Design of a Parallel Mechanism with Large Joint Clearances for Precise Motion

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ABSTRACT

A parallel mechanism consists of a platform connected in parallel to a base by several links and kinematical chains. The motion of these mechanisms is controlled by motors. The purpose of this project is to demonstrate the precise motion that can be achieved in parallel mechanisms despite large clearances in the joints. This project will involve the design of a parallel mechanism with joint clearances that determines the platform position and corresponding motor locations. It will also function to measure the forces in the links.

EXECUTIVE SUMMARY

Dr. Bamberger came to our group with a unique project. We were tasked to design and manufacture a planar parallel mechanism with large joint clearances. The large joint clearances will allow us to investigate if high accuracy platform positioning can be accomplished in the macro scale before an attempt is made in the micro scale. The scaled geometry of this mechanism is predetermined by Dr. Bamberger's research, which means that we are free to design this mechanism to whatever size necessary, as long as the geometrical scaling remains consistent.

The general setup of the mechanism includes a triangular platform, three parallel linkages and three linear motors. A 10 to 20kg constant force will be applied to the platform. Engineering specifications for the project include: the forces on the linkages must be measured within 3% of the force applied, the motor position must be determined to within 2% of the total motor displacement, the platform must have three degrees of freedom (x, y, θ), and the platform must not rotate about the x or y axes more than ½ degree.

The platform position was determined with an average error of 4.1% in the x direction, an average error of 3.0% in the y direction and an average error of 6.4% for Θ, rotation about the Z axis. The link forces were not able to be determined to within 3% of the force applied due to strain gauge sensitivity. All other engineering specifications were met. Detailed results will be discussed in the body of the report as well as recommendations for future work.

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PROBLEM DESCRIPTION

We were asked by Dr. Hagay Bamberger, Research Fellow, University of Michigan, Department of Mechanical Engineering, to design and manufacture a parallel mechanism to demonstrate that precise motion can be achieved despite large joint clearances. We were requested to model this on a large-scale so that a similar design can be incorporated into small-scale Microelectromechanical systems (MEMS) in the future. The mechanism will be required to measure the platform position and determine the location of the motors. It will also be required to measure the forces in the links.

BACKGROUND AND BENCHMARKING

In order to gain a better understanding for the motivation and challenges of our planar parallel mechanism project, we investigated the interests and work of our sponsor. Dr. Bamberger is an expert in dynamic systems, microelectromechanical systems (MEMS), and parallel robots. He has helped develop and patent a new six degree of freedom parallel robot for MEMS fabrication [1]. Due to current MEMS fabrication techniques, two dimensional shapes and planar geometries are created most successfully [2]. Consequently, a planar mechanism is a logical fit for MEMS. General design considerations include the type of robot, choice of actuators, and joint selection [2]. Our team was given customer requirements and specifications for the type of robot and actuators. Therefore, joint selection will be the primary design concern for mechanism functionality.

Similar Mechanisms

Current manufacturing techniques enable manufacturers to fabricate devices that have small tolerances. Consequently, joints can be manufactured with small clearances allowing them to operate with high precision. Even though there are other machines that can achieve the same movements as our proposed device, they are not relevant benchmarks because they are not affected by the challenges of large joint clearances. The majority of challenges due to large joint clearances can be seen in 18^{th} century machinery, which commonly suffered from rapid joint wear, vibrations, noise and shocks [3]. Significant improvements in macro manufacturing have been made since; however, the inaccuracies of 18^{th} century joints are still visible in joints with large clearances today.

Joints

One of the most important components of our planar linear mechanism is our joints. We must choose joints capable of producing large clearances as determined by our customer needs and engineering specifications. There are eight main types of joints: revolute, prismatic, cylindrical, helical, spherical, plane pair, gear pair, and cam pair as pictured in Figure 1 [4]. Each joint has a specific degree of freedom, translation, rotation, or other. The joint degree of freedom is broken down for each of the eight main types of joints in Table 1 [4].

