

Mechanical Development of an Actuation System for a
Parabolic Solar Trough Collector

by

Juan Felipe Carrillo

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

Bachelor of Science in Mechanical Engineering

at the

Massachusetts Institute of Technology

June 2013

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Abstract

This thesis documents my personal contribution to the development of a hydraulic-based actuation system for a solar trough collector. The goal of this project was to design the actuation system using hydraulic actuators for a four meter solar collector prototype in Pittsfield, New Hampshire. After considering several hydraulic system architectures and conducting in-depth analysis into two of them, the idler pulley scheme was chosen. This mechanism uses a double rod end hydraulic actuator connected to wire rope wrapped around a capstan drum and an idler pulley. The model was optimized for mechanical performance, and it is expected to be a more cost effective option than the existing actuation system in New Hampshire once the controls equipment required to actuate the hydraulic cylinders for the new design is specified.

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Acknowledgments

I would like to thank Professor Slocum and Professor Campbell for their valuable support and guidance throughout this project. I would also like to thank Nevan Hanumara for ordering parts and ensuring that we had all of the resources needed to work. Finally, I would like to thank my teammates Aaron Flores and Katarina Popovic for their support and contributions to this project which would not have been possible without their help.

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I. Introduction

1.1 Goals

The goal of this project is to design, evaluate, and price a cost effective actuation system that eliminates the flaws (outlined in section 1.3.4) associated with an existing actuation system of a four meter trough prototype in New Hampshire while satisfying the required functional requirements.

1.2 Functional Requirements

The functional requirements for the actuation system were set in place by the Italian company ENI, the sponsor of this project. The requirements can be broken down as follows:

1) Rotational Range

The solar troughs are being designed so that they can be installed in a hot desert climate under severe weather conditions. ENI requires the solar trough to be able to rotate $\pm 110^\circ$ such that they can face away from strong winds during inclement weather and prevent any damage.

2) Prototype Interface

There is a full-scale, four meter wide by twelve meter long solar trough prototype in New Hampshire. The existing actuation system on this prototype has several drawbacks and must be replaced. Once a final design for the new actuation system has been established, a full-size version will be built so that it can be tested for robustness with the New Hampshire prototype.

3) Service Life

This is a requirement established by the sponsor. Given that these solar troughs will be installed in desert conditions and will be exposed to harsh environmental conditions, it is important to choose the best materials against corrosion and abrasion to ensure that the actuation system will last a long time. Particular attention is placed on the wire rope selection to ensure it will endure environmental conditions and load requirements.

4) Design Cost

As this project is sponsored by a private company for commercial purposes, the design must be cost competitive. This means that the new design must be on par or below the cost of the prototype in New Hampshire. To make this project financially attractive, the cost must be close to \$2000. [1]

5) Actuation

The solar troughs being designed have very low torsional rigidity. It is estimated that the trough will deflect 16 times less if it is driven from both sides as opposed to a single side. Therefore, they must be driven from both ends to prevent any damage from torsional forces. [1]

6) Torque Rating

Collaboration with a group of ENI at the Polytechnic Institute of Milano determined a maximum torque from wind load based on a computational fluid dynamics model. Their analysis concluded that the actuation system needs to hold a maximum torque of 1500 Nm.

1.3 Existing Design

1.3.1 Background

When the team inherited the project in February 2013, substantial work had been done on the production of the solar troughs. In particular, there was analysis done on the production of the mirror surfaces, fabrication methods, and materials. Noteworthy was the decision to use rolled steel plates coated with a reflective film that perform nearly as well as glass panels but at a fraction of the cost. [1]

1.3.2 Solar Trough Structure

The current solar trough consists of coated rolled steel plates supported by two bulkheads on each side. Each bulkhead contains a steel end plate where the existing actuation system

attaches to. Truss supports attached to cement blocks support the solar collector on each side. Figure 1 shows the solar trough as well as the existing actuation system.



Figure 1: Solar trough collector with existing actuation mechanism

1.3.3 Actuation System

The existing actuation system consists of two, 2-inch-bore cylinders with a stroke length of 18 inches per trough end for a total of four single action hydraulic cylinders. These cylinders are attached to the trough via a crank plate which is connected to the trough by a crank shaft. Figure 2 shows a zoomed in view of the actuation mechanism.



