Machine Design of a Sun Tracking Solar Panel

Team 2

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1 Introduction

1.1 Background

The International Energy Agency predicts that the demand for global energy will increase by approximately 30% between the years 2016 and 2040, with 40% of the increase coming from electricity alone [1]. This creates a challenge to meet the growing energy demand without compromising the health of the planet.

One recent innovation that attempts to help meet the increased demand for energy sustainably is that of the photovoltaic panel (PV), commonly known as the solar panel. Such devices generate electricity for consumers using radiation energy from the sun. One 2015 study performed on solar panels in Austin, Texas, showed that "on average, solar panels avoid approximately 75% of yearly grid-related emissions and yearly grid-related water consumption" [2]. This shows that photovoltaic panels can plausibly assist in meeting energy demand in an environmentally sustainable way.

This same study, however, found that "the benefits [of solar panels] depend on the orientation of the panels" relative to the sun [2]. This is supported by a 2018 study which found that operational solar panel efficiency is approximately between 15% and 22.5% while research panels are about 46%, and that this efficiency is highly dependent on the cell orientation relative to the solar radiation field [3]. These findings create a challenge for environmentally interested designers; to better meet energy demand sustainably, solar panels can be equipped to orient themselves towards the solar radiation field automatically.

Overall, it has been established that to best utilize the technology of photovoltaic cells to meet growing energy demand, solutions must be developed which can orient the cells relative to the sun.

1.2 Motivation

The inspiration for this project was the work of Pavilion Renewables, a company which hosted one of the team's members for an internship. The company, based in Bahrain, aims to achieve water and energy security for current and future generations without sacrificing the planet's current health. Pavillion Renewables will serve as the client for this project.

The mission of Pavilion Renewables, combined with the existing knowledge gap surrounding the dependency of solar panel technology's efficiency on orientation, serves as the motivation for this project. By creating a mechanical solution that allows solar panels to reach higher efficiencies, Team 2 will achieve the MECH 325 course learning objectives while simultaneously generating a feasible concept for achieving future energy security.

1.3 Scope

The purpose of this project is to outline a feasible machine design which can accomplish the design challenge outlined in Section 1.1. This will be done while simultaneously accomplishing the course objectives of MECH 325. The pilot site for the mounting of this solar panel will be designated as a vertical-axis wind turbine in Bahrain. This pilot site selection is motivated by the client. The intent is that the design outlined in this paper could be mounted onto the vertical-axis wind turbines in order to generate additional power.

Due to time constraints and technical readiness of the team, some components of the final design were deemed beyond the scope of this project. The team will focus on specifying components that we have learnt to analyze through the MECH 325 course. The design of both the photovoltaic cell and the vertical-axis wind turbine are deemed outside the scope of this project. They will be taken as inputs to the design process. Similarly, the control systems of the device are deemed outside the scope of this project as such work lies outside the realm of the designers' expertise. Supplemental components, such as the frame used to mount the solar panel to the vertical-axis wind turbine will also be deemed outside the scope of this project as their in-depth analysis is not relevant to course objectives. Finally, the build and implementation stages of the device lie outside the scope of this project; this paper only outlines the intended design of the solution.

It should also be noted that the key resources used for the mechanical design of this system are the following:

- *Shigley's Mechanical Engineering Design (Eleventh Edition)* by Richard G. Budynas and J. Keith Nisbett
- *Machine Elements in Mechanical Design (Sixth Edition)* by Robert L. Mott, Edward M. Vavrek, and Jyhwen Wang
- *Fluid Power Basics* by Parker Training
- *Design Engineers Handbook* by Parker Training.

Thus, assumptions, simplifications and conventions from these resources may carry forward into the team's detailed design.

2 Key Project Aspects

2.1 Design Needs

Due to the time constraints surrounding the project and the focus of the course objectives, the team chose not to pursue in-depth stakeholder engagement. Instead, knowledge of Pavillion Renewables, the pilot site, the application area and the existing infrastructure were used to develop the needs for the project. They are shown in Appendix A along with their justifications.

2.2 Design Requirements and Evaluation Criteria

Once the needs for the project were developed, requirements and evaluation criteria were extracted. They are shown in Tables 1 and 2. The relative weightings of the evaluation criteria are shown in Table 2 as well, again based on knowledge of Pavillion Renewables, the pilot site, the application area and the existing infrastructure. These will be used during concept evaluation and selection in Section 2.3.

Table 1: Design requirements.

Design Requirement	**Justification of Requirement**
The design accepts 120V, 3-Phase AC power.	Power input (from client)
The design withstands winds up to 36.72 km/hr.	[4]
The design has a minimum design factor of 4.0.	Due to the dynamic loading situation posed by the environment (wind loading), the uncertainty surrounding these loads, and the reliance upon the electric power generated by the system, page 189 of the Mott textbook determines the minimum design factor to be 4.0 [7].
The design has 45 degrees of rotation on pitch and 120 degrees of rotation on the yaw axes.	Problem definition
The design withstands temperatures from 10 to 50 degrees Celsius.	[5]
The design is able to actuate 521.56 lbf.	Loading analysis (See Appendix B)
The design is made of a UV resistant material, or incorporates a method to protect against UV rays.	Problem definition
The service lifetime is at least 6 years.	Solar panel lifetime (from client)

2.3 Concept Generation, Evaluation and Selection

An in-depth concept generation, evaluation and selection process was not pursued due to time constraints and the nature of the course objectives. Instead, each team member generated 1-2 feasible concepts and a combination Weighted Decision Matrix (WDM) and Pugh Chart was utilized for concept screening and scoring. The evaluation criteria in Table 2 were used for this process, and the detailed combination WDM/Pugh Chart is outlined in Appendix G.

Table 2: Evaluation criteria.

Evaluation Criteria	Weight	Justification
Cost	25%	Since the client is a business, they are most likely to pursue a solution with the highest cost-benefit ratio. While the other evaluation criteria consider the benefit of the solution, the cost is relatively important to weigh down high cost solutions.
Simplicity	20%	Due to the technical readiness of the team and the time constraints of the project, a simpler project is more achievable and thus more likely to satisfy the client.
Manufacturability	15%	The easier a solution is to manufacture, the easier it would be for the client to adopt it into their system.
Reliability	15%	Since these systems would be relied upon to generate electricity for critical infrastructure, it is important that they remain reliable.
Predicted Mass	10%	As the solution is to be mounted above ground, concepts with lower predicted masses would make this process easier.
Predicted Compactness	10%	As the solution is to be mounted above ground, concepts with higher predicted compactness would most likely be easier to mount.
Creativity	5%	The team chose to give a small percentage weighting to more creative solutions as, all else equal, a novel idea may better fill a market gap. However, the other criteria are more predictive of a successful solution and thus the weighting of the evaluation criteria is low.

The final selected concept, called “Center Pivot + LinAx” is shown in Figure 1. It works primarily by using hydraulic cylinders to create linear motion and using bushings to facilitate rotation of the solar panel. This concept is explained in more detail in Section 2.4.

2.4 System Overview and Key Components

There are three primary motion mechanisms at play: two hydraulic cylinders, sets of two bushings in a cross formation to serve a similar purpose as a ball joint and a shaft immediately behind the solar panel with two bushings on it in order to facilitate sliding of the cylinders along the shaft. The combination of these three motion mechanisms gave the system the desired degrees of freedom. Figure 2 shows the overall flow of energy through the system to better illustrate the process.

This concept gave the team the following three key components to analyze with respect to the MECH 325 course objectives:

1. The Hydraulics
2. The Bushings
3. The Shaft

The system controls two directions in order to receive optimal solar energy. The selected concept function works by controlling the azimuth angle through the difference in length between the linear actuators and the elevation through the average length of the pistons. This function is not a linear function but there is a way to translate the target normal vector to piston length that is outside the scope.

