Design and Analysis of Hydraulically Driven Actuation System For a Parabolic Solar Trough

by

Katarina Popovic

Submitted to the
Department of Mechanical Engineering
in Partial Fulfillment of the Requirements for the Degree of

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Abstract

This thesis documents Katarina Popovic's contribution to the design of hydraulic cylinder actuation system for day to day solar trough sun tracking, a semester long project within 2.752 Development of Mechanical Products class. The primary goal of this project was to design a robust mechanical system while reducing the complexity and thus overall cost of the solar trough assembly for the production line. The mechanism suggested in this thesis actuates both sides of the solar trough simultaneously, as well as exploit hydraulic cylinder's full range stroke in order to deliver $\pm 110^{\circ}$ rotational requirement. The rotational motion is achieved through a pulley and a wire rope system, actuated by a single, double action double rod cylinder. As this project funding was received from our sole sponsor, an Italian multinational energy company, during the design process the ultimate goal of eventual production line was kept in mind. However slight design modifications have been made in order to install and test the actuation system on the already existing 4m solar trough prototype on site in Pittsfield, New Hampshire.

Thesis Supervisor: Alexander H. Slocum
Tile: Pappalardo Professor of Mechanical Engineering

Acknowledgments

First of all I would like to thank Professor Alex Slocum for all the knowledge that he has open-handedly shared over the course of the last year. I am very fortunate to have you as my mentor both in and out of the classroom. I would also like to thank Sandy Campbell for his enthusiasm and guidance, as well as valuable design reviews throughout this project. Secondly I would like to thank my teammates, Aaron Flores and Felipe Cariillo for their collaboration during the course of the semester. This project has been a great learning experience that would not be possible without your help. Lastly, but not least, to a fellow particle, Folkers E. Rojas: thank you for sharing you zealot enthusiasm for analysis as well as for all the help with formatting of this thesis.

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1. Introduction

As the price of oil increases renewable energy becomes more lucrative. In the U.S., renewable energy makes about 9% of the total energy produced. Currently, solar and PV energy sources makes up about 2% of the total renewable energy. Therefore, solar trough fields collecting solar energy that is fed into the national grid system makes up less than a one quarter of a percent of the total green energy used in the U.S. One of the main driving costs these systems encounter is the actuation and controls. This research seeks to develop a cost effective prototype actuation system for a solar trough. The research presented in this thesis was developed as part of a mechanical product development senior design course.

1.1. Motivation

The solar energy delivered to states west of the Mississippi River is on average 5kWh/m^2 per day, as shown in Figure 1 (EIA, Sept. 2012). Due to the efficiencies in solar collection technology the maximum energy expected from a standard unit is estimated about 1 kW/m^2. As stated by the National Research Energy Laboratory, the main cost reduction of solar concentrator technology will depend largely on: size scale up, production volume, and increased competition (NREL, Oct. 2003). Therefore, a robust and economical actuation system designed for scalability for the solar trough industry is one means to reduce cost and improve the proliferation of solar troughs.

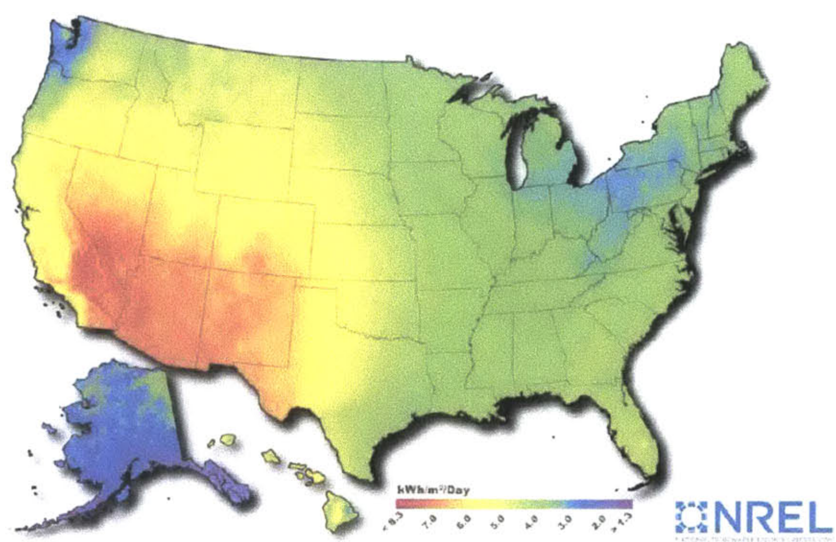


Figure 1 Concentrating Solar resources in the US (EIA, Sept. 2012)

Parabolic trough is a type of solar thermal collector that works by focusing sun rays onto a tube with circulating heat transfer fluid it represents an active energy converting system. The efficiency of the solar trough is dependent on many variables some of which are trough structural design, material selection, and wear (Figueredo, 2011). Energy efficiency of solar trough is also affected by the losses due to the structural support shadow casting while the Sun is on the northern side. As such, it is also important to design compact actuating systems to decrease the shading losses (O'Rourke, 2011).

1.2. Goals

The objective of this project is to design, evaluate and prototype a cost effective parabolic solar trough actuation system that will address flaws within the existing actuation system (O'Rourke, 2011). The design must take into account the environmental loading which includes hot dessert climate with heavy winds during desert storms. Some of design specifications are provided by Ente Nazionale Idrocarburi (ENI), an Italian multinational company, which is a key driving force behind this energy initiative project.

1.3. Functional Requirements

Several of the requirements were provided for the environmental conditions at the installation site. Other functional requirements are set to minimize cost, and complexity while at the same time ensuring that the design functions with the existing structure of solar troughs being developed by ENI as well as the respective scaled model installed at the New Hampshire site.

(1) Torque Rating

Torque requirement is necessary in order to ensure that the trough actuation system can support the solar trough at desired angle even at the harshest weather conditions. The collaboration with another group of ENI at the Polytechnic Institute of Milano determined the maximum torque based on the structural strength of the trough. Their analysis concluded that the actuation system needs to support a maximum torque of 1500 Nm, occurring at an 80^0 stall angle.

(2) Rotational Range

Unlike conventional solar troughs, the rotational angle of the system has to be at least 220^0 (i.e. $\pm 110^0$ from the mid plane). The extended range of motion is required for safety and storage purposes. During desert storms the solar troughs can experience wind loads above the design specifications. One of the options is to size the entire unit to withstand the foreseeable maximum loads; however, the cost of that unit would increase dramatically. As such, the rotational freedom of 110^0 in both clock wise and counter clock wise directions is needed to store and protect the solar trough in case of adverse environmental conditions.