Figure 2: Existing actuation system. Two single rod cylinders are attached to a crank arm connected to the solar trough collector

Although the current architecture of the mechanism is very simple, there are several geometric factors that prevent simultaneous use of both cylinders; at any given time, only one cylinder is exerting a force while the other is floating. The cylinders also include linear resistive transducers (LRTs) which are used for coarse tracking when the sun is not out. When the sun comes out, the fine tracking is driven by solar sensors. As previously stated, the baseline system is relatively simple mechanically, but it does have complexity in the wiring of the hydraulic system.

1.3.4 Design Flaws

There are numerous problems with the existing hydraulic actuation system in New Hampshire. The force applied by the cylinders on the crank arm, or moment arm, is always at an angle other than at 90° . This results in torque reduction and the need to oversize the cylinders to accommodate for the applied wind load torque required to hold the trough in position. In addition, two different cylinders must be used independently to move past a point of zero motion due to the geometry of the design. Finally, the shaft connecting the crank arm to the trough deflects considerably as the point of application of the force by the cylinders is far from the shaft attachment point on the trough.

II. Conditions of Modeling

2.1 Torque

The net wind load torque the solar trough experiences is of vital importance to the overall design of the actuation system. A static system model is used to determine the net torque required by the actuation system at any position to prevent the trough from rotating about its pivot point. Figure 3 illustrates the geometric analysis of the trough.

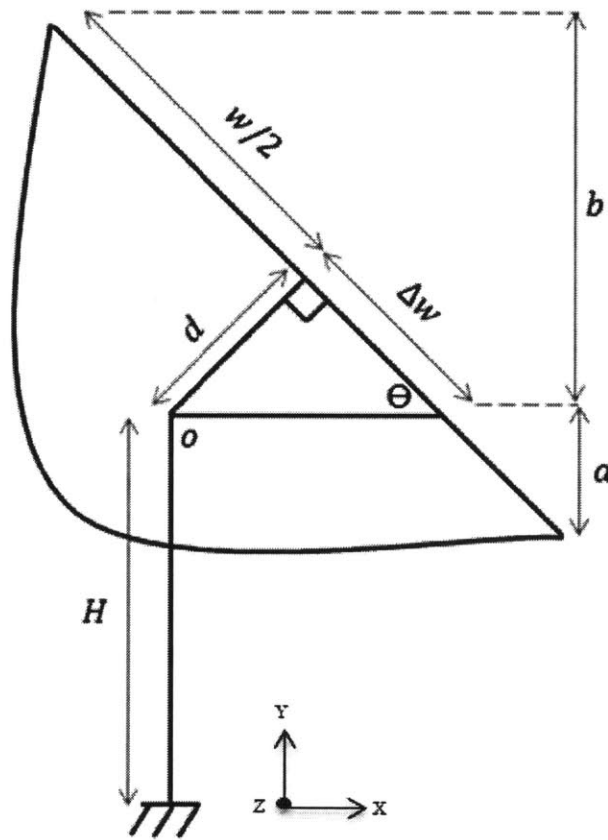


Figure 3: Schematic representation of solar trough rotating about center of rotation, O
In the figure above, w is the width of the trough, Θ is the angle the surface plane makes with respect to the horizontal, d is the distance from the surface plane of the trough to the center of rotation, H is the height from the ground to the center of rotation, and a and b are the projected areas of the bottom half and top half of the trough, respectively and are given by

$$a = \left(\frac{w}{2} - \Delta w\right) \sin\theta$$

$$b = \left(\frac{w}{2} + \Delta w\right) \sin\theta$$

To find the net moment about the center of rotation of the trough, assume a linear wind velocity profile with respect to the ground. The net torque about the center of rotation of the trough can then be expressed as

$$\tau_{net} = (y_b - H)F_b - (H - y_a)F_a$$

where y_b and y_a are the centroids of application of the forces on the upper and lower portions of the trough, respectively and are given by

$$y_b = \frac{.5\rho c_d \alpha^2 L \int_H^{H+b} y^3 dy}{.5\rho c_d \alpha^2 L \int_H^{H+b} y^2 dy}$$

$$y_a = \frac{.5\rho c_d \alpha^2 L \int_{H-a}^H y^3 dy}{.5\rho c_d \alpha^2 L \int_{H-a}^H y^2 dy}$$

F_b and F_a are the drag forces on the upper and lower portions of the trough, respectively,

$$F_b = .5\rho c_d \alpha^2 L \int_H^{H+b} y^2 dy$$

$$F_a = .5\rho c_d \alpha^2 L \int_{H-a}^H y^2 dy$$

where ρ is air's density, c_d is the coefficient of drag, and L is the dimension into the page. For the prototype in New Hampshire, the following parameters were used to define a torque graph with respect to angle position.