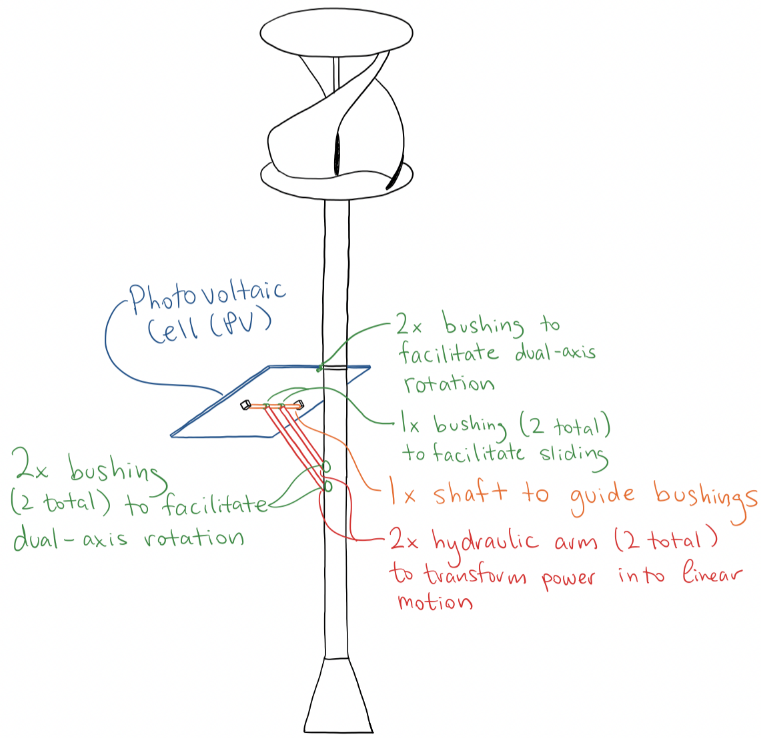


Figure 1: Final concept selected.

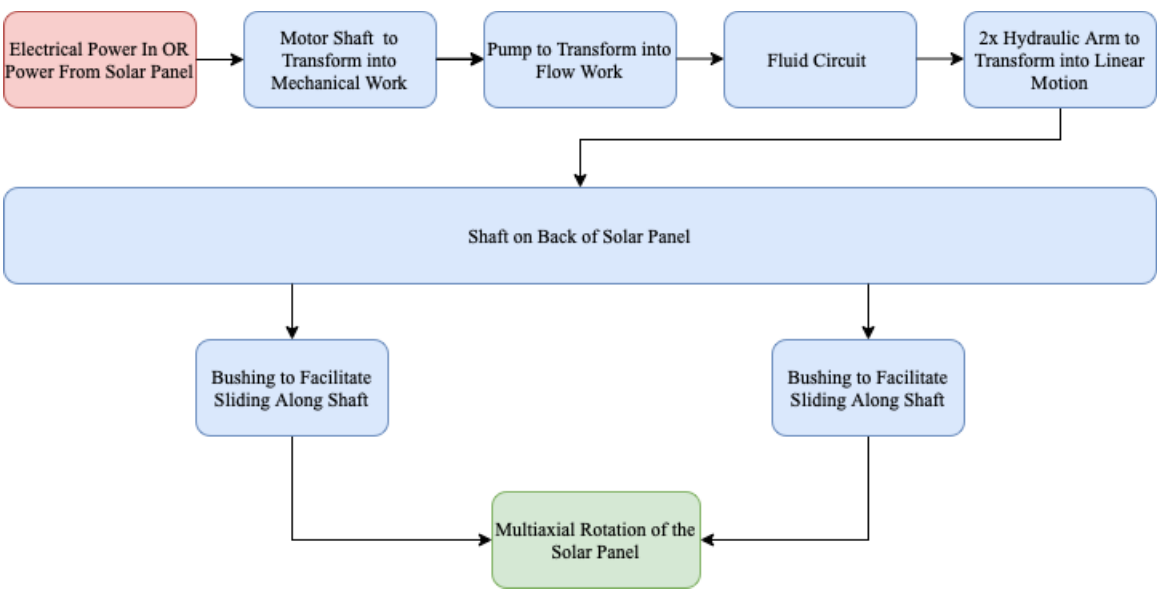


Figure 2: System overview diagram.

2.5 Key Assumptions and Simplifications

In order to begin analyzing the system, the team first had to make several key assumptions and simplifications to make the detailed design process feasible. They are shown in Table 3.

Table 3: Key assumptions and simplifications.

Key Assumption or Simplification	**Implications for Analysis**
The solution uses existing power infrastructure	The team will not need to pursue retrofitting or rewiring for their solution. There exists both backup power for the device from the existing power grid and there exists a solution to store any energy from the device.
The solution's components are shaded by solar panel	This is a reasonable assumption since the panel follows the sun along its path, and implies that there will be no solar heating experienced by the components.
The solution uses Pavillion's own mounting system	A mounting system would not need to be designed by the team.
The solution is to be designed for the worst case.	Any wind loads are assumed to be directly perpendicular to the panel, leading to the maximum drag force. These loads will also be calculated at the highest speed wind condition of 35km/h [4].
The wind loads experienced by the panel can be simplified.	The loading on the panel from the wind can be assumed to be evenly distributed across the panel. In addition, the vertical-axis wind turbine above the panel would normally have effects on the wind loading on the panel, but these effects will be ignored.

3 Detailed Analysis and Results

3.1 Hydraulics

To meet the load requirements presented above, hydraulic cylinders were specified according to Parker Hydraulics' design guidelines. PH graph and table references are prefixed with "PH" in this section, and are included in Appendix C. The constraints for the design included:

1. A body length of 55in and stroke length of 28in to achieve our full range of motion
2. A maximum force of 1050lbf per actuator
3. A safety factor of 4, according to Mott's recommendations

The results are summarised in Table 4 below.

Table 4: Hydraulic cylinder specifications.

Parameter	**Rating**	**Units**	**Source**
Stroke Length Factor	2	-	*PH Table B-4*
Rod Diameter	$1\frac{3}{8}$	in	*PH Graph B-1*
Bore Diameter	2	in	*PH Table B-1*
Operating Pressure	400	PSI	*PH Table B-1*
Operating Flow Rate	1	GPM	*PH Table B-8*

To achieve our required motion while minimizing bending loads on the cylinder, we selected a Case 6 mounting scheme according to PH Table B-4. Our effective length is thus 56in, double the nominal stroke length. To ensure the cylinder rod could bear this load, we consulted PH Graph B1 and found that, for this effective stroke length and loads up to 1050lbf, a $1\frac{3}{8}$in rod was sufficient. Given this rod size, the first compatible bore size was selected at 2in.

The blank side force for a 2in bore would need to exceed the rated force as specified above; equilibrium was computed to occur at 331 PSI, so the next standard pressure rating of 400 PSI was selected. The maximum force produced by the fluid on the rod end is quite low, but we recognized that the weight of the solar panel would allow us to retract the cylinders anyway. This phenomenon is illustrated in Figure 4. A cushion cap is fitted to the blank side to prevent damage during retraction as the solar panel applies a high load.

The solar panel does not need to operate at high speeds, so a rod velocity of 2.5 FPM is specified. By PH Graph B-2 it is observed that such a velocity does not drive cylinder design.

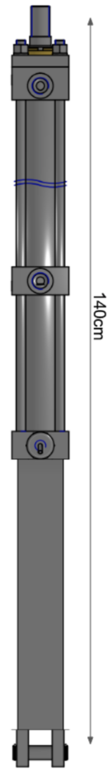


Figure 3: Hydraulic cylinder.