(3) Actuation

The existing solar trough prototype has a low torsional rigidity. Actuating solar trough from only one side will increase deflection at the center of the structure by approximately sixteen times (O'Rourke, 2011). Over time cycling the trough thru those deflections can lead to failure in the components. Therefore, the actuation system proposed must be mirrored on both sides of the solar trough to minimize structural deflections. Looking ahead towards serial in line positioning of the troughs while sharing the actuation system between two different structures is an additional incentive for this requirement fulfillment.

(4) Complexity

Since complexity is usually analogous to cost, minimizing complexity of the actuation system is crucial. One limitation of the existing system is the complex controller due to the multiple cylinders used for actuation of the trough that has further to do with the nature of mechanical design of the actuating system.

(5) Cost

Cost equals profit. Therefore, the cost of operation and parts must be comparable to existing technologies if not cheaper.

(6) Interface

The proposed drive system must be designed to interface with the existing 4 m solar trough prototype. This further comes down to actuating the trough at about its center of gravity point.

2. Existing Prototype

The proposed actuation system will interface with the already existing 4 m solar trough, Figure 2, located in Pittsfield, New Hampshire.

Figure 2: Existing Hardware

2.1. Background

The solar trough was built in the spring semester of 2011 as part of the 2.752 coursework and later as a research in the Precision Engineering Group. An actuation system was incorporated on the solar trough with a dual piston system, shown in Figure 3. However, the control method of the actuation is unreliable and the method adds loads on the shaft that cause it to undergo unfavorable torsion. Since the original installation very little development has been done on the unit.

2.2. Solar Trough Structure

Solar trough itself consists of rolled steel plates coated with a reflective film. This novel design replaced glass panels but at a fraction of the cost (Figueredo, 2011). On each end these panels are supported by a bulkhead that contains steel end tee shaped plate to which existing actuating system attaches to. Due to its size and the need for full 110^0 rotation in both directions, the solar collector structure is lifted by four cement blocks to a desired height.

2.3. Actuation System

The system currently installed in New Hampshire uses a lever arm design, Figure 3

Figure 3: Existing 4 m solar trough actuation system

It employs two inch bore single action single rod hydraulic cylinders with a stroke length of eighteen inches that apply force to the crank plate creating necessary torque for trough rotation. The crank is further connected to the shaft that passes through the troughs' center of gravity. Cylinders are actuated one at the time such that while right cylinder is engaged, and the left one is floating, the trough rotates clockwise, and vice versa. While it cannot be seen from the picture, same actuation system is mirrored and installed on the other side of the solar trough such that at each rotation there are two cylinders powering the rotational motion in each direction.

Although the existing hardware satisfies functional requirements set by ENI, there is room for improvement. The geometric constraint such as angular cylinder mounting does not allow fully exploitation of hydraulic force. As it can be seen from the above cylinders are mounted at an angle. As such, hydraulic force exerted onto the lever arm creates angle of 90^0 only once throughout the trough's 110^0 sweep thus applying maximum torque for a very brief point in time. Also the mounting point of the lever arm located at the end of the shaft coupled with the excessive hydraulic force and single shear effect results in visible shaft deflections during actuation of the system. Second of all, the system of two cylinders per side not only complicates the control scheme of consecutive actuation but due to the design constrains, it is necessary. In the current configuration, each of the cylinders goes through zero point motion once during their float time. Aligning with the lever arm on their retraction portion and thus applying no torque onto the lever arm is the reason behind using two single action cylinders in engaged/float combination instead of one constantly engaged dual action cylinder.

2.4. Loading Analysis

In the initial stage of this project, our group did first order calculations of torque requirements due to the wind loading. As such, the initial step was to determine the loads felt on the trough as a consequence of wind effects.

2.4.1. Wind Shear Loads

The wind velocity was modeled using wind shear power law with shear exponent of 1 and predicted reference speed of wind V_{ref} =15 m/s at the height of h_{ref} = 7 m. Equation 1 shows the relation between the wind strength and the respective chosen height.

$$v_{wind}(h) = v_{ref}(h_{ref}) \cdot \left(\frac{h}{h_{ref}}\right)^{a} \qquad \text{Equation 1}$$

The linear behavior can be represented graphically as shown in Figure 4.

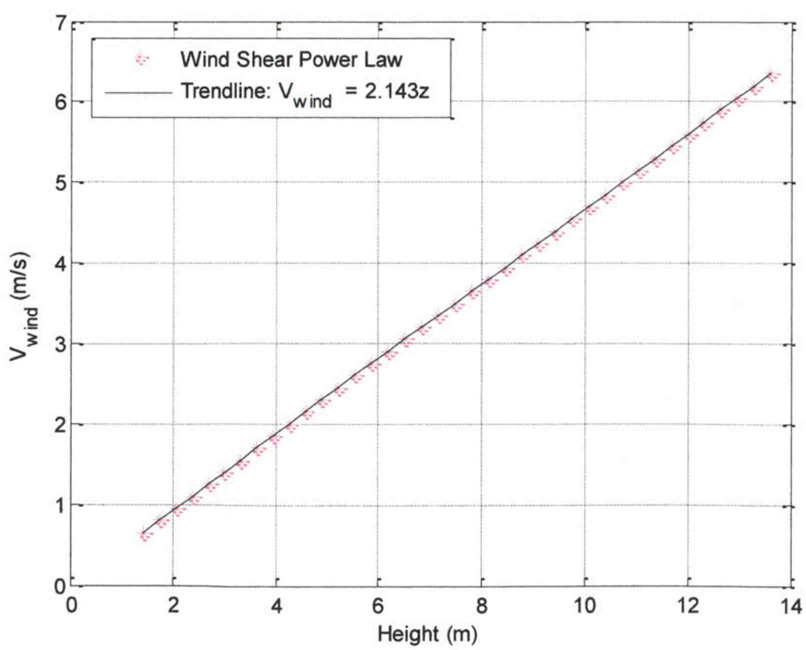


Figure 4: Wind Shear Power Law Prediction

Justification behind high shear exponent value is the nature of solar trough positioning in a matrix configuration and as such creating obstructions to the neighboring systems, as well as relatively short height of the structure.

2.4.2. Torque Calculations

Figure 5 shows a schematic layout used to illustrate the critical components used during the calculation process. Using the predicted wind magnitudes as well as results and dimensions associated with full 12 meter long, and approximately 6 m wide solar trough, torque around center solar trough's center of gravity was calculated by integrating over the wind exposed profile (i.e. equivalent to a + b as shown in Figure 5).