Table 1: Parameters used in wind torque model

Parameter	Value
ρ [kg/m ³]	1.2
c_d [dimensionless]	2
A	4.29
L [m]	12
H [m]	3.5

Figure 4 shows the moments resulting from the wind drag on the solar trough.

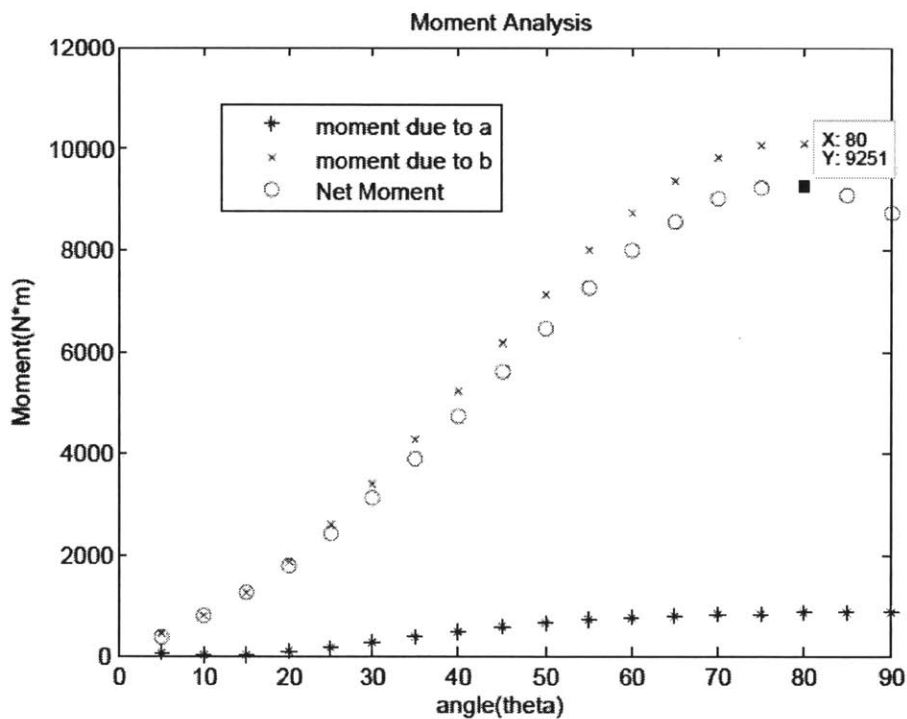


Figure 4: Moments on 4 meter parabolic trough prototype due to wind load. The moment on the projected areas "a" and "b" are shown as well as the net moment on the trough

As Figure 4 shows , the net moment on the solar trough due to wind load is 9,251Nm. This torque requirement was later revised as explained in section 4.2.

III Concept Development

3.1 Early Concepts

The team brainstormed several ideas for the actuation system from which two were selected for detailed analysis. These two architectures can be seen in Figure 5.

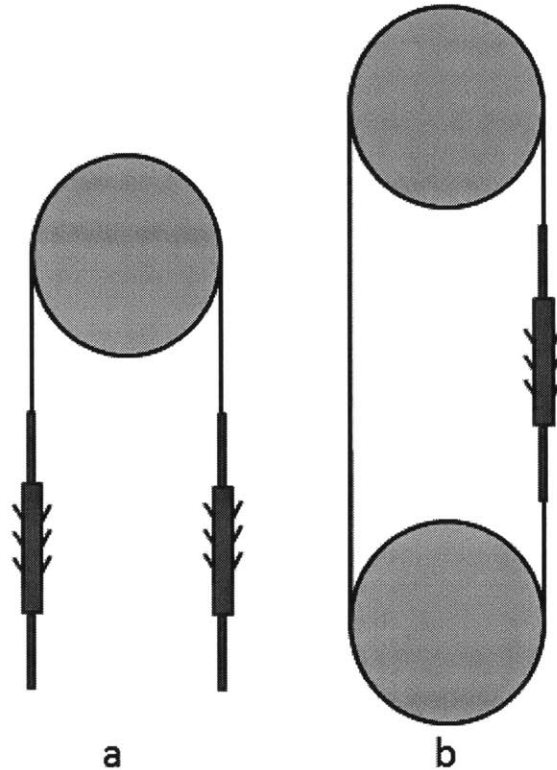


Figure 5: Two different architectures evaluated. (a) Single pulley (b) Idler pulley.

