)	Cylinder	Aluminum	Cyl Endcap 2-Hole	Emerson	https://ca	$1.00	per	piece	2	piece	$2.00	☐
203	Cylinder	Cylinder	Aluminum	Emerson Part	Emerson	https://ca	$1.00	per	piece	2	piece	$2.00	☐
205	Threaded Rod (to hold cylinders together)	Cylinder	Steel	1/4 - 20 threaded rod	Emerson	https://ca	$1.02	per	foot	3	foot	$3.06	☐
206	Tube Fittings	Cylinder	Plastic	Nylon pipe fittings (3/8" barbed x 1/4"NPT)	Emerson	https://ca	$0.55	per	piece	4	piece	$2.20	☐
207	Cylinder Mount Block	Cylinder	Aluminum	1/2" x 4" x 1"(max length 10") (base)	Emerson	https://ca	$1.18	per	inch	2	piece	$2.36	☑
301	Drive Shaft	Rotating Parts	Aluminum	1/2 " Diameter Aluminum Rod	Emerson	https://ca	$0.23	per	in	6	in	$1.38	☑
302	Scotch Yoke Arm	Rotating Parts	Aluminum	1/4" x 1" Aluminum Bar	Emerson		$0.14	per	in	2	in	$0.28	☑
303	Scotch Yoke Pin (screw)	Rotating Parts	Steel	1/4-20 by 1/2" screw	Emerson	https://ca	$0.15	per	piece	1	piece	$0.15	☐
304	Set Screw for Scotch Yoke Arm	Rotating Parts	Aluminum	Stainless Steel 8-32 set screw, 5/16" long	McMaster	https://wv	$2.77	per	25-pack	0.04	25-pack	$0.11	☐
305	Bushing	Rotating Parts	Bronze	1/2" ID x 5/8" OD x 1/2" depth lubricated bronze bu	Emerson		$0.60	per	piece	1	piece	$0.60	☐
401	Piston Arm	Sliding Parts	Steel	1/4 - 20 threaded rod	Emerson	https://ca	$1.02	per	foot	0.333333	foot	$0.34	☑
402	Piston Head	Sliding Parts	PVC	1 7/8" Diameter Plastic Rod (piston) (max length 9")	Emerson	https://ca	$0.10	per	in	1	in	$0.10	☑
403	Scotch Yoke Slot Bar	Sliding Parts	Aluminum	Aluminum Bar 1/2" x 2 1/4" (max length 12")	Emerson	https://ca	$0.73	per	in	3.25	in	$2.37	☑

Figure D.1: *Water Pump Bill of Materials*

D.2 Manufacturing & Ordering Analysis

We were able to get most of our materials from Emerson, which was ideal for our budget because everything was much cheaper. Since our design contained double piston-cylinders, we wanted to get a longer face plate to secure the assembly. The maximum length stock from Emerson was 10” and we would've needed 13”. However, upon searching through the catalog on McMaster, our group found that the stock was only available in 12” or 24” sizes; thus, both would result in maxing out our available budget for McMaster. We decided that we could use the 10” stock and connect the outer edges of our cylinder assemblies to the outer edge of the face plate stock with a modified mount block, as discussed in ***Sec. III.3***. The length of the faceplate would cause our pump to hit the pivot boss in the test setup, so we planned to angle our pump slightly to avoid this. However, after presenting our final design presentation, we were informed by our TAs that only including the inner mount blocks would be more than enough to secure the cylinder assemblies. This allowed us to use an even smaller faceplate only in the very center near the drive shaft and to switch back to the horizontal orientation. We were able to get this stock from Emerson, which saved money in our McMaster budget, and also saved us machining time. This was particularly helpful, so we could spend more money on our set screw and bearing from McMaster, with extra room in the budget to get a different size bearing later in the process after testing.

D.3 Cost Analysis

Our total material cost comes to a total of $44.82. Of this total price, 33% ($14.58) comes from the cylinder assembly. This fraction is approximately evenly split between the components in that assembly. 11% of the total cost ($5.01) comes from the Scotch Yoke Assembly. The majority of this is due to the Scotch Yoke slot bar, which costs $2.37. 1.5% of the total price($0.68), is due to the Piston Assembly. Finally, 55% ($24.55) comes from the Frame Assembly. This is primarily due to the two ball bearings we purchased. The first cost $7.31 and was found to be too small. The second costs $10.66 and is larger to fit the drive shaft. These bearings alone contribute 40% of the total price of materials for the pump. While this is on the expensive side for the total material cost, it ends up making very little difference towards the prototype cost, contributing only 0.5%. The material cost of subsequent pumps would be 16% cheaper as there would be no need to buy two bearings.