Figure 1: Types of Joints [4]

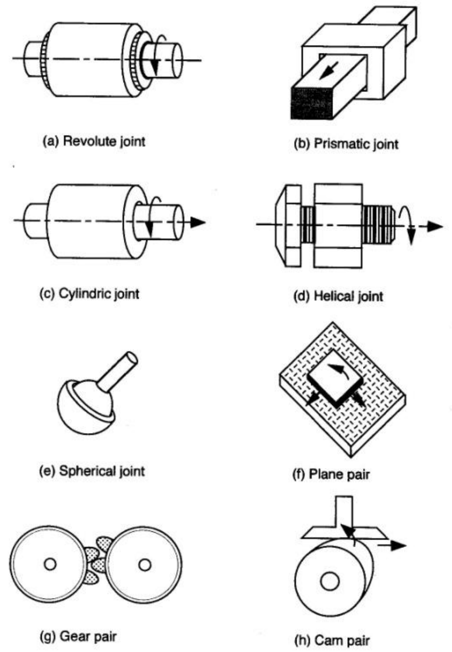


Table 1: Joint Degrees of Freedom [4]

Kinematic Pair	Symbol	Joint DOF	Rotational	Translational
Revolute	R	1	1	0
Prismatic	P	1	0	1
Cylindric	C	2	1	1
Helical	H	1	1	coupled
Spherical	S	3	3	0
Plane	E	3	1	2
Gear Pair	G	2	1	1
Cam Pair	C_p	2	1	1

For our project, we will be focusing on simple joints with one degree of freedom. This is due to the nature of the use of our planar parallel mechanism, MEMS research where fabricating revolute hinges is already well established [5]. For a planar mechanism with three degrees of freedom, we should assume that revolute and prismatic joints are most desired [4]. However, we need to take joint orientation into account for our design to achieve successful planar manipulation. All revolute joint axes must be perpendicular to the plane of motion [4].

Achieving the required joint clearances is possible with a revolute or prismatic joint when ordered to the correct specifications. The large joint clearances will allow us to investigate if high accuracy platform positioning can be accomplished in the macro scale before an attempt is made in the micro scale. Our primary obstacle in this investigation comes to this fundamental joint truth, "When clearance size is not suitable, not only displacement but also impact force will be caused, and as a result, system precision will be reduced [6]." Our planar parallel mechanism will explore this effect in more detail in the hopes of improving and better characterizing the platform's position accuracy.
Concurrently with designing the planar parallel mechanism, our team must develop a way to demonstrate platform position accuracy and measure link forces. To achieve this goal, we must investigate techniques to locate and track the platform's movement and measure forces.

Platform Locating
An increasingly popular way to track position is by the use of an accelerometer. Global Positioning Systems use accelerometers to track vehicle position when out of satellite contact, one example being near large buildings. The device operates similar to a damped mass attached to a spring. Since the mass and the force are both known at any point, instantaneous acceleration can be recorded. Integrating this acceleration twice over time will yield a displacement corresponding to a position. By increasing the sampling rate for the accelerometer, we can more accurately locate the platform's position with a Riemann Sum.

Accelerometers either contain one, two or three axes and these correspond to linear, planar and three dimensional measurements. For the purpose of locating a planar mechanism a device with two measurement axes would be required.

A fiber optics measuring device consists of a fiber optic probe containing a light source and a photo sensor. The light source is transmitted to the target and the reflection is received by the photo sensor. The photo sensor translates the reflected light to a voltage which is proportional to the distance between the fiber optic probe and the target. In order to obtain an accurate measurement, the probe must be within a few millimeters of the target [7].

Lasers can be used to measure distance by various methods including triangulation, time of flight, and the phase shift method [8]. In triangulation, a beam is pointed at an object. Using geometry, the distance to the object can be calculated by capturing diffuse reflections. The time of flight method calculates the time taken for a laser beam to bounce off of an object and return back to the originating device. This method is generally used for large distance measurement, but can be used for short distance with an accuracy of a few millimeters. The phase shift method sends a sinusoidal beam of light to an object and measures phase of the response. This method offers lower accuracy than the others for short distances [8].

The most basic method of platform location involves physical measurement. In order to use this technique, three permanent markers must be secured to the platform. The x and y positions as well as the platform angle, θ can then be measured using a grid or rulers near the platform or by recording the platform motion with a camera.