The single pulley architecture consists of a capstan drum that is attached to the trough. This capstan drum is driven by a wire rope whose ends are attached to two different double rod end cylinders that are fixed in position as seen in (a). Similarly, the idler pulley configuration consists of a capstan drum that is also attached to the trough. However, this design has an idler pulley below and in the same plane of the capstan drum. A wire rope whose ends are attached to a fixed double rod end cylinder wraps around both the capstan and the idle pulley and drives the system.

3.2 Small-Scale Model

To explore and analyze the two possible architectures, the team built a small-scale model of each configuration using a fiber glass trough built for a former phase of the project. An aluminum frame was constructed using 8020s to mount the fiber glass trough and on which the two different configurations were mounted, one on each side. Figure 6 shows the structure built to test both architectures.

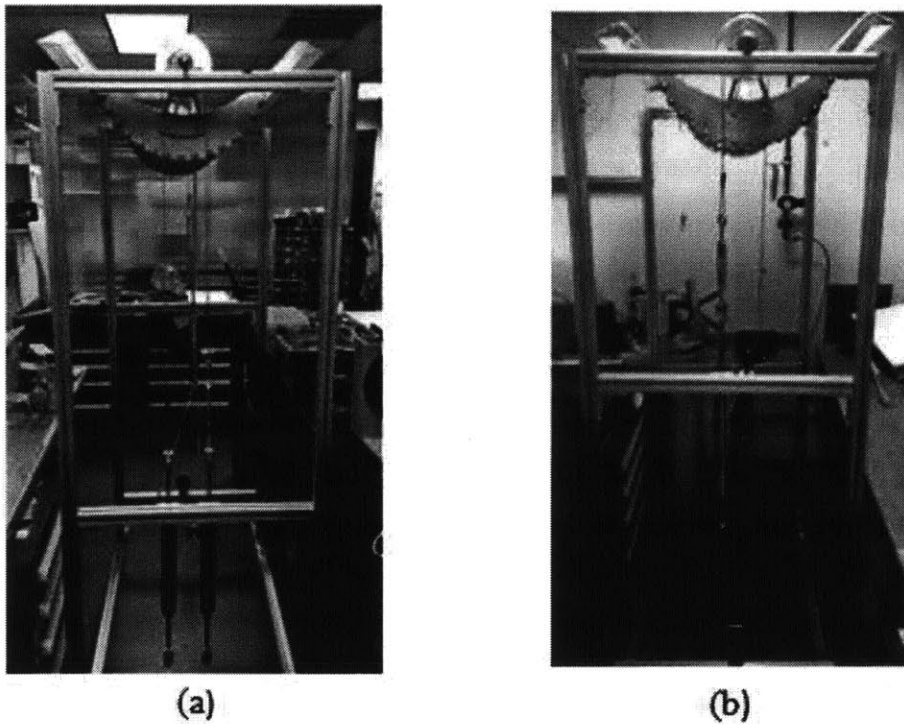


Figure 6: Small-scale model. Single pulley design (left) and Idle pulley design (right)

The results of this model not only helped the team understand the limitations of each design but also make a decision on which configuration was better option for the actuation system.

3.2.1 Single Pulley Architecture

The single pulley, two double rod end cylinder design showed a major drawback in its design. This drawback entails getting both cylinders to actuate simultaneously. This is a critical drawback as both cylinders must actuate together to prevent the wire rope to loose tension and slip from the capstan drum. When the wire rope slips, the actuation system is render useless. The team tried different pneumatic connection configurations between the cylinders to make them actuate simultaneously, but always one actuated before the other, , making simultaneous action of both cylinders extremely difficult..

3.2.2 Idle Pulley Architecture

The single cylinder, idle pulley configuration performed better than the single pulley architecture. However, one drawback found was that the idler pulley must align vertically with the cylinders and the capstan drum to prevent the idler pulley from bending and failing. Although problematic, this can be fixed by placing the supports of the idler pulley such that they are in the same plane as the cylinder rod and the capstan drum. Additionally, this design did not slip as the cylinder pulled the wire rope, making the trough rotate the required $\pm 110^\circ$.

3.3 Manufacturing Design and Costing

Cost was also considered apart from the mechanical performance of both architectures. Table 2 and Table 3 show the cost associated with each architecture.