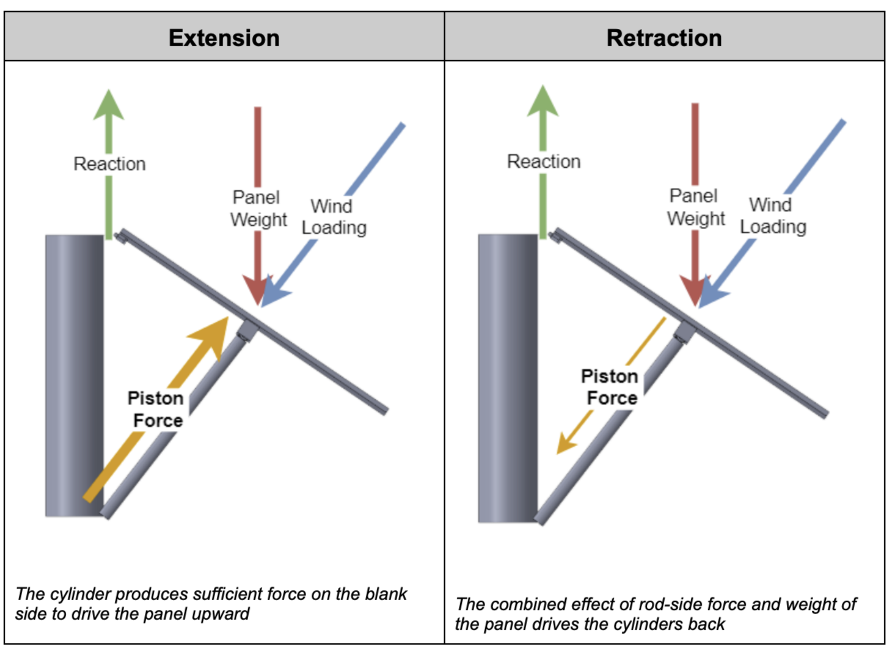


Figure 4: Piston forces during extension and retraction.

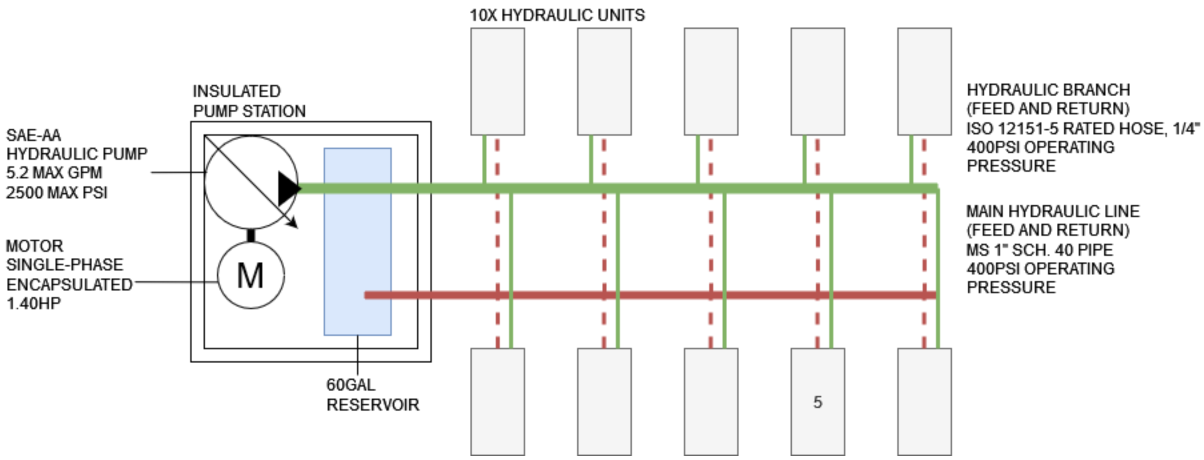


Figure 5: Network overview.

To achieve closed-loop controllability, an electrohydraulic cylinder with a built-in linear displacement transducer is specified. The cylinder operates in a dusty outdoor environment, so PTFE wiper-style rod seals are selected to prevent contamination.

The nominal retracted length for this cylinder is specified by Parker Hydraulics as being 45 inches, so a 10-inch extension is attached to the rod.

3.2 Hydraulic System Network

To simplify the system for mass operation, we have designed specifications for a network of ten units that connect to a central pump station which houses the motor, pump and reservoir.

Figure 5 shows the network overview.

Benefits of a network configuration include a reduced number of components; reduced component cost; a centralized location for maintenance; and focused sheltering for pump and motor from high temperatures and sand.

There are also some potential drawbacks of the system that can be mitigated. These risks are tabulated below, and should be considered in further design phases.

Table 5: Risks and mitigation methods for hydraulic system.

Risk	**Mitigation**
Total system failure due to pump malfunction	Secondary Pump
Total system failure due to leak in main feed line	Pipe, components, and joints are quality tested.
2[2]*Pipe failure due to ambient conditions	2[2]*Pipe is laid underground to protect against UV degradation and solar heating. Hydraulic hose is fed through the middle of the mast
Higher cost of piping and installation	Minimize piping distance

Table 6 below summarizes all the components required for the network. The motor and pump specifications are based on the flowrate of the system. This is determined by a piston velocity of 2.5fpm, which is based on the premise that a full extension of the piston would take a reasonable time of 30 seconds. The tank is based on the full fluid capacity of the system with a 10% contingency. The hose is simply rated by pressure, and a hose as thin as $\frac{1}{4}$in ID is feasible. The details for these analyses can be found in Appendix C.

Table 6: Hydraulics components.

Component Type	Component Features	Quantity
Pump	SAE-AA, 5.2 GPM, 2500PSI	1
Motor	Single Phase, 1.40hp	1
Tank	60gal, non-pressurized, non-reactive	1
Pipe	Zinc-Coated MS,1" Sch. 40	240m
Hose	ISO 12151- Rated, 3200 PSI @ 72F, 1/4" ID	60m
Hydraulic Cylinder	See previous section	20

Figure 8 below shows the fluid circuit diagram for one hydraulic (one unit) connected to the central pump and reservoir. Additional units would be installed in parallel.

Table 7 shows a list of components and their corresponding labels from Figure 8.

A few notable points from this circuit design is that the circuit operates in sequential order to keep the circuit simple and reduce the number of flow control valves that are required. Positioning will be hard coded to the control system, so the pump can be turned off between adjustments for reduced power consumption.A linear displacement transducer is installed by the manufacturer into the hydraulic cylinder to provide closed-loop feedback for the control system that will operate the valves.

3.3 Bushings

Six unique bushings are specified throughout the system. All bushings are Oiles 500 SP types, selected for their high PV value. Such high-strength bushings enable us to specify smaller diameters while supporting the same load. Solar tracking applications feature ultra-low bushing rotation speeds, so only the maximum P-value was considered. Using this in conjunction with force and design factor, the minimum length of bushing face area was computed. Any bushings that are oscillating can be completely enclosed. Sliding bushings are at risk of dust abrasion while moving, so PTFE wiper bushings are installed to prevent dust ingress into the bushing.

To select the design factor we considered that the loading over the solar panel's entire range of motion could change by a factor of four; thus, we selected a factor of 4. For a system to be safe for use around other people Mott states that a design factor minimum of 4 should be used and this was compounded, yielding a total design factor of 16. Since we have six unique bushings to specify we wrote a MATLAB script to implement bushing calculations from Shigley's Textbook to automatically calculate minimum bushing length. This minimum length is used to select a catalogued bushing with an equal or greater length. Finally, the catalog bushing length is used to calculate the safety factor.

Because of the 6 year life expectancy of the system as specified by the client, we calculated the wear factors based off of this value. The resulting wear factor for all bushings was in the range of 10^{-10} in, and we concluded that this value was small enough to be considered negligible to our design considerations.

Worth noting is bushing F which slides along the shaft as the cylinder extends or retracts. The bushings have a coefficient of friction of .05, requiring a piston rotation allowance of $\pm$ 3 degrees to allow the bushings to slide. This angle will not impact the play in the system or the accuracy of the tracking because the system will always rest on the limits at the 3 degrees.

Number	Component
1	Reservoir
2	Variable Flow Pump
3	Pressure Relief Valve
4	Solenoid Control
5	3-way Valve
6	Displacement Transducer
7	Double-Acting Piston
8	Outlet Filter
9	Solenoid Valve
10	Flow Control Valve

Table 7: Circuit Component List

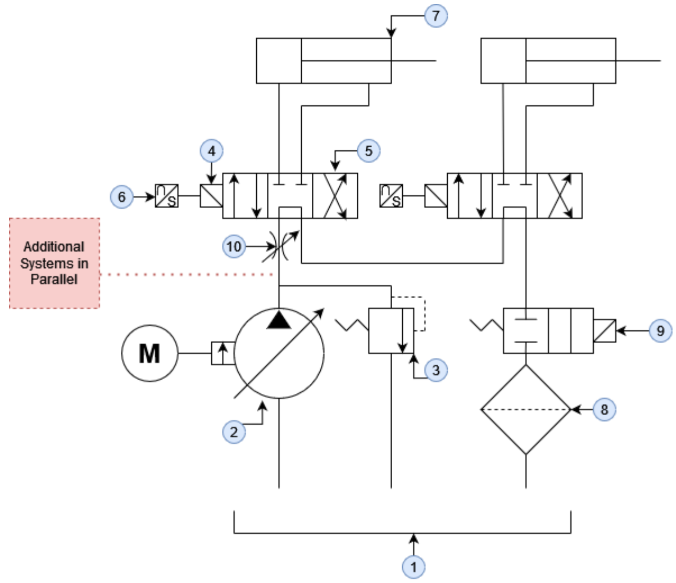


Table 8: Hyraulic Circuit

Table 9: Bushing specifications.