Total prototype cost: $3724.82
Non-Recurring Engineering Design Hours = n = 24hrs
Prototype manufacturing hours = m = 20hrs
Material costs = c = $44.82

$$(\$/pump) = n * \$120/hr. + m * \$40/hr. + c$$

$$(\$/pump) = 24hr * \$120/hr. + 20hr * \$40/hr. + \$44.82 = \$3724.82$$

Product Cost, Single Production Pump; if only one sold: $3835.2
($/pump) = Total prototype cost + (# holes + # threadings + # reamings + # milled flat surfaces + # turned straight surfaces + 10* # curved turned or milled surfaces + # inches of weld + # cuts + # hand finished edges or surfaces + # bends)*$1.20

$$(\$/pump) = \$3724.82 + (22 + 8 + 14 + 10 + 3 + 3 + 10 * 0 + 0 + 12 + 20 + 0) * \$1.20$$
$$(\$/pump) = \$3835.20$$

Product Cost per pump; if can sell 1000 pumps: $114.12
($/pump) = Total prototype cost/1000 + (# holes + # threadings + # reamings + # milled flat surfaces + # turned straight surfaces + 10*# curved turned or milled surfaces + # inches of weld + # cuts + # hand finished edges or surfaces + # bends)*$1.20

$$(\$/pump) = \frac{\$3724.82}{1000} + (22 + 8 + 14 + 10 + 3 + 3 + 10 * 0 + 12 + 20 + 0) * \$1.20$$
$$(\$/pump) = \$114.12$$

D.4 Fabrication Overview

Fabrication Choices
Our piston-pump design was conceptualized with the intent to not require many machined parts in order to ensure all of our members could machine the parts with our current skills, and to give the team plenty of time to assemble and potentially re-machine any parts that encountered complications. We decided to use 6061 aluminum stock for most of our machined parts because we thought it was the best balance between strength and manufacturability, especially given the low strength requirements determined from the stress analysis in ***Sec. V.2*** . Similarly, we used steel threaded rods because they did not require much machining and were readily available in Emerson. We used PVC for the piston heads because they provide a much smaller friction coefficient against the aluminum cylinders, without requiring lubrication or oil, thus saving room in our budget. Throughout our design process, we made decisions about our parts by heeding the principles of DFM/DFA (Design for Manufacturing/Assembly). This manifested via multiple design reviews aimed at assessing and formulating a machining plan for each part. We decided to use tapped holes/threaded rods for most connections instead of sharp 90-degree angles or irregular shapes in the aluminum. We used the mill for parts that needed to be squared up/cut down to size and required thru-holes or tapped holes. We used the lathe for the PVC piston heads because of their cylindrical shape. The lathe and mill provided us with high accuracy parts that were necessary for our tight tolerancing while we used the bandsaw to cut threaded rods because the dimensions did not require high accuracy, and we were able to bevel the ends with the belt sander to restore the threading.

Fabrication Complications:
We decided to schedule most of our machine shop slots as early as possible to give our group plenty of time to test the integration of all our parts. This turned out to be particularly helpful because there were some complications during machining. For the face plate, we forgot to adjust

the position of the edge finder to center the tool when drilling the holes. In addition, when threading one of the holes for the face plate, the tap broke and got stuck in the hole. We had to redo the face plate to fix these issues. We also had some trouble assembling our parts. Two of the holes in the cylinder mount blocks on perpendicular faces were too close together, so the screws were interfering with the threaded rods when trying to fit into the holes. Some of the holes had to be cleaned out or rethreaded. After some minor adjustments, everything fit together smoothly. We also had to redo our piston heads because they were originally cut on the bandsaw, which caused one of the two pistonheads to have both faces not perfectly flat, rendering them impossible to hold with the lathe chuck. One of the pistonheads therefore had to be redone using extra stock allowed for by our prudent budget considerations (*See:* ***Sec. III, Sec. VII.1***).

Part	Parts List #	Quantity	Machine	Process notes	Manufacturer
Piston head	402	2	Lathe	Lathe: Drill through-hole Use boring bar to create counterbore Reduce radius to size	Sean
Piston arm	401	2	Bandsaw, belt sander	Use bandsaw to cut to size Use belt sander to bevel ends	Steph, Julianna, Maia
Cylinder structural rods	205	8	Bandsaw, belt sander	Use bandsaw to cut to size Use belt sander to bevel ends	Steph, Julianna, Maia
Drive shaft	301	1	Bandsaw, lathe, mill	Use bandsaw to cut to size Use lathe to correct length Cut flats on mill using collet blocks	Sean
Scotch yoke slot bar	403	1	Mill	Use Bandsaw to cut to size Mill stock to size (3 faces) Use 3/8" end mill for slot	Isaac
Scotch yoke arm	302	1	Bandsaw, mill	Use bandsaw to cut to size Mill stock to size Drill 3 holes Tap 2 holes	Callen
Face plate	101	1	Mill	Drill 9 holes Tap 4 holes	Steph
Cylinder mount blocks	207	2	Mill	Mill stock to size Drill 4 holes each	Julianna