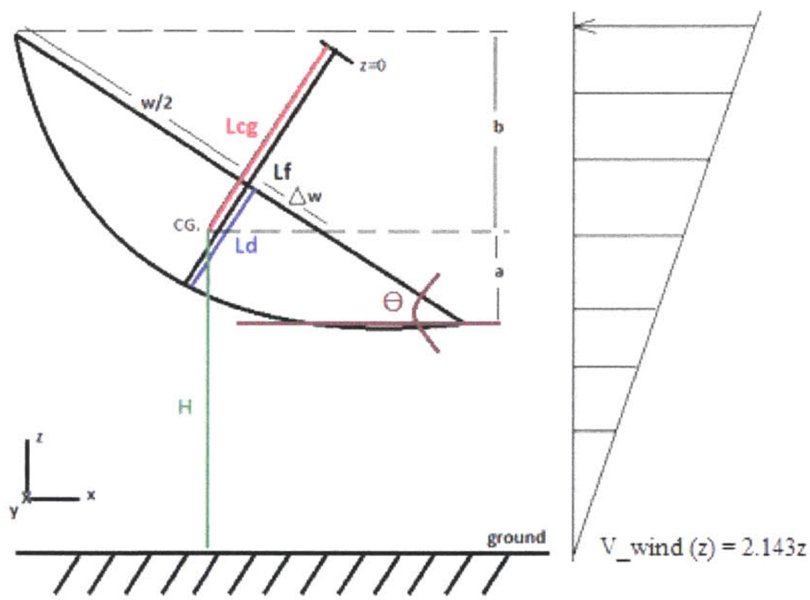


Figure 5: Solar Trough Dimensions

For the purpose of predicting the maximum torque the solar trough would experience, it was approximated that trough structure would be mounted such that center of gravity is 3.5 meters away from the ground (H), with length of 12m (L) and respective width 6m (w), a full scale system dimensions. In addition, based on the parabolic curve calculations of the trough, the depth was calculated to be 1.2m (L_D), and the distance between pivot point (center of gravity) and focal point was 1.1m (L_{CG}) with focal length of 1.71m (L_F). The projected areas, the surface on which wind force is acting are function of Θ, the angle the surface plane makes with respective horizontal such that

$$a = \left(\frac{W}{2} - \Delta w\right) \cdot \sin\theta$$
$$b = \left(\frac{W}{2} + \Delta w\right) \cdot \sin\theta$$
Equation 2

The net torque about the center of rotation of the trough is found by adding positive moment experienced by surface *a* and positive moment as result of lower surface area *b* such that

$$\tau_{net} = (z_b - H)F_b - (H - z_a)F_a$$
Equation 3

Where z_a and z_b are respectively centroid of the net force applied on the upper and lower portion of the trough such that

$$z_a = \frac{0.5 \cdot \rho c_d L \int_{H-a}^{H} V_{wind} \cdot z^2 dz}{0.5 \cdot \rho c_d L \int_{H-a}^{H} z^2 dz}$$
Equation 4

$$z_b = \frac{0.5 \cdot \rho c_d L \int_{H}^{H+b} V_{wind} \cdot z^2 dz}{0.5 \cdot \rho c_d L \int_{H}^{H+b} z^2 dz}$$
Equation 5

Respective upper and lower drag forces were calculated by integrating assumed linear wind profile previously calculated across the trough surface contour and are as such

$$F_a = .5\rho c_d L \int_{H-a}^{H} V_{wind} \cdot z^2 dz$$
$$F_b = .5\rho c_d L \int_{H}^{H+b} V_{wind} \cdot z^2 dz$$
Equation 6

Where ρ is air's density, c_d is the coefficient of drag, and V_{wind} is the trend line fitted to the wind power shear law previously calculated. As a result of these calculations it was found that the maximum torque happens at around 80^0 and it is on the order of 5000 Nm.

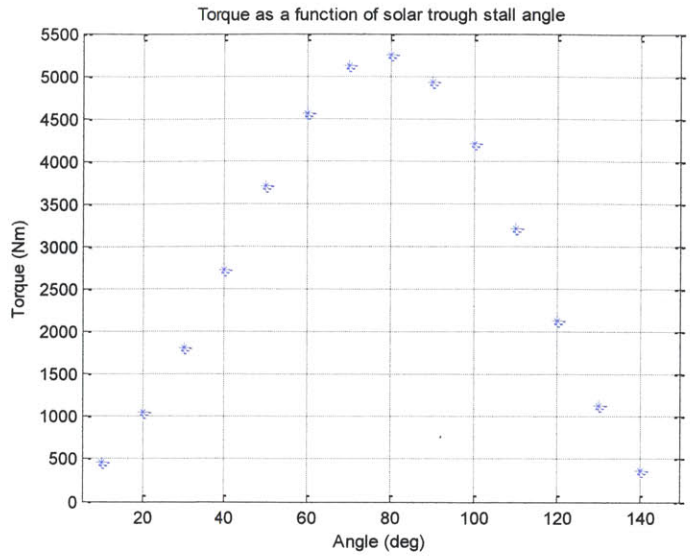


Figure 6: Wind Forces Torque about pivot point as a function of stall angle.

The results presented above were verified by the calculations perform by the design team and as such were taken as a starting point. However, during the progression of the research the design requirements changed, almost towards the end of the project. The collaborative team at the Politecnico di Milano performed a more detailed calculation on the structural strength of the solar trough. The result of their analysis is a new maximum torque of 1500 Nm occurring at 80 degrees (Bernasconi, Fossati, Giglio, & Lodi, 2013). The difference in obtained result is due to their in depth CFD analysis, and consideration of drag, lift as well as pitching moment coefficient while the first order calculation takes into consideration the predominant mode (drag coefficient) only.

3. Strategies

Three main strategies were developed as part of a team brainstorming collaboration as illustrated in Figure 7 : transmission ratio (a), double cylinder (b) and single cylinder (c). The inspiration for the rope and pulley actuation system as oppose to a lever arms came from cranes.

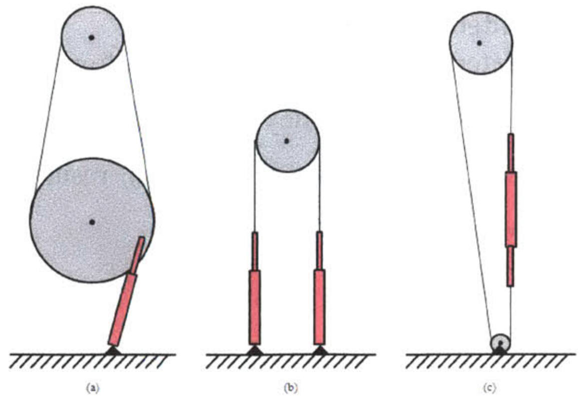


Figure 7: Early Concepts: (a) Transmission Ratio (b) Double Cylinder (c) Single Cylinder .