Force Measurement
It will be necessary to measure the reaction (tensile or compressive) forces on each of the three linkages that are connected to the platform. The appropriate choice will depend on the final scaling size of the mechanism, geometry and thickness of the linkages for mounting purposes. The following sections describe various types of force measurement equipment.

Force meter
A force meter uses a spring to measure the pushing or pulling force of an object. These devices are most commonly used as scales to weigh a suspended object. Another application is to measure the oscillation forces of an object in dynamic motion. The spring in the force meter could potentially affect the dynamic behavior of the mechanism.

Strain Gauge
These devices are used to measure the separation distance between two points located on an object. This information can be used to calculate force as well as pressure and acceleration. There are two main types; electrical and optical [9, 10].
Electrical strain gauges measure strain by changes in electrical conductivity. The gauge pattern is placed on an object so that the lengths of material are parallel to the direction of force. As the object elongates, the lengths elongate and increase the output resistance. These devices can be found in pressure

transducers where changes in pressure cause them to expand and contract. They can also be positioned in groups at various positions to measure the strains in multiple directions, which is called a rosette. These gauges are also commonly found in load cells, which are generally used to measure forces on large structures [9].

Optical strain gauges work in the same manner as the electrical type; however they are different in makeup. They are comprised of fibers from glass (4-9 microns) and surrounded by a layer of pure glass (~125 microns in diameter). Optical strain gauges measure strain using light instead of electrical conductivity like electrical strain gauges [10].

Since there are so many options in method, size, type and quality of measurement devices, accuracy and precision of the tools chosen for our applications will be a function of the price of the tool. Cost will be considered in more detail for Design Review II. Therefore, as we develop our preliminary designs we will have a better understanding of which measurement devices we will chose.

PROJECT REQUIREMENTS AND ENGINEERING SPECIFICATIONS

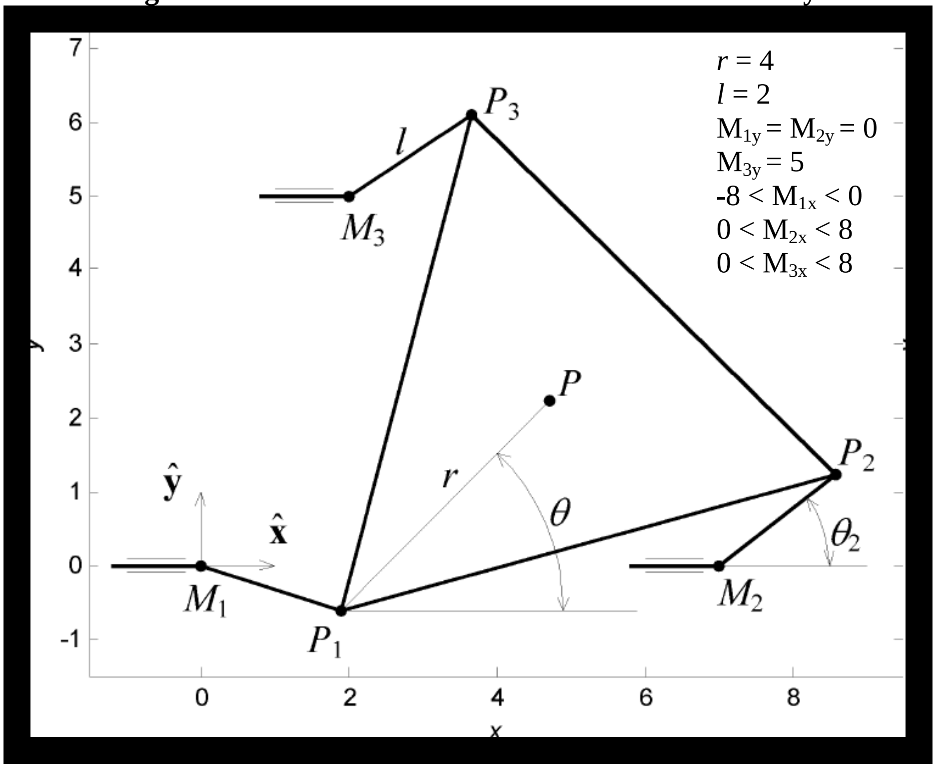
We determined several customer requirements with our sponsor to design an effective parallel mechanism with joint clearances that will accurately measure platform position, determine motor location, and measure the forces in the links. We then used background research and engineering knowledge to correlate engineering specifications with each requirement to ensure the requirements are met. The customer requirements and engineering specifications are outlined in Table 2.