Table 2: Single pulley small-scale model cost

Part	QTY	unit price	total cost
Capstan pulley	1	\$ 50.00	\$ 50.00
nuts/anchors	1	\$ 10.00	\$ 10.00
wire rope	11	\$ 0.87	\$ 9.57
turbuckle	1	\$ 16.37	\$ 16.37
compression sleeves	1	\$ 9.31	\$ 9.31
4way, 3 position valve	1	\$ 133.00	\$ 133.00
pneumatic cylinders	2	\$ 89.49	\$ 178.98
piston rod clevis	4	\$ 6.28	\$ 25.12
Nose mounting bracket	2	\$ 3.87	\$ 7.74
Speed Muffler	2	\$ 3.56	\$ 7.12
1/4" tube fititng	3	\$ 1.84	\$ 5.52
1/4" T fitting	2	\$ 3.89	\$ 7.78
1/8" tube fitting	3	\$ 1.87	\$ 5.61
Polyethylence tubing	1	\$ 13.29	\$ 13.29
		Grand Total	\$ 479.41

Table 3: Idler pulley small-scale model cost

Part	QTY	unit price	total cost
Capstan pulley	1	\$ 50.00	\$ 50.00
Idler pulley	1	\$ 10.00	\$ 10.00
nuts/anchors	1	\$ 10.00	\$ 10.00
wire rope	4.7	\$ 0.87	\$ 4.09
turbuckle	1	\$ 16.37	\$ 16.37
compression sleeves	1	\$ 9.31	\$ 9.31
4way, 3 position valve	1	\$ 133.00	\$ 133.00
pneumatic cylinders	1	\$ 89.49	\$ 89.49
piston rod clevis	4	\$ 6.28	\$ 25.12
Nose mounting bracket	2	\$ 3.87	\$ 7.74
Speed Muffler	2	\$ 3.56	\$ 7.12
1/4" tube fititng	3	\$ 1.84	\$ 5.52
1/4" T fitting	2	\$ 3.89	\$ 7.78
1/8" tube fitting	3	\$ 1.87	\$ 5.61
Polyethylence tubing	1	\$ 13.29	\$ 13.29
		Grand Total	\$ 394.44

IV. Final Design and Full Scale Four Meter Prototype

4.1 Selection of actuation architecture

Based on the results of the small-scale model testing and manufacturing costs, the team decided to choose the idler pulley architecture. This design proves to be superior to the single pulley configuration based on lower complexity, better controllability, and lower cost.

4.2 Part Design and Selection

Two weeks before the project was due, new CFD wind load torque results came from ENI. These results varied significantly from the expected wind load torque model results obtained in the torque modeling section 2.1. As opposed to an expected torque of about 10,000Nm, the CFD results showed a torque of 1,500Nm. This forced the team to drastically change the size of the architecture components to account for the new lower torque requirement.

In addition, the main factor driving the design of the actuation system was the ratio between the capstan drum diameter, D , and the wire rope diameter, d . This factor is important because the ratio D/d determines the life cycle of the wire rope. To extend the number of bending cycles of a wire rope, the minimum industry recommended standard for D/d is 20 [2]. Based on this constraint, the team chose a ratio of 24. This drove the selection of all other components of the actuation system.

4.2.1 Capstan Drum

The capstan drum for the actuation system has a diameter of 12.5in and will be manufactured with welded steel plates and pipes. The pulley has two side plates that prevent the wire rope from slipping off of the pulley during operation. The pulley will be attached to the solar trough in New Hampshire using a bolt pattern. Figure 7 shows a CAD model of the capstan pulley.

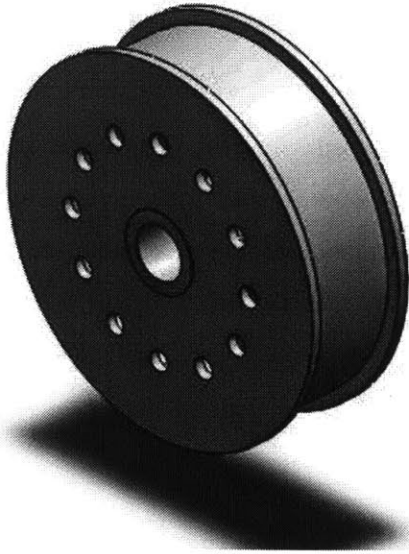


Figure 7: Capstan pulley for four meter prototype

For more detailed information about the capstan drum, please refer to the group report of this project.