Figure D.2: *Fabrication Overview*

D.5 Fabrication Timeline

Task	Member	Start	End	Progress		Estimated Time (hrs)	Actual Time (hrs)	Material	Machining Notes
WEEK 1 (4/27-4/30)									
Part Ordering	Maia	4/27	4/27	100%		2	2	N/A	N/A
Skotch Yoke Slot Bar (1)	Isaac	4/29	4/29	100%		2	3.5	403	Mill
Skotch Yoke Arm (1)	Callen	4/29	4/29	100%		0.5	2	302	Mill
Piston Arm (2)	Julianna	4/29	4/29	100%		0.5	0.5	401	Bandsaw
Drive Shaft (1)	Sean	4/30	4/30	100%		1	1	301	Lathe/mill
Face Plate (1)	Steph/Callen	4/29	4/29	100%		2	2	101	Mill
Cylinder Mount Block (Inner)	Julianna	4/29	4/29	100%		3	3	207	Mill
WEEK 2 (5/3-5/8)									
Piston Head (2)	Sean	5/4	5/4	100%		1	2	402	Lathe
Threaded rod (8)	Julianna/Maia	5/3	5/5	100%		1	0.5	205	Hacksaw/belt sander

Figure D.3: *Fabrication Timeline (link)*

E Performance Analysis

In one minute, our pump pumped 2.8 liters of water, which is respectable but not remarkable. Living up to our team name, the pump leaked profusely during operation—water squirted out of the holes for the piston arms when the pistons retracted. Somewhat-loose tolerances between the piston heads and the cylinders allowed water to pass to the wrong side of the heads, which likely decreased the pump's overall output in two ways: first, some of the water that entered each cylinder failed to leave via the output tube, and second, the water on the wrong side of each piston head had to be forced out the other side, which likely consumed power and slightly decreased the operating speed. Aside from the leakage around the piston heads, the pump was watertight, sturdy and quiet.

However, many pumps performed better than ours. They:

- Used more cylinders. Thursday Group 2 had three cylinders and pumped 5.3 liters, while Wednesday Group 2 had four cylinders and pumped 8.6 liters.
- Leaked less. Even among dual-cylinder scotch-yoke pumps (the most common design), some pumps performed better, at around 4 liters per minute. The piston travel lengths were relatively consistent across pumps, but ours was particularly leaky.

In our design process, we made assumptions and compromises that contributed to our pump's performance. First, we selected a two-cylinder design. This decision limited our pumping rate significantly—the number of pistons has a powerful influence on pump performance. However, since we did not know exactly how limiting it would be, we decided to choose the simpler design so we could execute it well. Second, we prioritized low-friction sliding over tight-tolerance sealing between the piston heads and the cylinders. This gave us a leakier pump than we would have liked.