Table 2: Customer Requirements and Corresponding Engineering Specifications

Customer Requirements	Engineering Specifications
• Joint clearance simulation	• Joint clearance = 0.1279x, x=1,2,3…
• Planar Motion	• Platform has 3 degrees of freedom (x, y, θ) • Platform must not rotate about the x or y axes more than ½ degree
• Measure platform position	• Motor position determined to within 2% of motor distance traveled
• Measure link forces	• Link forces determined to within 3% of weight applied
• Scaled to predetermined geometry	• See Figure 2
• Programmed to move on its own	• 3 motors required
• Apply force to close joint clearances	• 10 to 20kg force required

The geometry (Figure 2) was calculated by our sponsor as background for the project. In the figure, *r* represents the distance from a corner of the platform to its center, *l* is the length of each link, and M_1, M_2, and M_3 are the motor locations. The numbers specified in the figure represent the relative values for each parameter. The dimensions can be scaled as long as the relationships of each of the parameters remain constant.

Figure 2: Predetermined Parallel Mechanism Geometry



A House of Quality diagram (Figure 3) was used to determine the requirements and specifications that are most important for the parallel mechanism design and the correlations that exist between the engineering specifications.

Importance of Customer Requirements

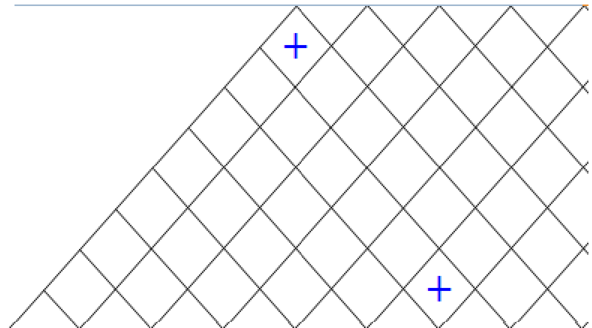
The weights of the requirements were determined by comparing two requirements at a time for each possible combination. The more important requirement was assigned a value of 1. After this process was complete, we summed the values for each requirement and divided by the total number of combinations to get a relative weight. Based on this analysis, we determine the most important requirements to be simulating the joint clearances and determining the platform and motor positions.

Importance of Engineering Specifications

To determine the key engineering specifications, we ranked the relationship between each customer requirement and engineering specification as “Strong”, “Moderate”, or “Weak”. We then determined that the key engineering specifications for our mechanism include designing the joint clearances and determining the motor position accurately.

Figure 3: House of Quality

Legend		
⊙	Strong Relationship	9
○	Moderate Relationship	3
▲	Weak Relationship	1
++	Strong Positive Correlation	
+	Positive Correlation	
—	Negative Correlation	
▼	Strong Negative Correlation	
▼	Objective Is To Minimize	
▲	Objective Is To Maximize	
x	Objective Is To Hit Target	