The capstan effect is employed in all of the strategies in order to take into consideration worst conditions as well as decrease design performance sensitivity to the friction of coefficient.

3.1. Transmission Ratio

The aim of the transmission ratio strategy is to decrease the torque that hydraulic cylinder needs to provide in order to support the necessary torque to keep the trough stationary at harsh conditions. The strategy is implemented using a hydraulic cylinder actuating an intermediate stage that then drives the solar trough wheel. The ratio between the diameter of the intermediate stage and the solar trough wheel can be used to provide the transmission ratio needed to use the existing hydraulic cylinders as illustrated in Figure 8. The strategy was the result of trying to reuse the hydraulic cylinders already employed in the old design and reduce the cost of the necessary system alteration.

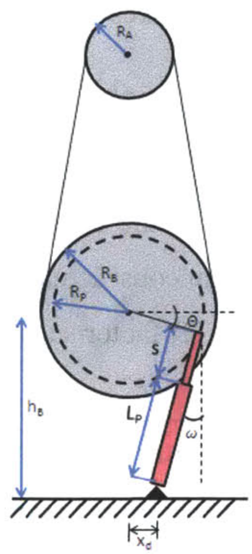


Figure 8: Transmission ratio schematic diagram

3.1.1. Hydraulic Cylinder Placement

The first step was to determine the optimum point of ground attachment of the hydraulic cylinder with respect to the lower driving pulley center of rotation (x_d). Since hydraulic force is always at an angle with respect to the moment arm (R_p), resulting toque (τ_p) has a parabolic shape dependent of ω as shown in Equation 7

$$\tau = F_p R_p \sin(90 - \omega) = F_p R_p \cos(\omega) \qquad \text{Equation 7}$$

In order to take advantage of the upper torque range torque needs to be maximized which happens when cos (ω) = 1 or equivalently

$$\sin(\omega) = \frac{R_p - x_d}{L_p + s} = 0 \qquad \text{Equation 8}$$

Cylinder was placed so that the $x_d = R_p$. As such, torque magnitude at the point of minimum and maximum stroke extension is equal.

3.1.2. Transmission Ratio Optimization

Transmission ratio is defined as a ratio between the driving (R_b) and the driven (R_a) pulley. Consequently, the required angle in order to achieve $\pm 110^0$ rotational motion is such that

$$\theta = \frac{110}{TR} \therefore TR = \frac{R_b}{R_a} \qquad \text{Equation 9}$$

Due to the particular stroke length of eighteen inches and bore size of two inches, the hydraulic force (F_p) is constrained. Minimum torque applied is depended on both the hydraulic force as well as maximum angle that piston makes with the lever arm (R_p)

$$\tau_{min} = F_p R_p \sin\left(\frac{\pi}{2} - \theta\right) \qquad \text{Equation 10}$$

Consequently, safety factor is a ratio between the torque supplied by the torque delivered by the piston (τ_p) and the required torque at the driving pulley (τ_b). Note that the first one is twice the minimum torque due to simultaneous actuation from both ends, while the required torque at the lower pulley is multiplied by TR factor such that

$$SF = \frac{\tau_b}{\tau_p} = \frac{TR \cdot \tau}{2\tau_{min}} \qquad \text{Equation 11}$$

Solving this optimization problem by running a raster in MATLAB range of solutions was acquired (Figure 9).

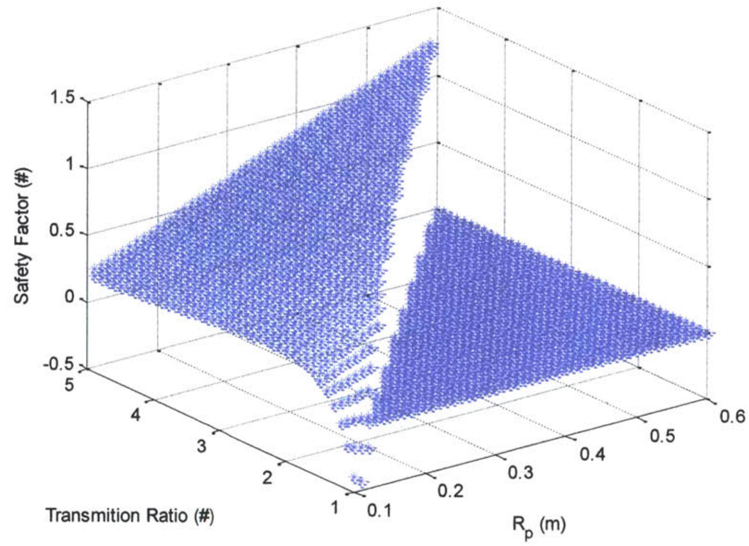


Figure 9: Safety Factor as a function of transmission ratio (TR) and delivered torque (τ_b)

All the zeros on the above plot represent cases in which the cylinders cannot achieve desired sweep angle in order to deliver necessary rotational motion to the trough as a result of defined cylinder stroke length, and at which the critical angle is achieved. Critical angle of 180^0 at which the cylinder will pass through the point at which hydraulic force is aligned with the moment arm is undesirable because of zero torque failure mode out of which cylinder could get out. Given relations and possible solutions can be better observed on the contour plot (Figure 10).

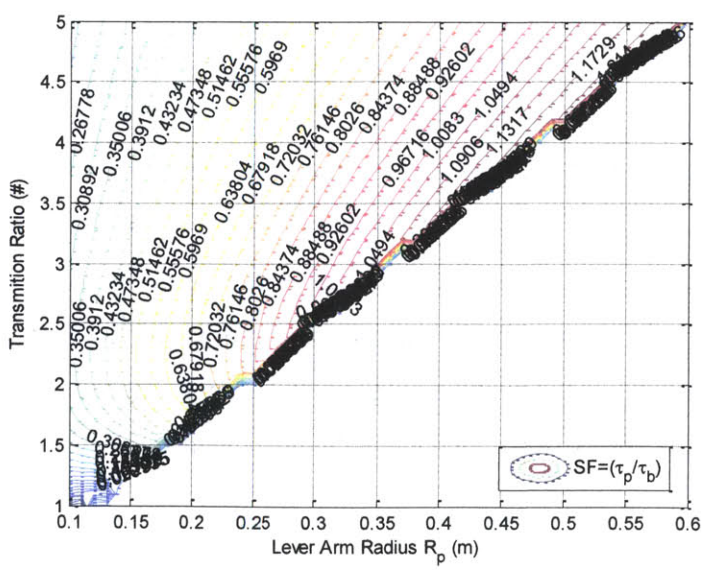


Figure 10: Safety factor contour plot as a function of lever arm radius and transmission ratio.