F Appendix

F.1 Sketches

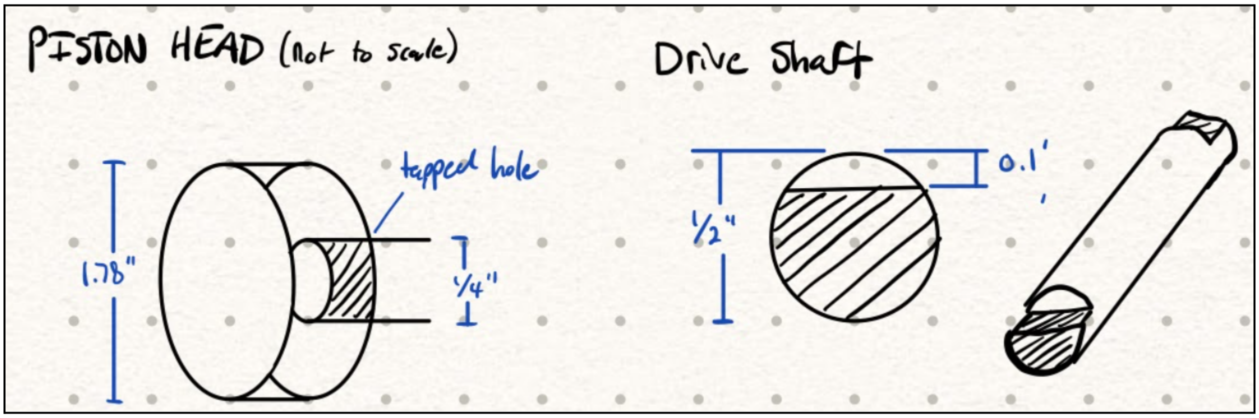


Figure F.1: *Drive Shaft Preliminary Design Sketch*

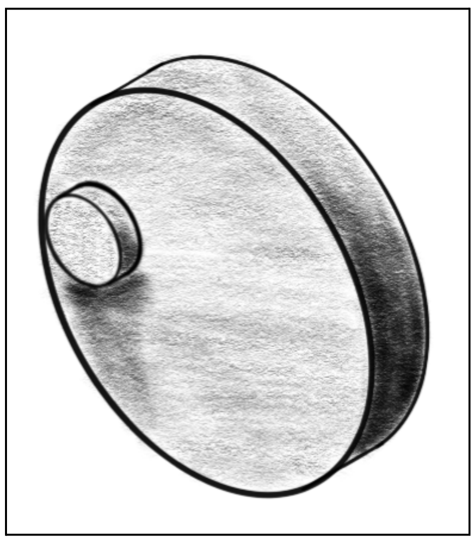


Figure F.2: *Original Scotch Yoke Pin and Circle Design Sketch*

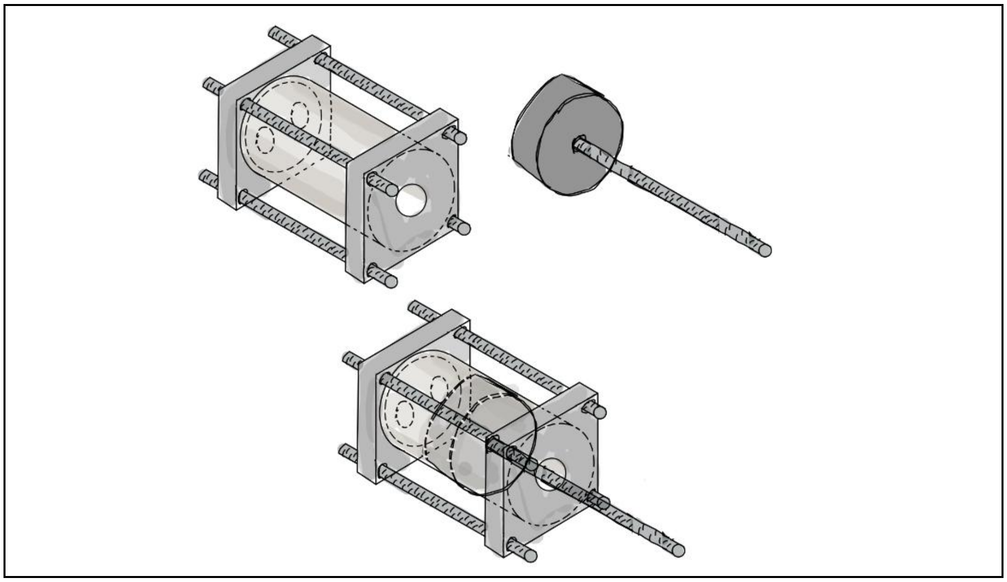


Figure F.3: *Piston-Cylinder Assembly Sketch*

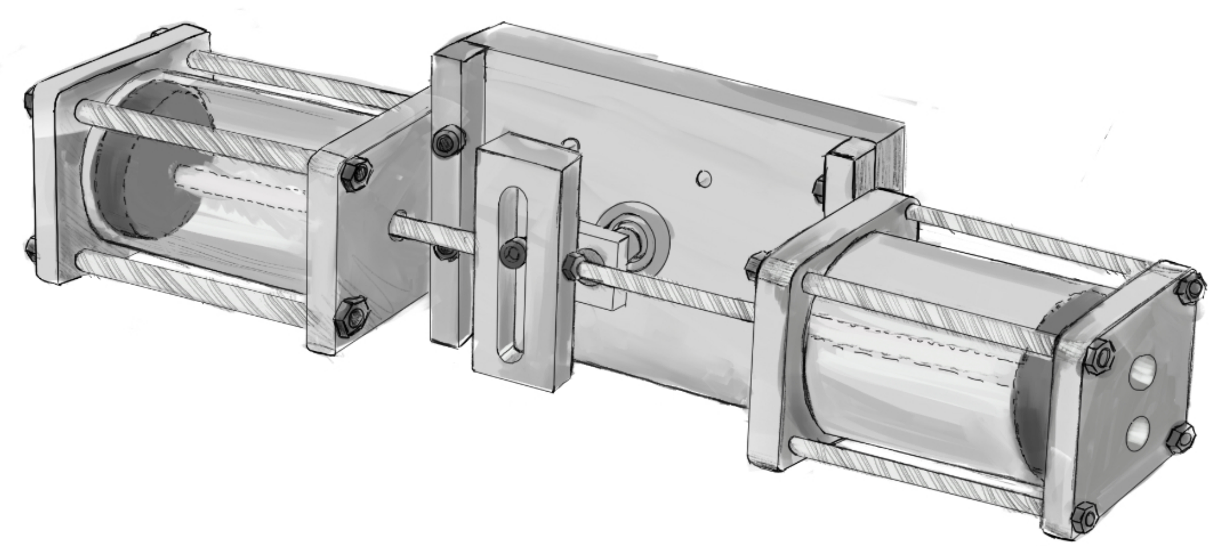