4.2.2 Wire rope

The wire rope is composed of a 7x19 construction of diameter of 3/8in. The wire rope was selected to have a breaking strength of five times that of the working load. For more information about wire rope selection, please see the group report for this project as well as [Analysis and Design of Hydraulically Driven Actuation System of a Parabolic Solar Trough](#) 2013 MIT undergraduate mechanical engineering thesis by Katarina Popovic.

4.2.3 Hydraulic Cylinder Actuators

The specifications of the hydraulic cylinders resulted directly from the design specifications of the capstan drum. To exert a torque of 1,500Nm with a 12.5 in. diameter capstan, a force of about 9,500N is needed. To reduce costs, an off-the-shelf double rod end actuator that met closest the 1,500Nm torque requirement was chosen with the additional

requirement that it operate at 3,000PSI due to existing hardware in the New Hampshire that sets this operating pressure. Table 4 shows the specifications of the hydraulic actuator chosen.

Table 4: Double rod end cylinder specifications

Feature	Description
Pressure rating	high pressure
Fluid	hydraulic
Brand	Peninsular Cylinders
Type	dual action, double rod end
Bore diameter	1.5in.
Rod diameter	1.0in
Stroke length	23.5in
Rod end	clevis
Mount style	mid-trunnion

Although a nose mounting bracket was used in the small-scale model to attach the cylinders to the frame, a trunnion mount was chosen for the full-scale, four meter prototype in New Hampshire to accommodate for misalignment that could occur while mounting the capstan drum and idle pulley. This resulted in an additional cost of \$150 per cylinder. Given the above cylinder specifications, Table 5 gives the actuation system technical specifications.

Table 5: Actuation system technical specifications

Parameter	Description	Units
Pull force per cylinder	13101	N
Torque per cylinder	2079.8	Nm
System torque	4159.6	Nm
Torque requirement	1500	Nm
Safety factor	2.8	

The safety factor is 2.8, which could be on the high side considering the frequency that a torque of 1,500Nm is encountered. However, this safety factor resulted from choosing an off-the-shelf cylinder as opposed to using a custom made cylinder.

4.3 Actuation System Rendering

The new proposed redesign consists of a (1) capstan drum that is bolted onto the trough frame, (2) an idler pulley that is mounted in double shear at the bottom of the trough, and (3) ancillaries such as the steel cable, turnbuckle, and compression sleeves. The capstan drum that will be tested on the 4' x 6' New Hampshire trough will be manufactured with welded steel plates and pipes. The idler pulley is designed to be mounted in a double shear bracket on a concrete block underneath the hydraulic cylinder. Last, the steel cable is wound with 900° of contact around the capstan drum and attached through the idler pulley to both ends of the cylinder. Figure 8 through Figure 11 show the different components of the new design.

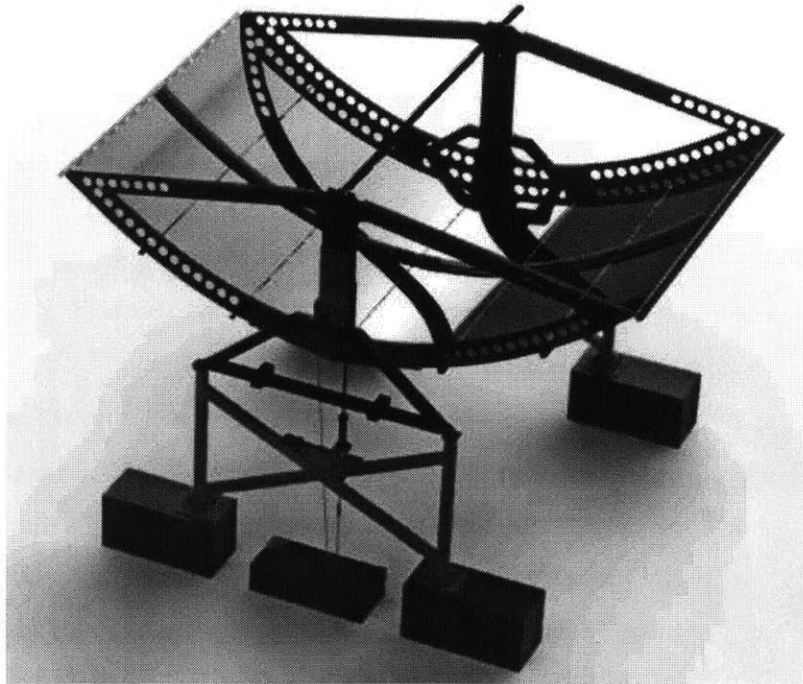


Figure 8: Rendering of actuation system installed on New Hampshire parabolic solar trough