Even though this transmission design can provide as much as 10,000 Nm, as it can be seen from the contour plot above, the safety factor of this system barely approaches value of 1.2. This is result of increased torque needed to actuate the solar trough due to the transmission multiplication factor. Because of this feature, as well as more complex system compared to the rest of the concepts we came up with, this design was excluded early in the process.

3.2. Double Cylinder

The double cylinder strategy is aimed at decreasing the complexity of the overall system by eliminating the intermediate stage; however, the design does require the use of two hydraulic cylinders. As such, only one pulley driven by two cylinders on each side can be used to actuate the trough, as shown in Figure 7b. The cylinder on the right provides necessary force for clock wise rotation of the trough, while the left one performs the counter clockwise motion.

Although, using the single action single rod cylinders reduces the complexity of the system, there is a disadvantage. The efficiency of the unit is decreased since the push and pull forces differ based on the cross sectional areas on each side. Additionally, single action single rod cylinders have only one side pressurized while in use, thus engaging only one portion (pull) of their cycle. Since the system requires rope under constant tension, and in order to simplify the hydraulic system network and necessary controls needed in order to actuate both pistons simultaneously. Due to the complexity of the controls, another course other courses of action (e.g. dual action double rod cylinders) are examined.

Using dual action double rod cylinders has its own challenges, one of which being the attachment of cylinders to the existing structure. There are many, out of which at least two possible mounting configurations appropriate for our system, as shown in Figure 11. Configuration on the left (Figure 11a) uses mid trunnion mounts which allow cylinder rotation in one axis depending on the side or front attachment. The one on the right (Figure 11b) uses extended steel tubing attached to the piston body that is supported by rod and pinion. In this case there is only one degree of freedom but the attachment point is shifted from the piston to the steel tube fitting. This is more cost effective due to the generic mount; however, unnecessary components increasing assembly time are being added to the system.

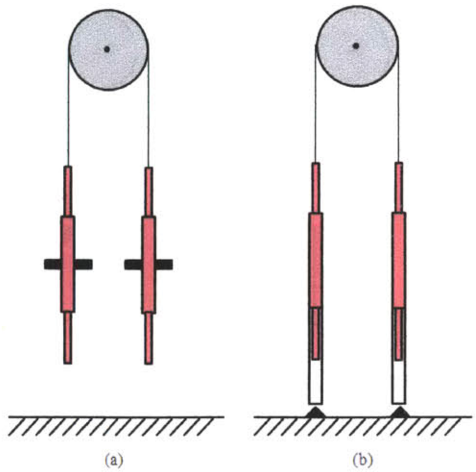


Figure 11: Double Cylinder Mounting Configurations.

Both of the attachments are capable of withstanding the installation misalignment of cylinder position with respect to the pulley. The only difference is that trunnion mounting is less common and thus more expensive than rod and pinion mounting. Additionally, later configuration can have an additional hollow steel tube that would be fixed to the cylinder, allowing needed space for rod extension while employing more economical mounting option.

3.3. Single Cylinder

The third strategy, single cylinder design (Figure 7c) has the least number of parts and has a simpler control method. Due to the continuous rope path, the design utilized the dual action double rod to its full extent. While the upper chamber is pressurized the trough motion results in clockwise rotation, and lower chamber similarly controls the counter clockwise rotation.

As well as in the previous strategy, there are two mounting options as shown in Figure 12, (a) trunnion and (b) nose bracket. Once again, trunnion is capable of compensating for installation misalignment between the two pulleys, the nose mounting bracket was not, but the price difference is almost half.

Out of three presented options, this design was the simplest when hydraulics controls were taken into consideration.

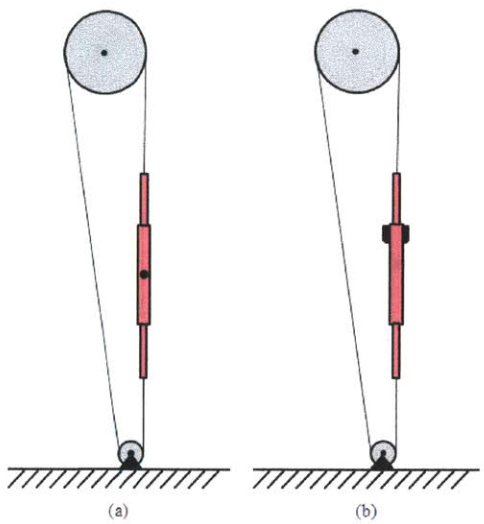


Figure 12: Single Cylinder Mounting Configurations

3.4. Strategy Evaluation

The three strategies with their respective mounting configuration are compared in Table 1 to determine the course of action based on meeting the functional requirements for the system design. The transmission system had the lowest torque as well as range flexibility. It also requires a great deal of adjustment to the existing structure. The double cylinder rod and pinion extended mount overall performance is satisfying except the unnecessary additional components and increased assembly complexity.

The three designs left ranked equally. However, no alignment capability present in the nose bracket design was determining factor. After considering several hydraulic system architectures we have decided to conduct in-depth cost analysis into two of them, single and double trunnion mount options.

Table 1 Strategy evaluation based on the functional requirements

Strategy	Torque	Range	Complexity	Interface	Total
Transmission Ratio	-1	-1	+1	-1	**-2**
Double Cylinder - Trunnion	+1	+1	0	0	**2**
Double Cylinder - Extended	+1	+1	-1	0	**1**
Single Cylinder - Trunnion	+1	+1	0	-1	**2**
Single Cylinder – Nose Bracket	+1	+1	0	0	**2**

4. Performance & Cost Comparison

Due to the higher safety torque requirement factor, the single cylinder trunnion and the double cylinder trunnion designs were chosen for further work based on the strategy evaluation.

A more detailed cost estimate of the two strategies is shown in Table 2. The most expensive unit of both designs is dual action double rod hydraulic actuators and the respective trunnion brackets. However, the double cylinder has two of each while the single has only one. This at the first look minor difference increases the overall cost of the system significantly. This difference multiplies as the system scales up.