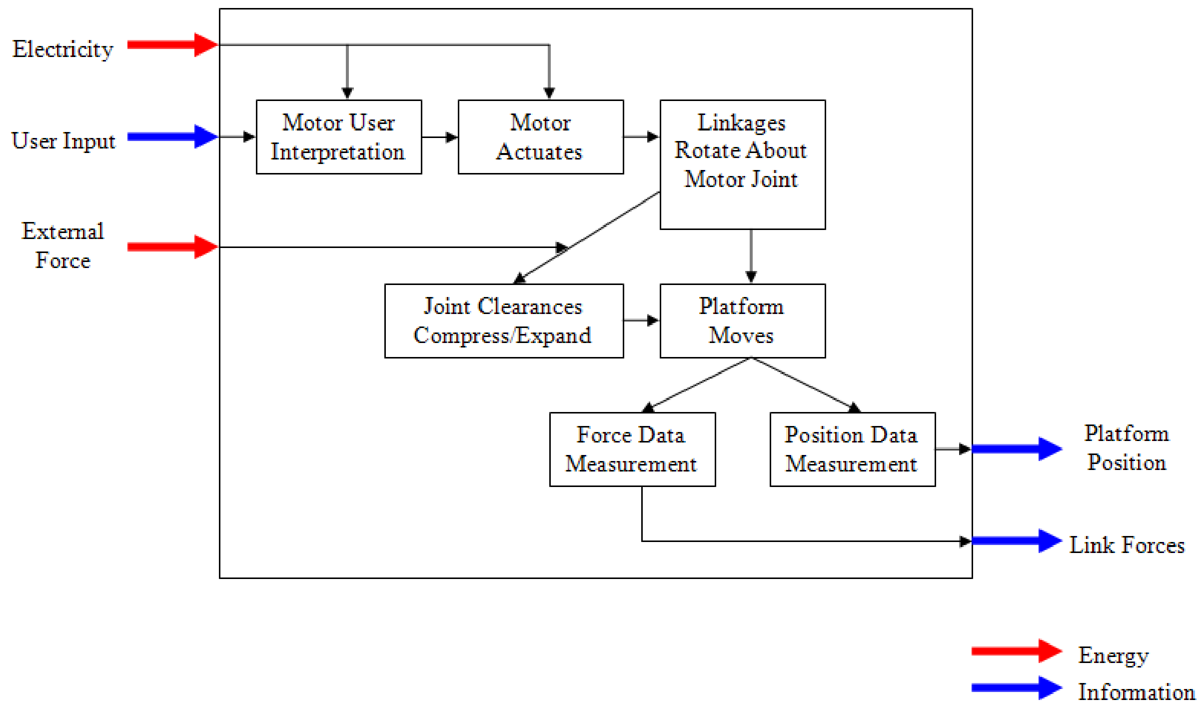
Row #	Max Relationship Value in Row	Relative Weight	Weight / Importance	Demanded Quality (a.k.a. "Customer Requirements" or "Whats") / Quality Characteristics (a.k.a. "Functional Requirements" or "Hows")	1	2	3	4	5	6	7	8
				Column #	1	2	3	4	5	6	7	8
				Direction of Improvement: Minimize (▼), Maximize (▲), or Target (x)								
				Quality Characteristics	The joint clearance must be scalable to 0.1279	The platform must not rotate about the x or y axes	The platform must have 3 DOF (x,y,theta)	The motor position must be determined to within 2% of the distance moved	Forces should be determined to within 3% accuracy of weight applied	Predetermined geometry	The mechanism requires 3 motors to simulate the required motion	A force of 10 to 20 kg is required
1	9	24.0	0.2	The mechanism must simulate joint clearances	⊙	○	○					○
2	9	19.0	0.2	The mechanism must move with planar motion		⊙	⊙					○
3	9	28.0	0.3	The mechanism must measure platform and motor position accurately			○	⊙				
4	9	10.0	0.1	The mechanism must measure the forces in the links					⊙			
5	9	14.0	0.1	The dimensions of the mechanism must be scalable to predetermined research geometry						⊙		
6	9	0.0	0.0	The mechanism must be programmable to move on its own							⊙	○
7	9	5.0	0.1	A force large enough to close the joint clearances must be applied to the mechanism		○			○		○	⊙
				Target or Limit Value	Achieve dimension to within +/- 5%	z-dir rotation < 1/2 degree	Greater freedom is better	Less than 2% is better	Less than 3% is better	Achieve dimensions to within +/- 2%	Use less only if constrained by time	Achieve force to within 2%
				Difficulty (0=Easy to Accomplish, 10=Extremely Difficult)	8	6	4	8	8	1	2	5
				Max Relationship Value in Column	9	9	9	9	9	9	9	9
				Weight / Importance	315.0	171.0	243.0	252.0	90.0	183.0	144.0	174.0
				Relative Weight	20.0	10.9	15.5	16.0	5.7	11.6	9.2	11.1

Engineering Specification Correlations
We determined several correlations between our engineering specifications by close examination. First, the size of the joint clearance will determine the magnitude of force required to close the clearance. Also, increasing the motor size will increase the value by which we scale the predetermined geometry. Finally, the force that can be applied to the mechanism will depend on the motors we use.