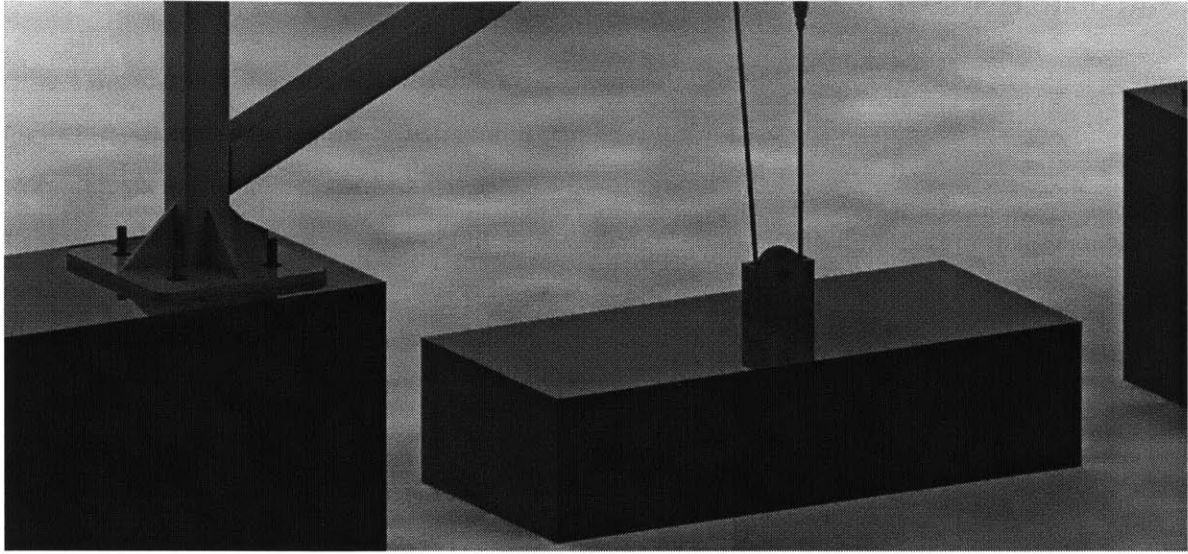


Figure 9: Idler pulley attached to cement block



Figure 10: Double rod end cylinder with trunnion mount

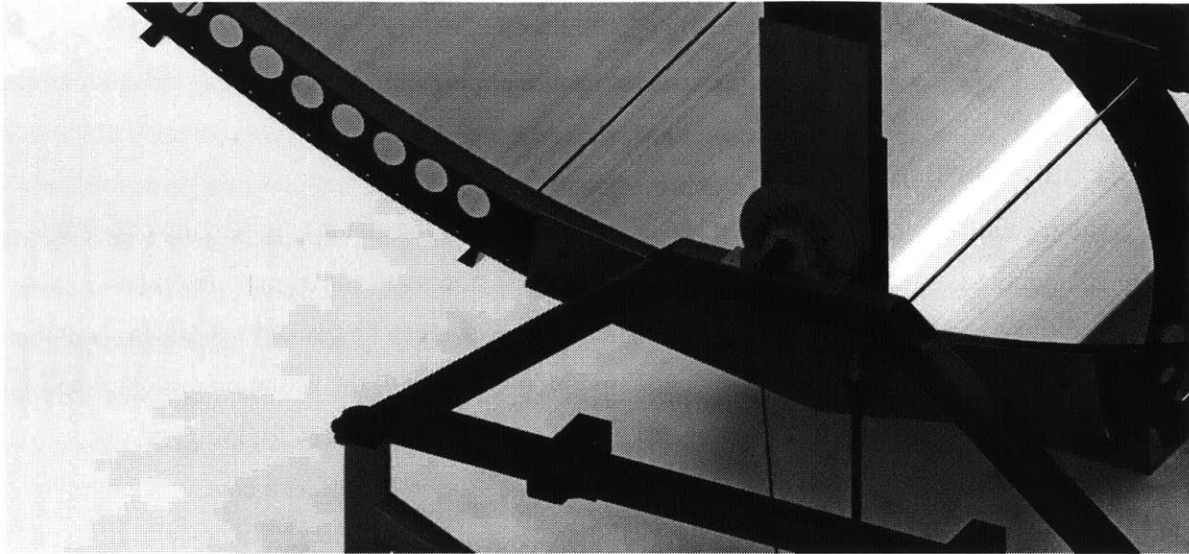


Figure 11: Capstan drum attached to solar trough collector

4.4 Cost Estimation

Based on the components selected for the current design, a cost estimate was compiled. Table 6 shows this estimated cost. Note that this does not take into account the controls necessary to drive the hydraulic actuators.