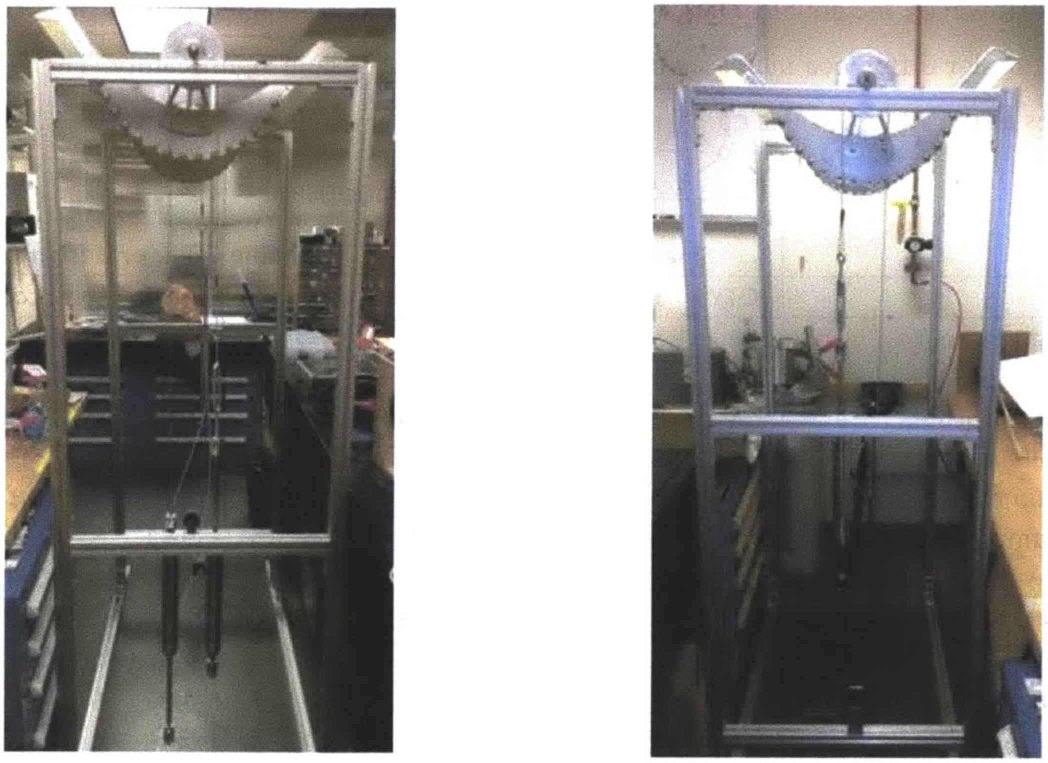
Table 2: Small Scale Cost Comparison

	Single Cylinder Design			Double Cylinder Design		
Part	**QTY**	**unit price**	**total cost**	**QTY**	**unit price**	**total cost**
Capstan pulley	1	$ 50.00	$ 50.00	1	$ 50.00	$ 50.00
Idler pulley	1	$ 10.00	$ 10.00	0	$ 0.00	$ 0.00
nuts/anchors	1	$ 10.00	$ 10.00	1	$ 10.00	$ 10.00
wire rope	4.7	$ 0.87	$ 4.09	11	$ 0.87	$ 9.57
Turnbuckle	1	$ 16.37	$ 16.37	1	$ 16.37	$ 16.37
compression sleeves	1	$ 9.31	$ 9.31	1	$ 9.31	$ 9.31
4way valve	1	$ 133.00	$ 133.00	1	$ 133.00	$ 133.00
pneumatic cylinders	1	$ 89.49	$ 89.49	2	$ 89.49	$ 178.98
piston rod clevis	4	$ 6.28	$ 25.12	4	$ 6.28	$ 25.12
Nose mount bracket	2	$ 3.87	$ 7.74	2	$ 3.87	$ 7.74
Speed Muffler	2	$ 3.56	$ 7.12	2	$ 3.56	$ 7.12
1/4" tube fitting	3	$ 1.84	$ 5.52	3	$ 1.84	$ 5.52
1/4" T fitting	2	$ 3.89	$ 7.78	2	$ 3.89	$ 7.78
1/8" tube fitting	3	$ 1.87	$ 5.61	3	$ 1.87	$ 5.61
Polyethylene tubing	1	$ 13.29	$ 13.29	1	$ 13.29	$ 13.29
Grand Total		$ 394.44			$ 479.41	

While the single cylinder design is the more cost effective option both units are comparable in price, due to the very similar components list, as well as overall actuating mechanism. The team chose to have both prototypes built in order to put them under performance evaluation testing to determine the future course of action.

5. Proof of Concept Prototype

Using the fiber glass solar trough that was part of Precision Engineering Research Group (Figueredo, 2011), our group has installed double cylinder design on one and single cylinder design on the other end of the structural support as shown in Figure 13. The main purpose behind this small scale model besides evaluating assembly difficulty level, and overall performance of the systems is ability to control the speed of cylinder actuation.



(a) (b)

Figure 13: Small Scale Model (a) Double Cylinder Design (b) Single Cylinder Design.

5.1. Design

The main structure of the small scale model is made out of 80/20 aluminum extrusion for easy of prototyping. The cylinders used are dual action double rod pneumatic actuators. The pneumatic actuators were chosen over hydraulics due to easier implementation and readily available air hose pressurized air in the lab setting. Main component used in actuating cylinders was manual air control five way three position valve. Speed control was achieved by utilizing generic speed mufflers that controlled the air flow supply by adjusting the opening using the screw.

Since there was no external disturbances and as such no torque requirements, all assembly components were designed for possible misalignment of the center shaft and necessary torque due to the weight of the trough itself.

5.2. Testing

Both designs were tested independently of each other. The opposite sides of trough had all the components permanently installed and were controlled by separate valves and speed muffler pairs. While the pressure rating did not matter in this stage of testing, all the tests were run at about 80 psi pressure. However, designs were sensitive to the air flow intake; different performance results were recorded at different flow intakes.

5.3. Results

Double cylinder design has proven to be quite complex when assembly took place. Placement of the cylinders at the exact stroke length with respect to the trough angular position was rather tedious process. In addition, simultaneous actuation of the cylinders happened only at the high enough air flow intake. The speedy extension and retraction of the pistons caused slack in the wire rope and thus slippage at the pulley, the exact effect that we are trying to control.

On opposite end, single cylinder design is less difficult to set up and almost self-aligning. Also, because of the single cylinder, the control was not a problem and the desired actuation speed was achieved. As a result, the wire rope was taught all the time and there is no slippage present. The trough consistently achieved same angular position even after extended period of actuation.