FUNCTIONAL CONCEPT GENERATION

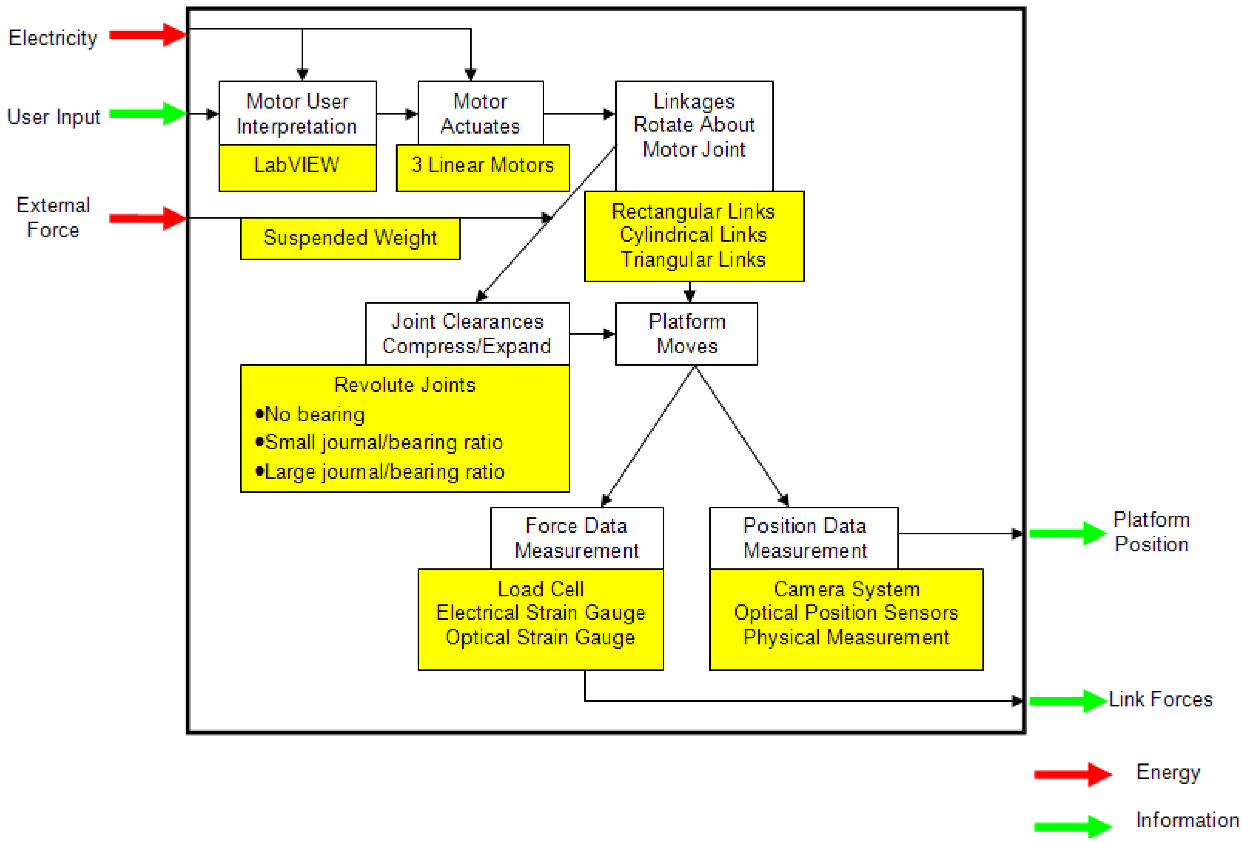
In order to develop an optimal design fitting our user needs and design specifications, our team divided the aspects of the parallel mechanism into inputs, functions, and outputs by creating a functional decomposition diagram. Figure 4 displays our functional decomposition diagram. We broke our inputs and outputs into energy and informational sources. The parallel mechanism inputs are electricity, an external force, and a user input specifying motor position. The functions include motor user interpretation, motor actuation, link rotation, joint clearance compression and expansion, platform movement, and force and position data measurement. The outputs include the platform position and axial link forces. Strategically breaking down our mechanism into inputs, functions, and outputs allowed our team to gain a more comprehensive look at the goals of our mechanism and areas requiring design.