	Design Factor	Load lbf	Lifetime (years)	ID (mm)	FD (mm)		L (mm)	SF Actual
A	11	260	10	20	N/A	L_catalog	30	11.8
B	16	130	10	25	50	L_catalog	20	19.7
C	16	100	10	25	N/A	L_catalog	25	32.0
D	16	200	10	20	45	L_catalog	25	16.0
E	16	130	10	25	N/A	L_catalog	20	19.7
F	16	260	10	100	100	L_catalog	50	98.5

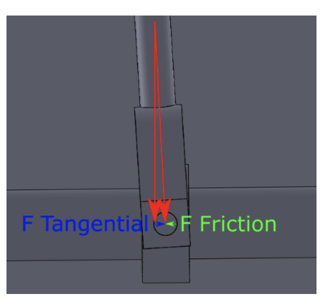


Figure 6: Bushing forces.

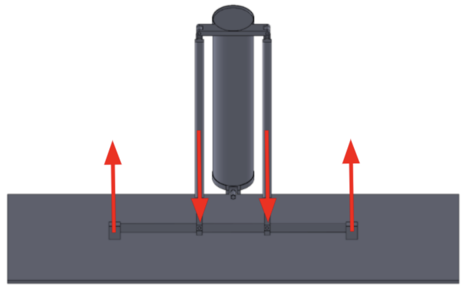


Figure 7: Shaft loading.

3.4 Shaft

The sliding shaft used a design factor of 16 for the same reasons as described in the preceding section. We expect a 260 lbf load to be applied at each point on the shaft as shown in the diagram below. The distance between the reaction force in the middle and outside gives us a maximum bending moment in the shaft of 7800 in lbf. It is difficult to determine how the moment varies along the entire length of the shaft for this particular loading configuration, so we assumed that the maximum moment is alternating, as that is the worst case. Using the fatigue calculations listed in Shigley's in conjunction with this moment and the specified design factor, a safety factor of 19 was calculated for a 100mm OD x 62.5mm ID x 2000mm L shaft made of ASTM A131 Steel.

4 Conclusions and Recommendations

Finally, our list of recommended elements for this system are summarized in the table below along with their safety factors:

Table 10: Summary of components.

Component	**Selection**	**Safety Factor**
Hydraulic Cylinders	PARKER HYDRAULICS 2HX	4
Hosing	ISO 12151-5 Rated Hydraulic Hose	4
Tank	60gal. Plastic Tank	4
Pump	Hydraulic Pump, SAE-AA	4
Bushing A	2x Johnson Metall Oiles 500 SPB-202830	22
Bushing B	2x Johnson Metall Oiles 500 SPFG-2520	25
Bushing C	2x Johnson Metall Oiles 500 SPB-253325	40
Bushing D	1x Johnson Metall Oiles 500 SPFG-2025	25
Bushing E	4x Johnson Metall Oiles 500 SPB-253320	25
Bushing F	2x Johnson Metall Oiles 500 SPB-10012050	125
Shaft	OD 100mm x ID 62.5mm X L 2000mm	19

All these elements have been selected to meet the requirements outlined in Table 1 and score well in the evaluation criteria of cost, reliability, mass, manufacturing, and simplicity. For compactness and creativity, our system on aggregate has won those categories already. Within cost, a noteworthy aspect is that all of our high safety factors contribute negligibly to cost increase (relative to specifying lower safety components) and so we need not bring those components down to our baseline 4.0 safety factor. A bill of materials, in Appendix F, tabulates the relative expense against each component, with the total cost of the design amounting to $14,618.30 (USD).

While the design meets the calculated requirement safety factors, to ensure safety before implementation, we recommend some further steps be taken. These include:

- Gathering more empirical data on the placement environment to reduce uncertainties regarding locational wind, other weather phenomena, local gusts, any fluid interference with the wind turbines, and proximity to humans.

- Further develop and detail the component design: Once we do a FMEA we would focus on specific parts for ex. We foresee splitting the bushing guide shaft into two would be worthwhile by reducing bending moments, along with overall system mass (which is an evaluation criteria as well)
- Developing our own system for mounting, since in a complete failure event there is risk to human life. A custom mounting frame and fixing force analysis would inspire greater confidence to proceed to a prototype phase.
- Implement Solar Tracking Control System: this would involve a sun detection/tracking algorithm driving the hydraulics. This is to ensure that our system is sufficiently efficient in its actual purpose (maximum solar energy collection) to justify a build stage.

Certainly taking into account the recommendations of this report and accomplishing the further steps outlined above, we support this system and would ultimately propose to move to make a proof of product that encompasses all of the client's requirements, which we do meet, as well as a demonstration of the control system functioning and the system model being self contained.

5 Appendices

5.1 Appendix A: Design Needs

Shown in Table 11 are the design needs which were used to extract the design requirements and evaluation criteria.

Table 11: Design needs.

Need Based on Knowledge of Pavillion Renewables, the Pilot Site, the Application area or the Existing Infrastructure	**Justification of Need**
The design accepts power from the existing infrastructure.	While the design will often power itself through the solar panel, it needs to be able to accept power from the existing infrastructure. This is in case the power generated by the solar panel is not enough to power the device.
The design withstands the wind loads of the environment surrounding it.	Since the pilot site for the design is a vertical-axis wind turbine, it is implied that wind speeds will be high in the application area. Thus, wind loads should be considered in the design.
The design is able to succeed in the worst-case loading scenario.	For the purposes of safety, it is crucial that the device does not create any hazards for nearby personnel. In addition, since wind turbines could be relied upon for critical electrical infrastructure, it should be as reliable as possible.
The design achieves dual-axis rotation for the solar panel to orient it in the direction of the solar radiation field.	Dual-axis rotation is implied by the function of the device.
The design withstands the temperatures of the environment surrounding it.	Since the pilot site for the design is in the Kingdom of Bahrain, a traditionally warm region, temperature effects should be accounted for.
The design is able to be mounted onto the mast of the vertical-axis wind turbine.	Mounting onto the mast of a vertical-axis wind turbine is implied by the function of the device.
The design achieves actuation of a heavy solar panel.	Actuation of a heavy solar panel is implied by the function of the device.
The design is not vulnerable to ultraviolet degradation over its lifespan.	Since a solar panel is aimed to seek out solar radiation as a part of its function, the components of the design should not degrade due to ultraviolet radiation of the sun.
The design is cost-effective.	As the pseudo client is a business, their interests would likely incline them to a more cost-effective solution.
The design requires minimal maintenance.	By nature of the pilot site being a wind turbine, the device will most likely be situated in a generally remote location. Thus, maintenance could be challenging and the less of it the better. In addition, since wind turbines could be relied upon for critical infrastructure, it is dangerous if the system is consistently down for maintenance.
The design has a long service life.	As wind turbines are to be utilized to generate power for large-scale infrastructure, they are relied upon for many critical functions. This makes it dangerous if the system needs to be replaced frequently.