Table 6: Cost estimate of new actuation system design

Part	QTY	unit price	Total cost
double rod end cylinder	2	\$ 789.00	\$ 1,578.00
rod end clevis bracket	4	\$ 112.00	\$ 448.00
cylinder support to truss	2	\$ 30.00	\$ 60.00
turnbuckle	2	\$ 70.71	\$ 141.42
wire rope	50	\$ 2.79	\$ 139.50
eye fittings	8	\$ 37.90	\$ 303.20
trunnion mount	2	\$ 50.00	\$ 100.00
idler pulley mount	2	\$ 20.00	\$ 40.00
capstan pulley(12.5in)	2	\$ 150.00	\$ 300.00
idler pulley(3.90in.)	2	\$ 12.00	\$ 24.00
idler pulley shaft	2	\$ 10.00	\$ 20.00
Grand total		\$ 3,154.12	without hydraulic controls

V Hydraulic Controls

The parabolic solar troughs being designed must follow the sun throughout the course of the day at an optimal rate to generate electricity. Thus, it is necessary for the new hydraulic actuation system to integrate a control scheme such that the sun can be tracked continuously and reliably. One of the most important factors influencing the control of the actuation system is the flow of fluid through the cylinders. Section 5.1 explores the flow necessary for the parabolic solar trough to track the sun during normal operation and Section 5.2 offers a control scheme for delivering this flow rate to the cylinder.

5.1 Flow Calculation

The speed of the piston rod is given by

$$v = .5\Omega D$$

where v is the speed of the piston rod, Ω is the rotation rate of the earth around the sun, and D is the diameter of the capstan drum. The earth rotates around the sun 360° in 24 hours, the equivalent of 0.25° per minute. Substituting the capstan diameter from section 4.2.1 and the rotation rate of the earth into the above equation, the speed required for the piston rod to rotate the capstan attached to the solar trough and be able to track the sun is 0.0397m per minute. The volumetric flow rate required to provide this speed is given by

$$\vartheta = vA$$

where ϑ is the flow rate and A is the effective area of the cylinder. Using the cylinder specifications given in Table 4 and the speed calculated above, the required ϑ is $2.51 \times 10^{-5} \text{ m}^3$ per minute.

5.2 Control Scheme

The flow required to track the sun is considerably low and difficult to control mechanically. As a result, pulse width modulation should be implemented as a control scheme to

avoid using expensive and complex valves that could otherwise provide the required flow. The average flow rate, Y , using pulse width modulation is given by

$$Y = Tv$$

where T is the duty cycle and v is the peak flow. In this case, Y is equal to ϑ from section 5.1. To obtain v , the pump specifications must be known. The pump in New Hampshire used to drive the old design will be used for the new design. Table 7 gives the specifications of the pump that will be used.

Table 7: Existing hardware: pump with driving motor

pump brand	Oil dyne Model 156
rating	0.156 cubic centimeters /min
motor	synchronous AC @ 1725RPM
flow rate	2.69e-04 cubic meters/min

From the average flow rate equation and the information above, the duty cycle and cycle time of the new actuation system should be .093 and 60 seconds, respectively.

VI Future Work

There is considerable work left to be done for the actuation system to be completed and be implemented. The controls portion for the hydraulic actuators has not been addressed completely. The pulsation valve, along with other necessary hydraulic components, needs to be sized to effectively control the actuators to the desired rotational speed requirements. In addition, safety mechanisms must be implemented to prevent accidents from occurring in the event that the wire rope breaks. Finally, a study should be done on the economic significance of using a grooved capstan as opposed to a plain capstan suggested in this design. A grooved capstan increases the life cycle of wire rope at a greater initial manufacturing cost. It will be of interest to see if this cost increase reduces costs in the long run from avoiding to replace worn down wire rope.

V References

[1]Hanes Supply,INC. *Wire Rope 101*. N.p.: n.p., 2012. PDF.

[2]O'Rourke, C.R. (2011). *Mechanical Development of the Actuation System of a Parabolic Solar Trough*. B.S. Thesis. Massachusetts Institute of Technology: U.S.