Figure F.4:: *Full Assembly Sketch*

F.2 Part Drawings

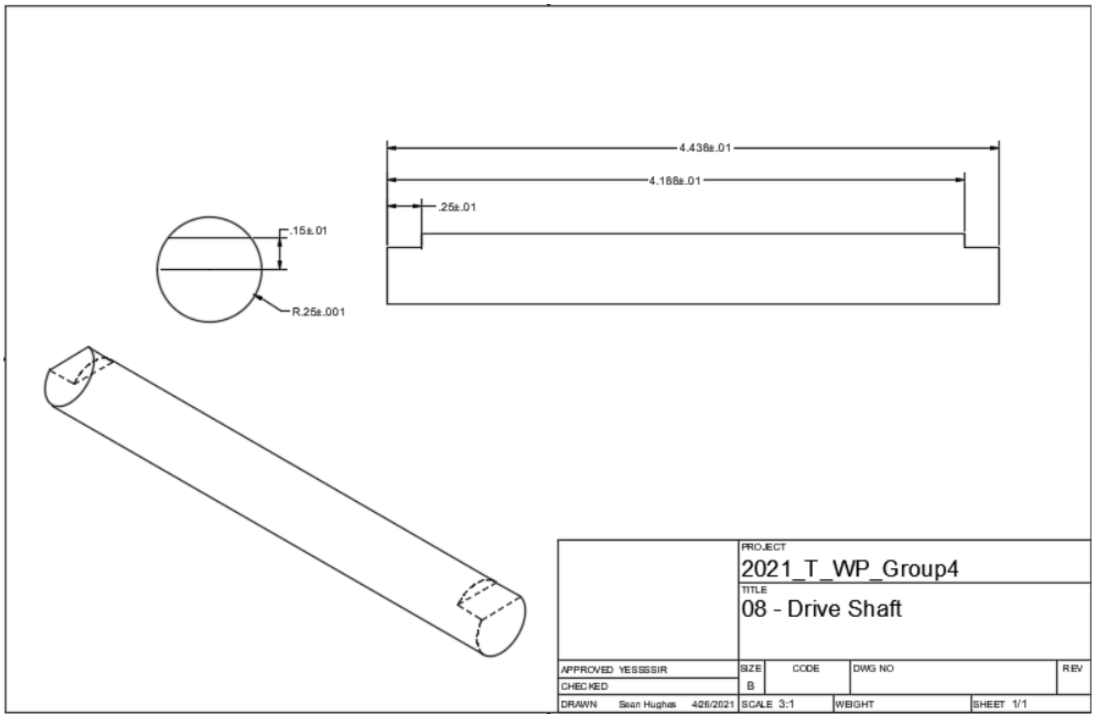


Figure F.5: *Drive Shaft Drawing*

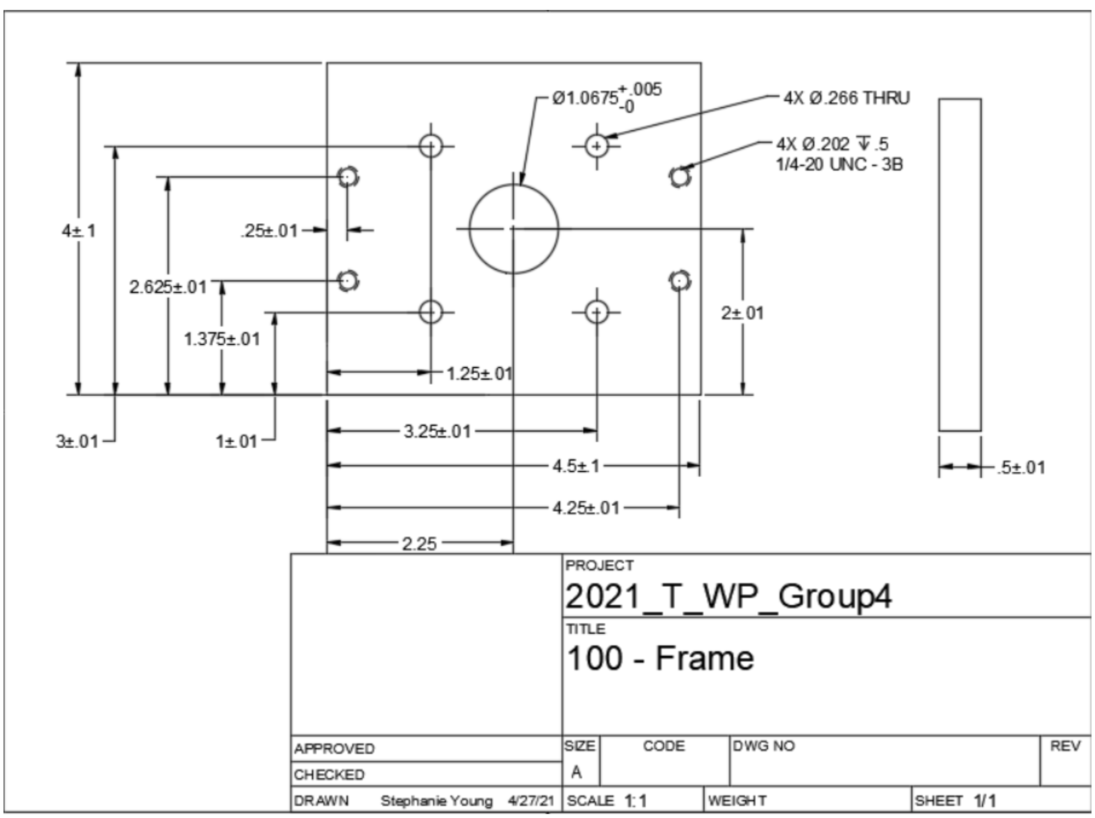


Figure F.6: *Face Plate Drawing*

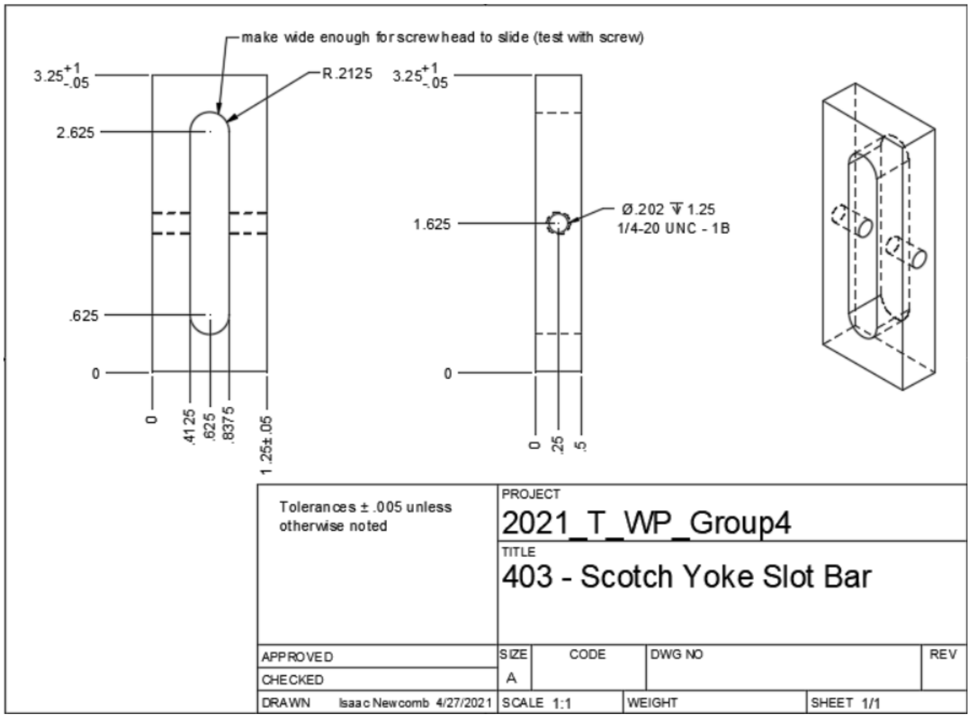


Figure F.7: *Scotch-Yoke Slot Bar Drawing*

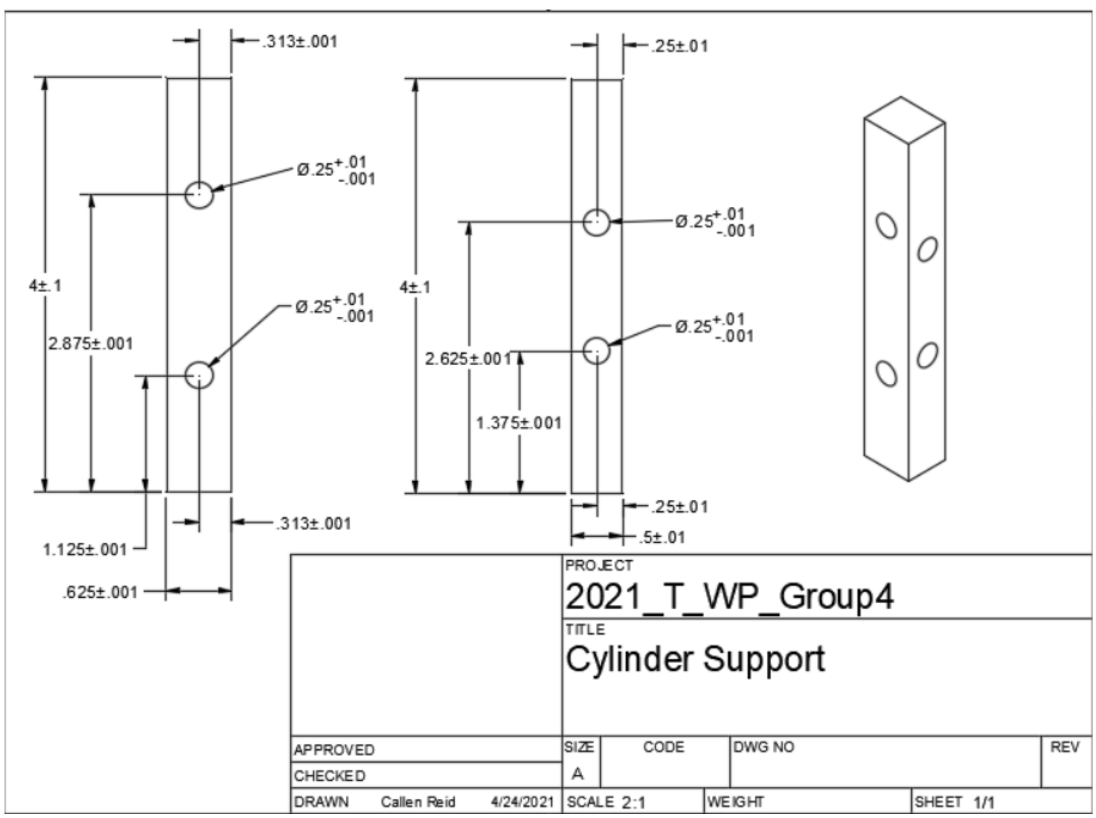