6. Full Scale Prototype

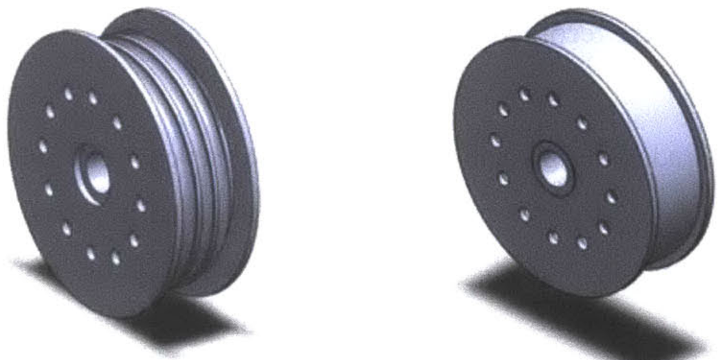
Based on the results from the small scale testing, together with the cost benefits, the single cylinder design is the best strategy to implement it in a full scale model.

6.1. Sizing Considerations

Our actuation design was divided in three main sections: pulley, wire rope and accessories, hydraulic actuators, and respective controls.

6.1.1. Capstan Pulley

The 12in pitch diameter capstan pulley, designed by a teammate Aaron Flores, is intended to hold the tension of the rope employing capstan effect. The helix allows for 900^0 of contact in addition to necessary clearance which takes into consideration wire rope diameter. The lebus grooved design was chosen for decreased wear and tear of both pulley and the wire rope. In addition, as wire rope slippage prevention during both installation and operation, our design employed end plates (Figure 14). However, due to the machining cost, we have opted for a simplified pulley design, consisting of circular tubing and without he grooves. Lastly, the pulley is bolted to a tee shaped plate with six grade 5 SAE ½in bolts in circular 8in diameter pattern. This method gives safety factor of four and was based on the mounting space availability and the torque requirements.



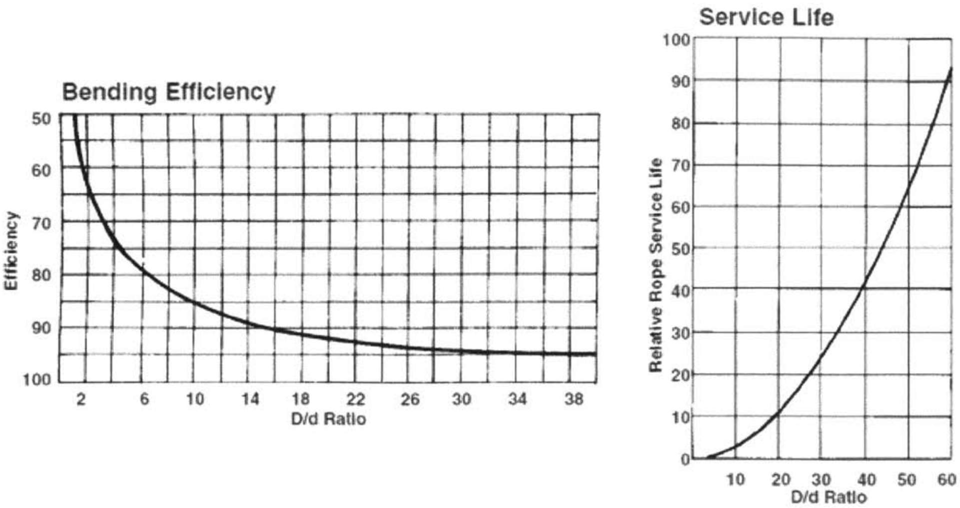
(a) (b)

Figure 14: Capstan pulley designs (a) grooved lebus (b) flat winch.

6.1.2. Wire Rope

The system goes through two cycles a day, first when the trough is tracking the sun, and the second when it is being retracted to its initial position for the next day sun tracking. Two times a day, every day of a year, for thirty years adds up to about twenty two thousand cycles throughout the expected service of the system As such, the life cycle of components is of great importance.

There are two driving factors that were taken into consideration for wire rope selection: (a) driving pulley radius, and (b) expected tension due to torque requirements (Figure 15). In order to maximize bending efficiency of the wire rope, driving pulley diameter to wire rope diameter ratio was kept above the minimum requirement of 24, while in order to achieve life expectancy of about 25 years the ratio has to be 30 or more. In addition, due to repetitive cycling of the rope, the breaking strength was determined by industry recommended safety factor of 5 with respect to the expected working load.



(a) (b)

Figure 15: Wire rope specifications - function of driving pulley/wire rope diameter ratio (Installation, Operation and Maintenance Recommendation)

The chosen wire rope for a 12in pulley was a 3/8 in diameter 7x19 constructions steel wire rope, giving a ratio of 30. Besides the importance of diameter ratio driving factors, this specific construction was chosen due to the availability of strong end grip fittings (discussed in the following section) designed for lifting that are not as readily available for the generic 8x19 structure usually employed in the pulley systems.

6.1.3. Wire Accessories

For wire rope end fittings as well as turnbuckle selection, working load was main selection guidance. Strong grip end fittings were chosen over a regular compression sleeve technique because of (1) their ultimate strength capabilities to withstand 100% of wire rope breaking strength, as well as (2) inspection hole that allows for proper installation result.

Turnbuckles selection is on contrary rated depending on their workload due to already built in proof load of two and a half, and ultimate load of five safety factors. The high strength corrosive resistant closed body turnbuckles were chosen due to enclosed threads and protection of the debris and particle dust.

6.2. Hydraulic Cylinder Specifications and Controls

As a group we have decided to go with dual action double rod cylinders with trunnion as a mounting option. Double rod feature decreased the complexity of the controls systems and necessary valves needed to actuate the system. The trunnion mounting option has been chosen to compensate for installation misalignment of pulleys giving the cylinder flexibility to align with the rope. For more details and in depth analysis and selection process for the hydraulic system and controls used in the final design please refer to my team mate thesis (Carillo, 2013).

6.3. Assembly

The assembly process should follow similar steps as the small scale model. Components such as driving pulley, idler pulley, as well as mounting of the hydraulic cylinder and all the necessary hydraulic connections should take place first. When installing the wire rope and turnbuckle it is recommended to fully extend the turnbuckle before making all the right connections. The tensioning would come only after the cylinder is engaged. While that is taking place, constant eye should be kept on the solar trough alignment with respect to the stroke extensions.