5.2 Appendix B: Load Calculations

The total force can be modelled as follows:

$$\sum F = F_{PanelWeight} + F_{WindLoad}$$

Wind Load Calculation:

Assumptions:

1. For worst case, wind is directly perpendicular to panel for maximum drag.
2. Wind is evenly distributed over panel (Panel area is fairly small so this is reasonable, but discrepancies will be observed)
3. Complex fluid interaction due to the wind turbine is not factored into this calculation. We recommend collecting empirical data to fully investigate this. Factor of Safety should account for this uncertainty.

$$F_{WindLoad} = \frac{1}{2}v^2 C_D A$$

Coefficient of Drag (C_D) = 1 [Square Profile]
Air Density (ρ) = 1.126$\frac{kg}{m^3}$ [4]
Maximum Wind Velocity (v) = 10.2 m/s (99.7% Confidence from a 10-year dataset) [4]
Panel Area (A) = 10.22 m^2 [10]

$$F_{WindLoad} = (\frac{1}{2})(10.2)^2(1)(10.22) = 1197.75N = 269.27lbf$$

Panel Weight Load Calculation:

Assumptions:

1. For worst case, panel is positioned horizontally where weight acts perpendicular to the panel
2. For worst case, pistons bear the full weight of the panel. In reality, mounting at the top of the panel will provide a supporting reaction force.

$$F_{PanelWeight} = mg$$

g = 9.81 m/s^2 [7]
m = 114.4 kg [10]

$$F_{PanelWeight} = (114.4)(9.81) = 1122.26N = 252.29lbf$$

*No moment loading because all forces act symmetrically about centerlines on panel

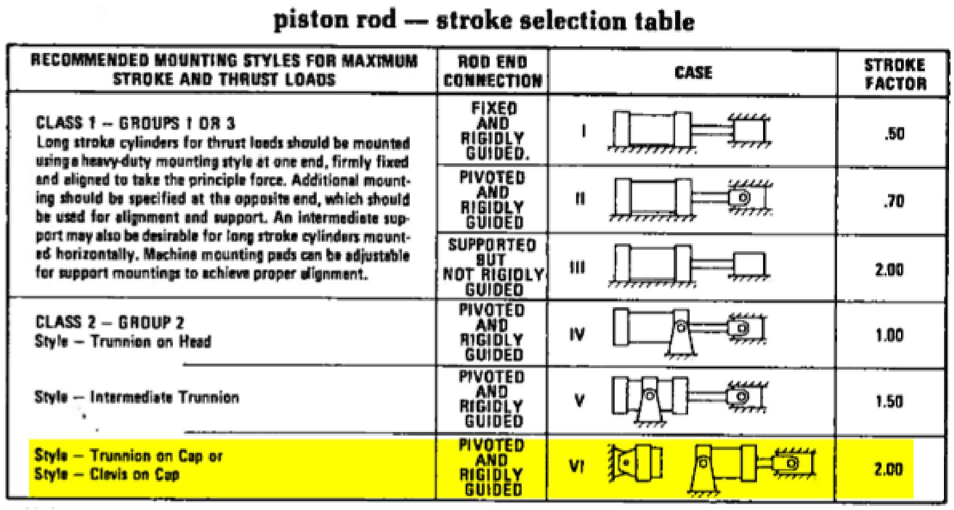
Total Loading on Panel:

$$\sum F = F_{PanelWeight} + F_{WindLoad} = 269.27lbf + 252.29lbf = 521.56lbf$$

5.3 Appendix C: Hydraulic Cylinder Selection Tools

5.3.1 Stroke Factor

The cylinder has pin joints on both ends, corresponding to case VI. From table b-5 of the Parker Design Engineer Handbook this yields a stroke factor of 2.0, as shown in Figure 8 below.



piston rod — stroke selection table

RECOMMENDED MOUNTING STYLES FOR MAXIMUM STROKE AND THRUST LOADS	ROD END CONNECTION	CASE	STROKE FACTOR
CLASS 1 – GROUPS 1 OR 3 Long stroke cylinders for thrust loads should be mounted using a heavy-duty mounting style at one end, firmly fixed and aligned to take the principle force. Additional mounting should be specified at the opposite end, which should be used for alignment and support. An intermediate support may also be desirable for long stroke cylinders mounted horizontally. Machine mounting pads can be adjustable for support mountings to achieve proper alignment.	FIXED AND RIGIDLY GUIDED.	I	.50
	PIVOTED AND RIGIDLY GUIDED	II	.70
	SUPPORTED BUT NOT RIGIDLY GUIDED	III	2.00
CLASS 2 – GROUP 2 Style – Trunnion on Head	PIVOTED AND RIGIDLY GUIDED	IV	1.00
Style – Intermediate Trunnion	PIVOTED AND RIGIDLY GUIDED	V	1.50
Style – Trunnion on Cap or Style – Clevis on Cap	PIVOTED AND RIGIDLY GUIDED	VI	2.00

Figure 8: Stroke Factor Selection - Table b-5 [9]

5.3.2 Rod Selection

Graph b-1 from the Parker Design Engineer Handbook dictates the selection of the piston diameter based on a length of 60in and operating thrust of 1050lbs. The graph indicates that a piston rod diameter of **1 3/8 in** is appropriate. This is shown in Figure 9 below.

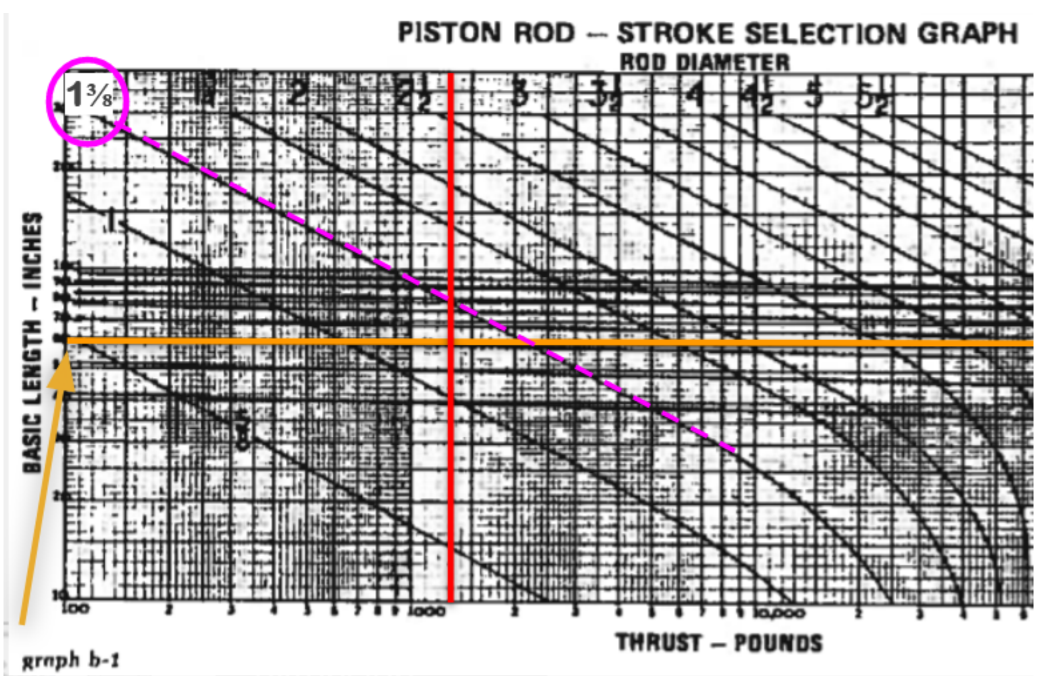


Figure 9: Rod Diameter Selection - Graph b-1 [9]

5.3.3 Bore Selection

From Table b-5 in the Parker Handbook, we see that a **2in** bore size is the smallest bore size to support a piston size of 1 3/8 in, this is shown in Figure 10. We selected this size for compactness, reduced weight, and reduced cost.

CYLINDER BORE-INCHES	PISTON ROD DIA.-INCHES	PISTON ROD AREA SQ. IN.
1	0	0
	1/2	0.196
	5/8	0.307
1½	0	0
	5/8	0.307
	1	0.785
2	0	0
	5/8	0.307
	1	0.785
	1-3/8	1.485
2½	0	0
	5/8	0.307
	1	0.785
	1-3/8	1.485
	1-3/4	2.405

Figure 10: Bore Diameter Selection - Table b-5 [9]

5.3.4 Nominal Pressure

With a known bore diameter we can calculate the operating pressure using the following relations:

$P = \frac{F}{A}$
F = 1043.12 lbs (See Appendix B)
A $= \pi \cdot r^2 = \pi \cdot 1^2 = 3.142in^2$

$P = \frac{(1043.12)}{(3.142)} = 332.04psi$

We decided to round this number up to **400psi** to allow us to specify a standard pump and motor.