Figure 4: Functional Decomposition Diagram



After creating the functional decomposition diagram, our team brainstormed multiple solutions for each function. A compilation of all of our function solutions can be found in Appendix D. This list includes all ideas even obviously infeasible ones. We narrowed down this list by evaluating all the ideas based on feasibility, available technology, and a go/no-go rating based on our user needs and design specifications. Feasibility was assessed by determining if the proposed design solution was possible to use in our mechanism. Available technology was evaluated based on if the proposed function solution was able to currently be manufactured or purchased. The go/no-go rating was based on whether the function solution fit all the user needs and design specifications. This three screen elimination process allowed our team to further investigate the best functional design solutions as seen in Figure 5 where the possible design solutions have been incorporated into the functional decomposition diagram.

Figure 5: Functional Decomposition Diagram with possible design solutions



The functional solutions for motor user interpretation and motor actuation have been previously specified by our customer. Dr. Bamberger asked our team to use three linear motors from the University of Michigan's Reconfigurable Manufacturing Lab. The motors will move with positions provided to us in exe files. We plan to use LabVIEW to control the motors since our team is most familiar with this software. Additionally, although the external force is an input, its functional solution was limited to one by the feasibility, available technology, and go/no-go rating based on our user needs and design specifications. Therefore, our team's main design considerations are links, joints, and force and position measurement.

Possible functional design solutions are a result of innovative ideas and background research. Link design solutions include rectangular, cylindrical, and triangular shapes. Force measurement design solutions include load cells, electrical strain gauges, and optical strain gauges. Position measurement design solutions include a camera system, optical position sensors, and physical measurement. Joint design solutions include revolute joints with no bearing, a small journal to bearing ratio, and a large journal to bearing ratio as more clearly defined in Figure 6.

Figure 6: Possible Joint Designs

As our team progressed in the design process, we frequently visited our functional decomposition diagram with possible design solutions to make sure every function was fully evaluated. The next section describes how our team selected the best functional design solution for our project.

FUNCTIONAL CONCEPT SELECTION

In order to evaluate which functional solutions best fit the needs of our project, our team developed scoring matrices for the main design considerations: link, joint, force measurement, and position measurement designs. Each scoring matrix lists criteria used to evaluate the designs and uses a reference system to score each design. The reference choice is arbitrary and is scored with all averages, "o". The remaining designs are compared to the reference design as either better, "+", or worse, "-". Finally, the better (+), average (o), and worse (-) traits are tallied to determine the net design rating and consequently the best functional solution.

Links

Link designs were evaluated based on the ease of force measurement device application, ease of joint fabrication, maximum link footprint, minimum cost, and minimum weight. Ease of force measurement device application is important in our design because we must measure the forces in each link. Successful force measurement depends on the area available to properly mount a device such as a strain gauge. Therefore, maximizing the link's footprint for mounting is critical. However, it is also important to minimize the cross sectional area of the links to ensure that the strain in the links is large enough to measure. Ease of joint fabrication is important because fabricating joints with large clearances is one of the main concerns determined by the scope of our project. Additionally, cost and weight are included in the selection criteria to minimize funding needed by our sponsor and decrease the stress on the joints and motors.

Link design solutions considered include links with rectangular, cylindrical, and triangular cross sectional areas. Cylindrical links were arbitrarily chosen as the reference design and therefore rectangular and triangular links were compared accordingly. Based on our link design scoring matrix in Table 3, rectangular links are the best functional solution. Rectangular links provide a flat surface for the force measurement device providing the maximum link footprint. Additionally, rectangular links provide a shape most compatible with fabricating revolute joints.