Figure F.8: *Cylinder Support Drawing*

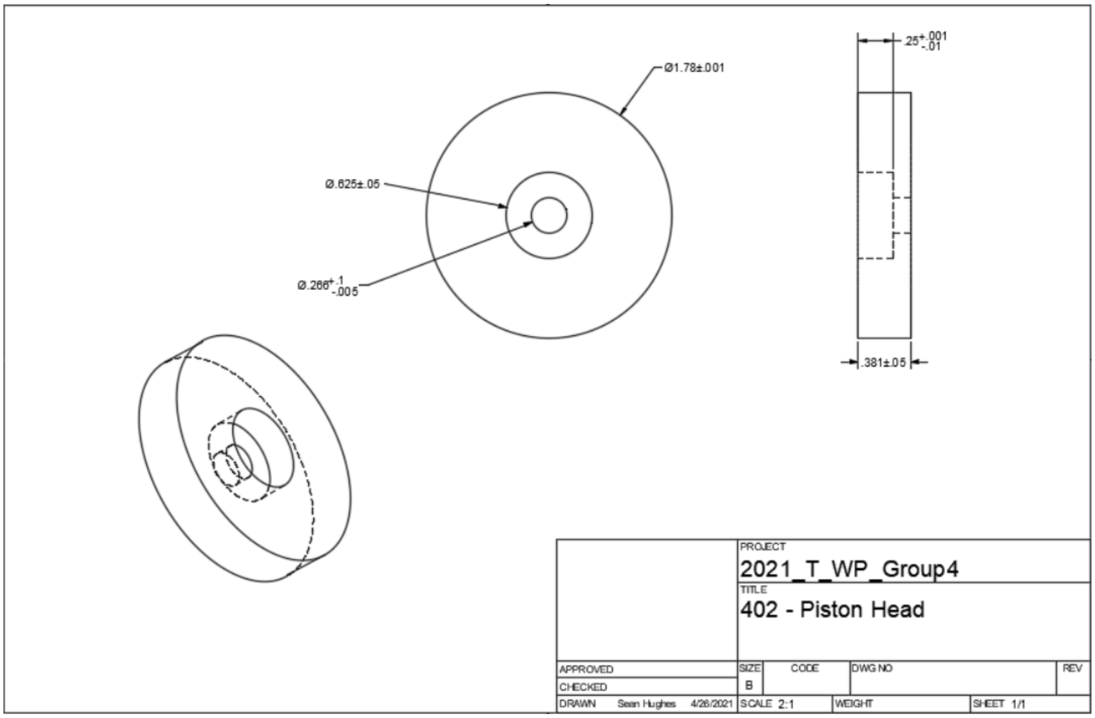


Figure F.9: *Piston Head Drawing*

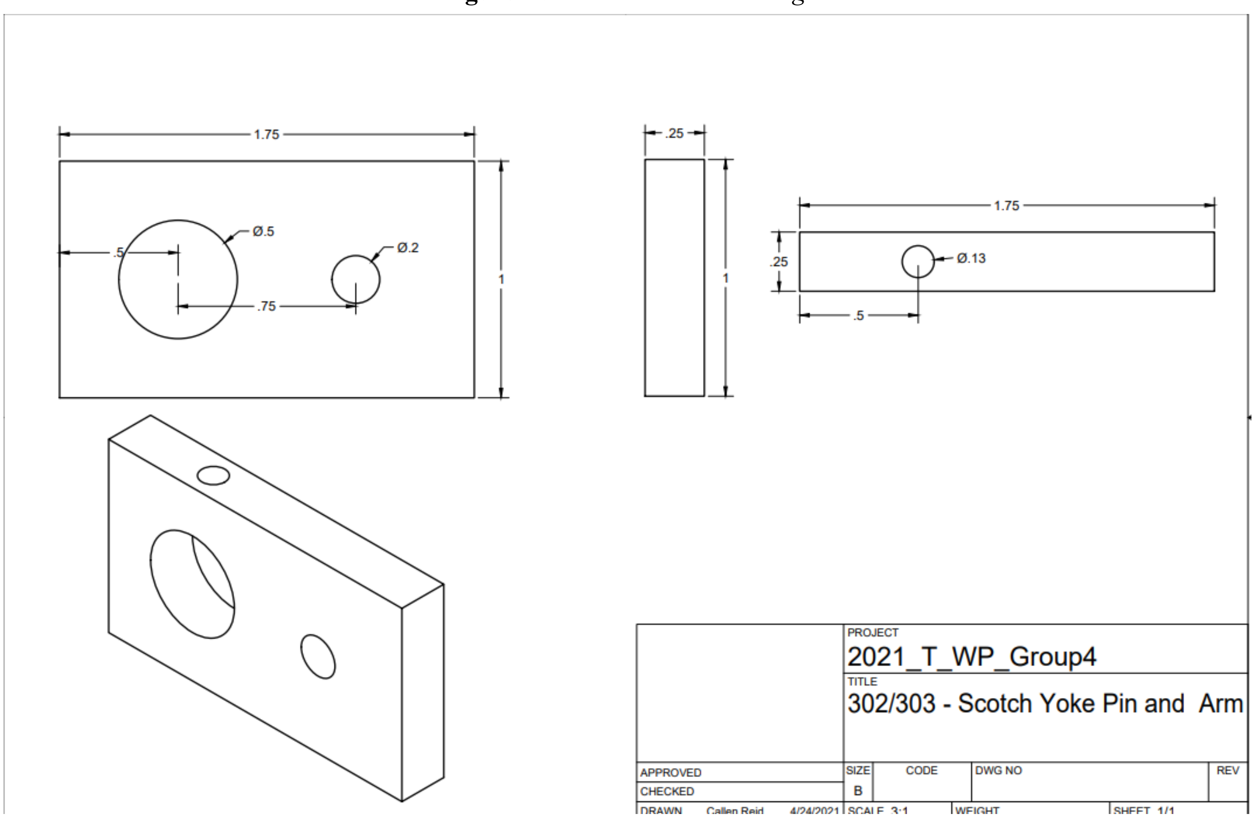


Figure F.10: *Scotch-Yoke Pin and Arm Drawing*

F.3 Gantt Chart

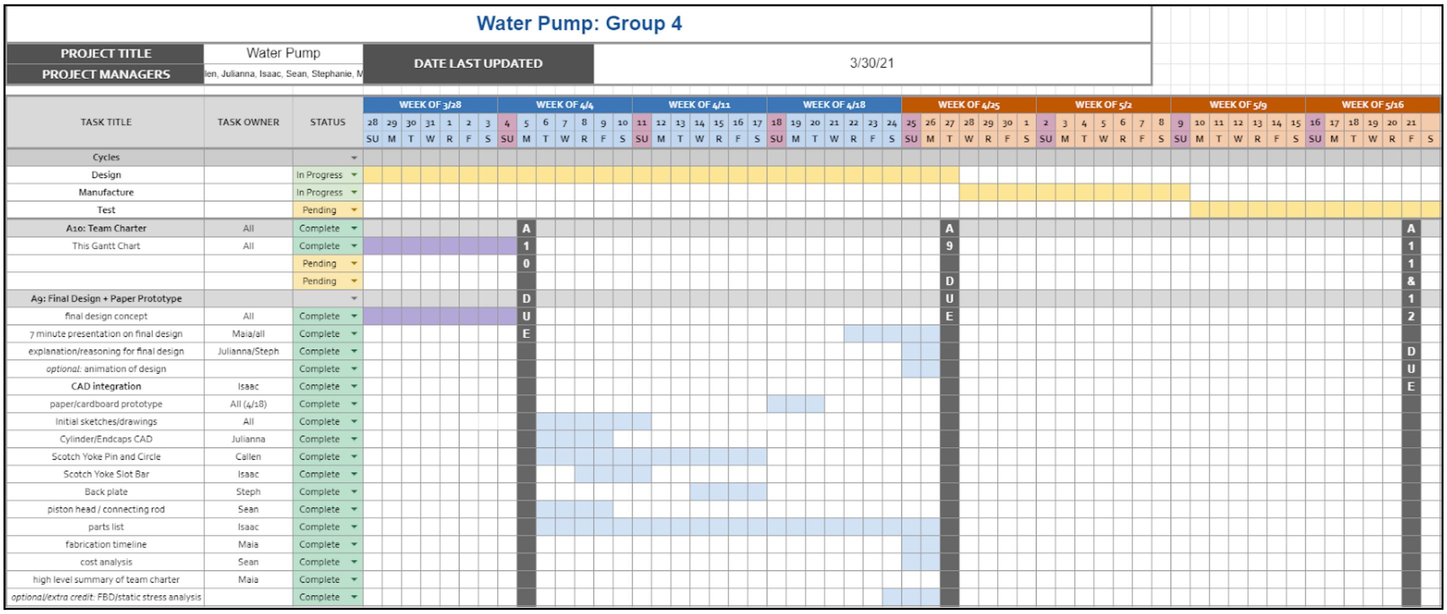


Figure F.11: *Water Pump Gantt Chart (link)*

F.4 Equations

$$(900\ RPM)(\frac{9}{70})(\frac{1\ min}{60\ s}) = 1.9\ Hz \quad \textbf{Eq. (1)}$$

$$(4.5\ lb \cdot ft)(4.44822\ N/lb)(12\ in/ft)\ /\ (0.5\ in) = 3700\ N \quad \textbf{Eq. (2)}$$

F.5 References

[1]

“Overview of materials for PVC, Extruded,” *MatWeb.com*. http://www.matweb.com/search/DataSheet.aspx?MatGUID=bb6e739c553d4a34b199f0185e92f6f7 (accessed May 12, 2021).

[2]

“Aluminum 6061-T6; 6061-T651,” *MatWeb.com*. http://www.matweb.com/search/DataSheet.aspx?MatGUID=b8d536e0b9b54bd7b69e4124d8f1d20a&ckck=1 (accessed May 12, 2021).

[3]

“AISI 1012 Steel, cold drawn,” *MatWeb.com*. http://www.matweb.com/search/DataSheet.aspx?MatGUID=f1b9fbe9b7874617992ef04445a29ffd&ckck=1 (accessed May 12, 2021).