5.3.5 Flow rate, Pump, and motor Selection

From table-b5, PH Design Engineer's Handbook:

Rod-Side Flow Rate (10FPM) = **0.86 GPM** Blank Side Flow Rate (10FPM) = **1.63 GPM**

This is shown in Figure 11 below:

Target Piston Velocity = 2.5fpm [based on the target full piston extension time of approximately 30s]

$Q = \frac{Q_{10}}{U_{10}} \cdot U$

$Q_{Rod-Side} = \frac{(0.86GPM)}{(10FPM)} \cdot (2.5) = 0.215GPM$
$Q_{Blank-Side} = \frac{(1.63GPM)}{(10FPM)} \cdot (2.5) = 0.408GPM$

Pistons are operated sequentially, therefore the required flowrate for the system is the flowrate of one piston

CYLINDER BORE-INCHES	PISTON ROD DIA.-INCHES	PISTON ROD AREA SQ. IN.	CYLINDER NET AREA SQ. IN.	FLUID DISPLACEMENT AT 10 FT. PER MINUTE PISTON VELOCITY G.P.M.	C.F.M.
1	0	0	0.785	0.41	0.054
	1/2	0.196	0.589	0.30	0.041
	5/8	0.307	0.478	0.16	0.033
1½	0	0	1.77	0.92	0.123
	5/8	0.307	1.46	0.76	0.101
	1	0.785	0.98	0.51	0.068
2	0	0	3.14	1.63	0.218
	5/8	0.307	2.84	1.48	0.197
	1	0.785	2.36	1.23	0.164
	1-3/8	1.485	1.66	0.86	0.115

Figure 11: Flow rate at a 10FPM fluid velocity - Table b-5 [9]

per unit.

In a single unit, $Q_{Blank-Side}$ is limiting. Therefore we will take the minimum required flowrate for the system as 0.408GPM.

Since piston velocity is not a critical requirement, we have chosen to round this value to 0.5 GPM to easily specify standard components. Since network will require 10 units, the required flow rate for the network will be 5GPM

From Table c-1 (Figure 12), PH Design Engineer's Handbook, a 1.40hp single phase motor is appropriate.

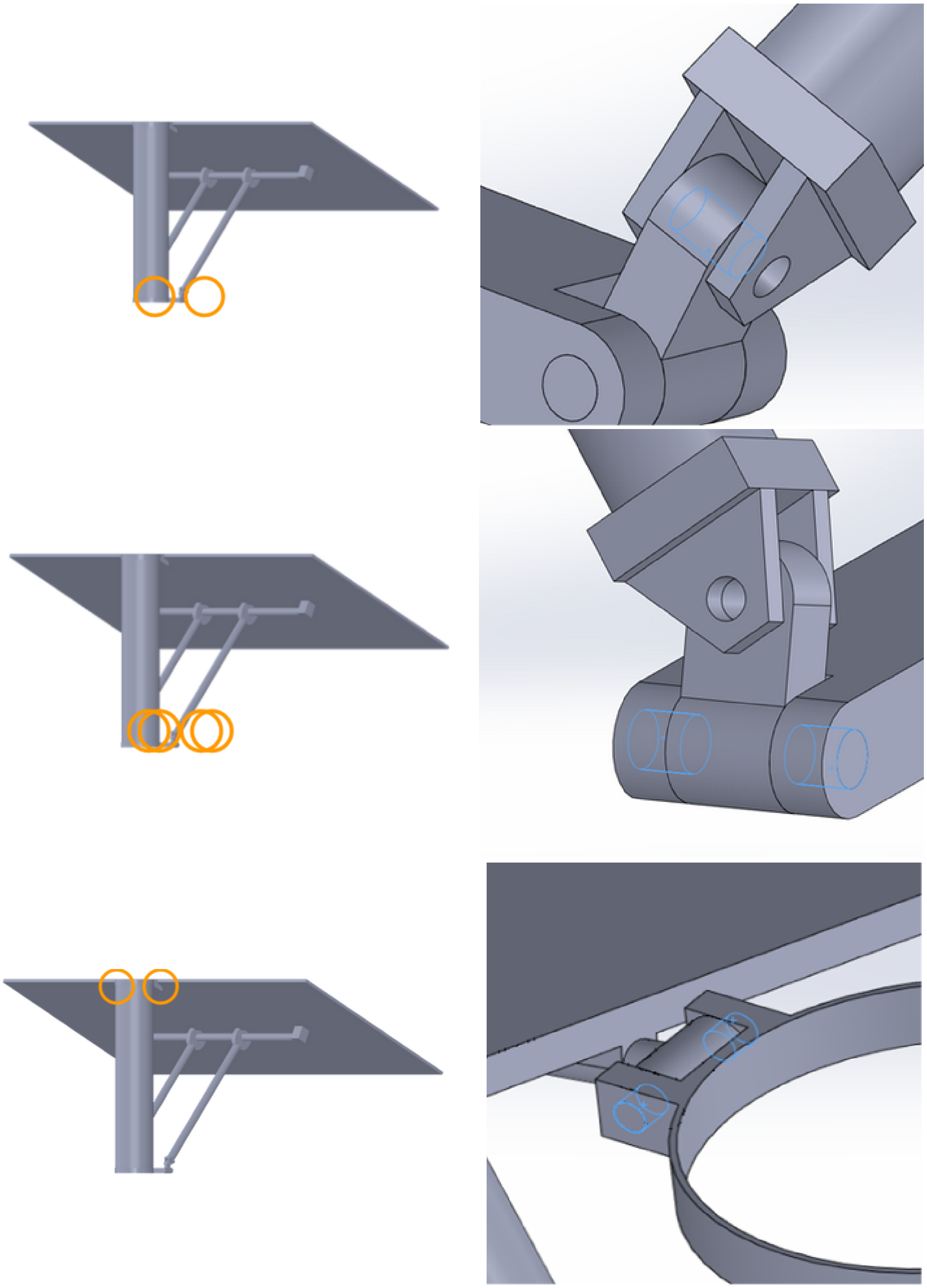
GPM	100 PSI	200 PSI	250 PSI	300 PSI	400 PSI
1/2	.04	.07	.09	.11	.14
1	.07	.14	.18	.21	.28
1-1/2	.10	.21	.26	.31	.41
2	.14	.28	.35	.42	.56
2-1/2	.17	.34	.43	.51	.69
3	.21	.42	.53	.63	.84
3-1/2	.24	.48	.60	.72	.96
4	.28	.56	.70	.84	1.12
5	.35	.70	.88	1.05	1.40

Figure 12: Motor Power Requirements (hp) - Table c-1 [9]

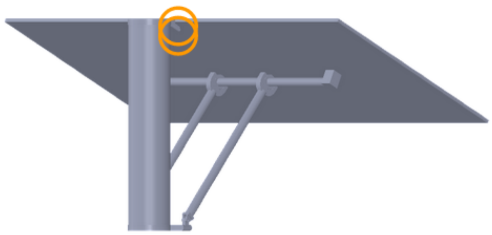
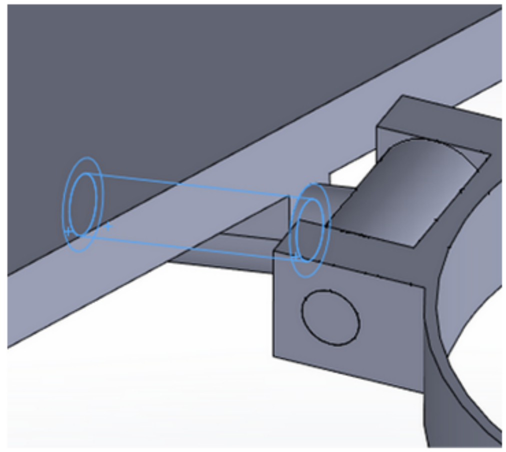
We have selected a suitable hydraulic pump based on the minimum flow requirements of 5GPM and the pressure requirement of 400PSI. The cheapest option from McMaster-Carr to meet these requirements was the 6296K47 model Hydraulic Pump, SAE-AA, 5.2 gpm Maximum Flow