6.4. Expected Results

It is expected that the performance of the full model would not defer much at all if any from the small scale prototype results. Due to the size of the system, extra precaution should be however taken when the fine tuning of hydraulic system as well as tensioning of the rope takes place.

7. Full Scale Prototype

While we did not have time to build our design on the prototype in New Hampshire, the integrated system can be seen in the solid works rendered model, see Figure 16, done by my teammate Aaron Flores.

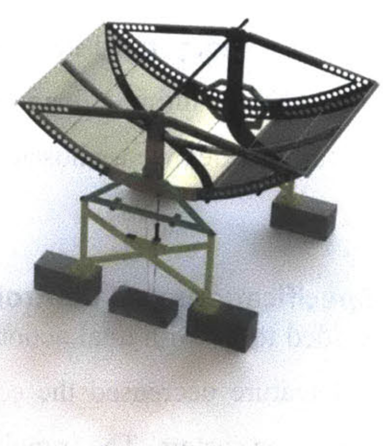


Figure 16: CAD of actuation on 4 m scale prototype.

7.1. Design Advantages

The design has addressed the major flaws of the existing hardware. First of all the hydraulic force is always being applied at the 90^0 angle at the lever arm (pulley radius) and the torque is being maximized throughout the cylinder full stroke movement. Consequently, the driving pulley is mounted between the trough and the structural support in double shear and as such, the deflection of the auxiliary shaft is minimized (Figure 17).

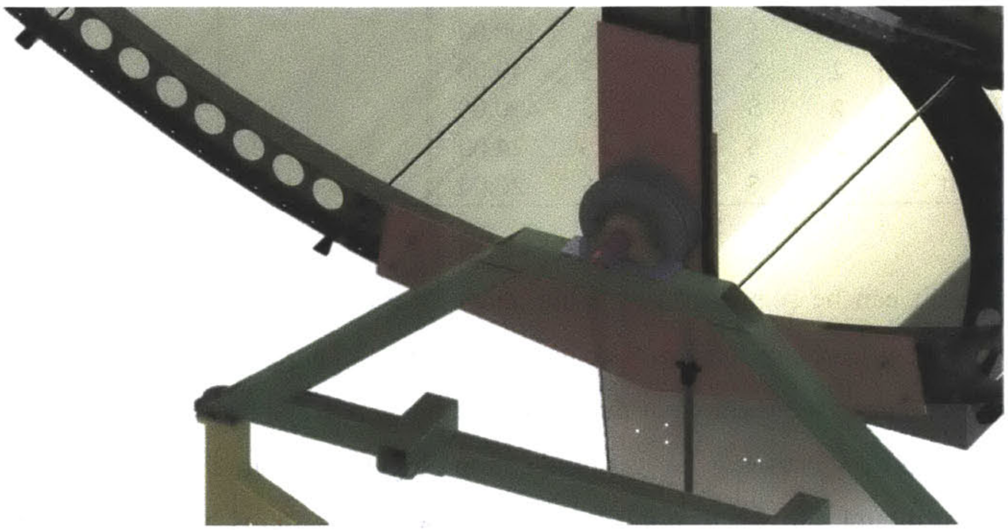


Figure 17: Close up of the driving pulley.

Finally, due to efficient usage of actuator driving power and the hydraulic cylinder dual action feature there is no need for a secondary cylinder. This greatly reduces the overall control systems in both aspects of overall parts number and consequently the price of the system.

7.2. Cost Comparison

The overall cost of the system is very important as it is directly related to our sponsor's profits. The rough cost comparison (* without cost estimate for control system) between the existing design and the one we have presented in this thesis shows that our presented design is almost twice as expensive as the present system (Table 3).

Table 3: Full Scale Cost Comparison

	Single Cylinder Design (NEW)				Crank Lever Arm Design (OLD)		
Part	**QTY**	**unit price**	**Total cost**	**Part**	**QTY**	**unit price**	**Total cost**
double rod cylinder	2	$ 789.00	$ 1,578.00	cylinder	4	$ 250.00	$ 1,000.00
Rod end clevis	4	$ 112.00	$ 448.00	spherical rods	4	$ 40.00	$ 160.00
cylinder support	2	$ 30.00	$ 60.00	roller bearing	4	$ 20.00	$ 80.00
Turnbuckle	2	$ 70.71	$ 141.42	clevis attachments	4	$ 20.00	$ 80.00
wire rope	50	$ 2.79	$ 139.50	crank shaft	2	$ 200.00	$ 400.00
eye fittings	8	$ 37.90	$ 303.20				
trunnion mount	2	$ 50.00	$ 100.00				
idler pulley mount	2	$ 20.00	$ 40.00				
capstan pulley	2	$ 200.00	$ 400.00				
idler pulley	2	$ 12.00	$ 24.00				
idler pulley shaft	2	$ 10.00	$ 20.00				
Grand Total		$3,254.12*			$1,720.00*		

As it can be seen from the table above, the mechanical part of our system consists of more parts and as consequence costs more. The main price difference present is in the hydraulics. Our system employs dual action double rod cylinder and uses trunnion mounting while the old design uses generic single action single rod hydraulic actuator with common rod and pinion mount. However, due to this difference our systems hydraulic controls cut forth of valves necessary to actuate the system and we are hoping that this reduction in complexity will bring our design au par with the old one.

One thing to note is that two weeks before the completion of this project our functional requirements for torque have decreased significantly. This is of importance because of the system was initially design for a much higher level of operation that the existing design could not support and as such the increase in cost was justified.

8. Conclusion

While the cost reduction of the system was not achieved, our present design choice should be au par with the existing one. However, the important thing to note is that it single cylinder design addresses the current flaws present in the NH system. In addition, the major improvement that our design has over the existing one is reduction in the control network for actuating the hydraulic cylinders. Decreasing the number of hydraulics actuators in half, the overall controls components have been cut in four. By having a minimally complex system, the overall possibility of failure is decreased within the hydraulic components.

9. Future Work

Besides in depth cost analysis on the hydraulic controls and a possibility of slight reduction in hydraulic actuators depending on the manufacturer as well as distributor, there are always improvements that can take place to reduce the cost and increase overall system performance.

First of all different mounting option should be looked into such as rod and pinion coupled with the appropriate steel sleeve fitting that would press fit onto the cylinder. Furthermore, it would be good to revisit the first transmission design option. While this design has major failure more where there is a zero point motion at which hydraulic force would align with the lever arm and system would be stalled, addition of safety stop in case of extreme conditions could be installed. Lastly, due to decrease in torque requirements, an electric motor might be a viable design to look into.

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