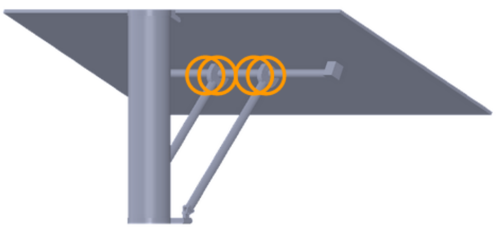
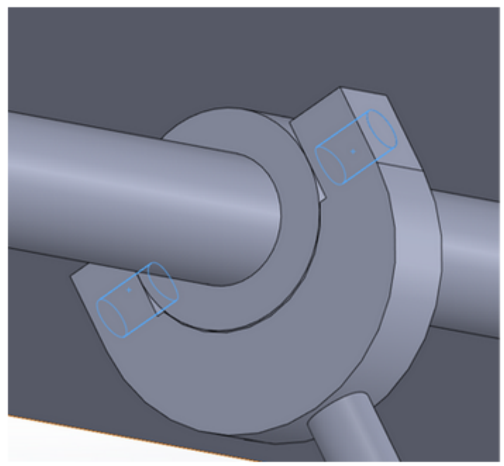
5.4 Appendix D: Bushings Calculations

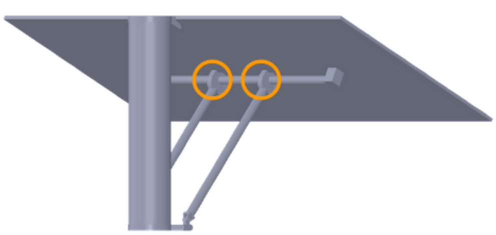
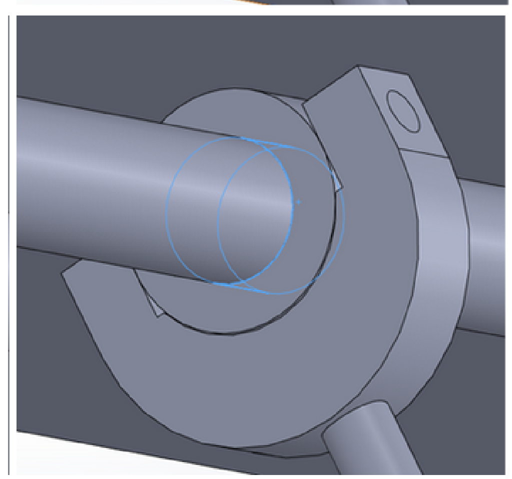
5.4.1 Bushing locations (left column) and bushing geometries (right column) for bushings A-C (top to bottom respectively)



5.4.2 Bushing locations (left column) and bushing geometries (right column) for bushings D-F (top to bottom respectively)

5.4.3 Calculation Procedure

Table 12: Bushing known values.

Property	Value	Source
Friction coefficient	0.05 [-]	Shigley's T12-8
Maximum Pressure Rating	4206.09 [PSI]	Oiles 500 Manual
K	0.6x10ˆ10 [-]	Shigley's T12-10
Linear f_1 Coefficient	2	Shigley's T12-10
Rotation/Oscillation f_1 Coefficient	1.3	Shigley's T12-10
F_2 Coefficient	1	Shigley's T12-9

We begin with the known values shown in Table 12. Using these values we can begin calculating the design load using the design factor and the expected loading as follows:

$$F_d = F \cdot n_d$$

where F is the expected loading and n_d is the design factor. We then calculate the required angular allowance using the equation below and setting it to 0 as follows:

$$sin(\frac{\pi \cdot \theta}{180} - cos(\frac{\pi \cdot \theta}{180}) = 0$$

Next, we calculate the maximum length of the bushing using Shigley's equation 12-31 and 12-33, as shown:

$$L_{min} = 5 \cdot D_{in}$$

$$L_{max} = max(L_{min}, max(4 \cdot F_d/(\pi \cdot ID \cdot P_{max}), L_{override}))$$

Next, we use Shigley's equation 12-28 to calculate the pressure:

$$P = \frac{F_{max}}{L \cdot D_{in}}$$

and we can then use this with Shigley's equations 12-32 and 12-27 to calculate the radial wear, linear sliding wear, face rotational wear, and total wear on the bushing as follows:

$$w_r = f_{1,r} \cdot f_{2,r} \cdot K \cdot F_d \cdot \text{Rev}_{total}$$

$$w_l = f_{1,l} \cdot f_2 \cdot K \cdot P \cdot S_{total}$$

$$w_{fr} = f_{1,l} \cdot f_2 \cdot K \cdot P_f \cdot S_{flangetotal} \cdot 25.4$$

$$w_t = (w_l + w_r) \cdot 25.4$$

These values allow us to calculate our safety factors and thus evaluate our bushings.

5.4.4 Script

The calculation procedure shown previously was automated for efficiency. The script used to complete the calculations is shown below.

```
function out = Bush(DF,F,life,Rev_day,Dis_day,ID,FD,L_over)%
%DF=design Factor
%F=force lb
%life in years
%Rev_day=revolutions per day
%Dis_day=linear distance per day ft
%L_over=length override for bushing calculation using catalogue bushing
%325 calc's Ethan Alexander
%problem properties
life_day=life*365;%days
F_max=F*DF;%lb
ID=ID/25.4;%in diameter
FD=FD/25.4;%in outer diameter
L_over=L_over/25.4;%in overide length
%%oiles 500SP properties
fs=.05;%fiction coefficient (Shig T12-8)
P_max=4206.09;%psi 29*10^6 (oiles 500 manual)
K=.6*10^-10;%in^3*min/(lbf*ft*h)
f1_l=2;%linear f1 coefficient (Shig T12-10)
f1_r=1.3;%rotation/oscillation f1 coefficient (Shig T12-10)
f2=1;%f2 coefficient (Shig T12-9)
%%sliding bushing angular play required
syms x;
eqn=sin(pi*x/180)-cos(pi*x/180)*fs == 0;
S=solve(eqn);
ang=real(double(S(2)));%deg
%bushing calc oiles 500SP
L_min=.5*ID;%(Shig E12-33)
L=max(L_min,max(4*F_max/(pi*ID*P_max),L_over));%in (Shig E12-31)
P=F_max/(L*ID);%(Shig E12-28)
SFd=P_max/P*DF/4*pi;
%oscillating/revolving wear
Rev_tot=life_day*Rev_day/60;%rev*hour/min = t*N
w_r=f1_r*f2*K*F_max*Rev_tot/(3*L);%(Shig E12-32)
%linear sliding wear
Dis_tot=life_day*Dis_day/60;%ft*hour/min = t*V
w_l=f1_l*f2*K*P*Dis_tot;%(Shig E12-27)
w=(w_r+w_l)*25.4;%in
%face rotational wear
Pf=F_max/(FD^2-ID^2)/4*pi;%psi
Disf_tot=FD/12*pi*Rev_tot;%ft*hour/min = t*V
w_f=f1_l*f2*K*Pf*Disf_tot*25.4;%(Shig E12-27)
SFf=P_max/Pf*DF/4*pi;
%pin shear
PS=4/3*F_max/(pi*ID^2/4);%max shear in a cylinder
Lmm=L*25.4;% length in mm
out=[ang,Lmm,w,w_f,PS,SFf,SFd]
```

5.4.5 Bushing Calculation Results

Shown in Tables 13-15 are the detailed results of the bushing calculations using both hand calculations and MATLAB.

Table 13: Bushings calculations results.

Bushing	Design Factor	Load (lbf)	Lifetime (years)	Revolutions per Day	Linear Distance per Day (ft)
A	11	260	10	0.25	0
B	16	130	10	0.25	0
C	16	100	10	0.25	0
D	16	200	10	0.25	0
E	16	130	10	0.25	0
F	16	260	10	0.25	9.3

Table 14: Bushings calculations results continuted.

Bushing	Wear Radial (in)	Wear Flange (in)	Pin Shear (lbf)	Flange SF	Radial SF	Min motion angle
A	0.000024	N/A	7831.1	N/A	11.8	N/A
	0.000026	N/A	7831.1	N/A	11.0	N/A
B	0.000027	0.000013	3645.0	94.0	19.7	N/A
	0.000033	0.000013	3645.0	94.0	16.0	N/A
C	0.000016	N/A	2803.9	N/A	32.0	N/A
	0.000033	N/A	2803.9	N/A	16.0	N/A
D	N/A	0.000021	N/A	52.97	16.0	N/A
	N/A	0.000021	N/A	52.97	16.0	N/A
E	N/A	N/A	3645.0	N/A	19.7	N/A
	N/A	N/A	3645.0	N/A	16.0	N/A
F	0.00095	N/A	455.6	N/A	98.5	2.86
	0.00095	N/A	455.6	N/A	98.5	2.86

Table 15: Bushings calculations results continued.

Bushing	ID (mm)	FD (mm)		L (mm)	SF Actual
A	20	N/A	L_catalog	30.00	11.8
			L_minimum	27.93	
B	25	50	L_catalog	20.00	19.7
			L_minimum	16.25	
C	25	N/A	L_catalog	25.00	32.0
			L_minimum	12.50	
D	20	45	L_catalog	25.00	16.0
			L_minimum	25.00	
E	25	N/A	L_catalog	20.00	19.7
			L_minimum	16.25	
F	100	100	L_catalog	50.00	98.47
			L_minimum	50.00	

5.5 Appendix E: Shaft Calculations

5.5.1 Shaft Material Properties

The properties of our chosen shaft material is shown in Table 16.

Table 16: Shaft material properties.

Density	7.80 g/cc	0.282 lb/in^3	Typical of ASTM Steel
Mechanical Properties	**Metric**	**English**	**Comments**
Tensile Strength, Ultimate	400 - 490 MPa	58000 - 71100 psi	
Tensile Strength, Yield	220 MPa	31900 psi	
Elongation at Break	21%	21%	in 200mm
	24%	24%	in 50mm
Bulk Modulus	160 GPa	23200 ksi	Typical for steel
Shear Modulus	80.0 GPa	11600 ksi	Typical for steel
Component Elements Properties	**Metric**	**English**	**Comments**
Carbon, C	0.26%	0.26%	
Iron, Fe	97%	97%	
Manganese, Mn	>= 0.65 %	>= 0.65 %	
Phosphorus, P	0.05%	0.05%	
Sulfur, S	0.05%	0.05%	

5.5.2 Calculation Procedure

We begin with the known properties shown in Table 17. We then calculate the design, or maximum accounted for, force on the shaft. We do this using the design factor and the expected force from the load calculations, as follows:

$$F_{des} = F_{max} = F \cdot D_F$$

where F represents the expected force and D_F represents the design factor.

We then calculate the area and second moments of area as follows:

$$I = \pi \frac{(D_{out}^4 - D_{in}^4)}{64}$$

$$A = \pi \frac{D_{out}^2 - D_{in}^2}{4}$$

Table 17: Shaft knowns.

Known Variable	Value
Yield Strength	31,900 psi
Ultimate Strength	58,000 psi
Outer Diameter	4 in
Inner Diameter	2.5 in
Length	80 in
Design Factor	16 in
Calculated Force	260 lb

By inspection, we can determine that the alternating moment will be greater than the static moment, so we set the values as per this evaluation:

$$M_{des} = M_{max} = M_{alternating}, M_{static} = 0$$

Finally, using these values, the known values in Table 17 and Shigley's equations 7-6 and 7-7 (DE-Goodman criterion) shown below, we can calculate the shaft stress to ensure it is acceptable and determine our factors of safety.

$$n = \frac{\pi d^3}{16}(\frac{A}{S_e} + \frac{B}{S_{ut}})^{-1}, \text{where } A = \sqrt{4(K_f M_a)^2 + 3(K_{fs}T_a)^2}, \text{and } B = \sqrt{4(K_f M_m)^2 + 3(K_{fs}T_m)^2}$$

5.5.3 MATLAB Script

The calculation procedure shown previously was automated for efficiency. The script used to complete the calculations is shown below.

```
%shaft
L=80;%in length
DF=16;%
F=260;%lb
F_max=F*DF;%lb
Mmax=F_max*30% in lb max moment on the shaft
OD=4;%in
ID=2.5;%in
I=pi*(OD^4-ID^4)/64%second moment of area
A=pi*(OD^2-ID^2)/4;%area
Kf=1%stress concentration factor
Se=31900;% yield strength
Sut=58000;%Ultimate strength
Ma=Mmax;%alternating Moment
Mm=0;%static moment
n=16*(16/(pi*(OD^3-ID^3))*(1/Se*(4*(Kf*Ma)^2)^.5+1/Sut*(4*(Kf*Mm)^2)^.5))^-1%(
    Shig E7-7)
```

5.6 Appendix F: Bill of Materials

Shown in Table 18 is the detailed bill of materials of the team's design.

Table 18: Bill of materials.

Component	**Quantity for 10 Units**	**$ (USD) for 10 Units**
Bushing A	20	$ 588.70
Bushing B	40	$ 743.80
Bushing C	20	$ 490.40
Bushing D	20	$ 700.50
Bushing E	40	$ 486.50
Bushing F	20	$ 1,194.10
Parker Hydraulics	20	$ 9,000.00
Hosing	100ft	$ 526.00
Tank	1	$ 602.00
Pump	1	$ 287.00
	Total:	**14,618.30**

5.7 Appendix G: Combination WDM and Pugh Chart

Shown in Figure 13 is the tool used by the team to evaluate concepts for the final design of the sun-tracking solar panel.

Mechanism Pugh Chart	Reliability	Cost	Mass	Compactness	Manufacturing	Simplicity	Creativity	TOTAL	WEIGHTED	RANK
	15%	25%	10%	10%	15%	20%	5%	100%	-	
Crescent-Worm (DATUM)	0	0	0	0	0	0	0	0	0	5
Sail-Panel	-1	-1	1	-1	-1	-1	1	-3	-0.7	11
Center-Pivot + LinAx	0	1	1	0	1	1	0	4	0.7	1
Origami Blinds	-1	-1	1	1	-1	-1	1	-1	-0.5	8
Planetary Rooftop	-1	-1	0	-1	1	0	1	-1	-0.3	7
Lazy Susan	-1	1	-1	-1	1	0	0	-1	0.05	4
Lazy Susan Gears*	1	1	-1	0	1	1	0	3	0.65	2
The Crane	-1	-1	-1	-1	1	-1	1	-3	-0.6	9
Basic Idea	0	0	0	-1	0	0	0	-1	-0.1	6
Gimbalized Hydraulics	-1	-1	-1	-1	1	-1	1	-3	-0.6	9
Yardarm	0	0	1	1	-1	0	1	2	0.1	3

Figure 13: Pugh chart with weights for concept evaluation.

References

[1] IEA. "World Energy Outlook 2017." IEA, https://www.iea.org/reports/world-energy-outlook-2017.

[2] Spiller, Elisheba, et al. "The Environmental Impacts of Green Technologies in TX." Energy Economics, vol. 68, 2017, pp. 199–214., https://doi.org/10.1016/j.eneco.2017.09.009.

[3] Kafka, Jennifer L., and Mark A. Miller. "A Climatology of Solar Irradiance and Its Controls Across the United States: Implications for Solar Panel Orientation." Renewable Energy, vol. 135, 2019, pp. 897–907., https://doi.org/10.1016/j.renene.2018.12.057.

[4] Buflasa, H. A., Infield, D., Watson, S., & Thomson, M. (2008). Wind Resource Assessment for the Kingdom of Bahrain. Wind Engineering, 32(5), 439-448. doi:10.1260/030952408786411976

[5] D.o.o., Y. M. (n.d.). Manama, Bahrain - Detailed climate information and monthly weather forecast. Retrieved from https://www.weather-atlas.com/en/bahrain/manama-climate#temperature

[6] Budynas, R. G., & Nisbett, J. K. (2020). Shigleys mechanical engineering design. New York, NY: McGraw-Hill Education.

[7] Mott, R. L., Vavrek, E. M., & Wang, J. (2018). Machine elements in mechanical design. New York: Pearson.

[8] Parker Training. (n.d.). Fluid Power Basics [PDF].

[9] Parker Training. (n.d.). Design Engineers Handbook [PDF].

[10] Solar One. (n.d.). S1-550-144M Solar Panel Data Sheet [PDF].