Table 3: Link Design Scoring Matrix

Link Design Selection Criteria	Rectangular	(Ref) Cylindrical	Triangular
Ease of Force Measurement Device Application	+	o	-
Ease of Joint Fabrication	+	o	-
Maximum Link Footprint	+	o	+
Minimum Cost	o	o	o
Minimum Weight	o	o	o
Sum of +'s	3	0	1
Sum of o's	2	5	2
Sum of -'s	0	0	2
Net Score	3	0	-1
Rank	1	2	3

Joints

Joint designs were evaluated based on the accuracy of the fabricated joint clearance, ease of manufacturing, link/platform footprint, minimum cost, minimum friction, precision, and scaling restrictions. Accuracy and precision of our joints are the primary concerns because we must incorporate a joint clearance specified by our customer since it is critical to our project's scope. Link/platform footprint refers to the area on the link or platform remaining after the joints have been manufactured. The largest remaining link/platform footprint after joint fabrication is desired for a force measurement device and tilt indicator application. Additionally, minimum cost and friction are included in the selection criteria to minimize funding needed by our sponsor and decrease the stress on the joints and limit potential tilt opportunities.

Joint design solutions considered include no bearing, small journal to bearing ratio, and large journal to bearing ratio. The small journal to bearing ratio joint was arbitrarily chosen as the reference design and therefore no bearing and large journal to bearing ratio joints were compared accordingly. Based on our joint design scoring matrix in Table 4, small journal to bearing ratio joints are the best functional solution. These joints provide high accuracy in joint clearance fabrication due to high accuracy bearings and journals. The nature of the small joint provides maximum link/platform footprint, minimal area for static and kinematic friction, and fits our scaling restrictions. Additionally, the use of small bearings reduces the overall cost of the joint when compared to a large journal to bearing ratio joint while providing less friction than a joint without a bearing.

Table 4: Joint Design Scoring Matrix

Joint Design Selection Criteria	(Ref) Small Journal/Bearing Ratio	No Bearing	Large Journal/Bearing Ratio
Accuracy	o	-	o
Ease of Manufacturing	o	+	+
Link/Platform Footprint	o	o	-
Minimum Cost	o	+	-
Minimum Friction	o	-	o
Precision	o	-	o
Scaling Restrictions	o	o	-
Sum of +'s	0	2	1
Sum of o's	7	2	3
Sum of -'s	0	3	3
Net Score	0	-1	-2
Rank	1	2	3

Position Measurement

Position measurement designs were evaluated based on their accuracy, ease of use, minimum cost, precision, and readability. Accuracy and precision of our position measurement readings are primary concerns because we need to be able to numerically characterize the motion of our parallel mechanism consistently. Ease of use and readability are also important selection criteria because determining the platform's position should be unambiguous and independent of the user or operator. Additionally, minimum cost is included in the selection criteria to minimize funding needed by our sponsor.

Position measurement design solutions considered include a camera, fiber optics, and physical measurement systems. Physical position measurement was arbitrarily chosen as the reference design and therefore the camera and fiber optic systems were compared accordingly. Based on our joint design scoring matrix in Table 5, a camera system is the best functional solution. A camera system provides a constant and consistent operator perspective to measure the platform's position. Cameras are easier to use than fiber optic systems which require additional processing equipment and are relatively inexpensive. A dependable web camera can be purchased for under a hundred dollars.

Table 5: Position Measurement Design Scoring Matrix

Position Measurement			(Ref)
Selection Criteria	**Camera**	**Fiber Optics**	**Physical Measurement**
Accuracy	+	+	o
Ease of Use	+	-	o
Minimum Cost	-	-	o
Precision	+	+	o
Readability	+	+	o
Sum of +'s	4	3	0
Sum of o's	0	0	5
Sum of -'s	1	2	0
Net Score	3	1	0
Rank	1	2	3

Force Measurement

Force measurement designs were evaluated based on their accuracy, ease of manufacturing, link assembly feasibility, minimum cost, precision, and system footprint. Accuracy and precision of our force measurement readings are primary concerns because we need to be able to numerically characterize the axial force in our links based on our user requirements and design specifications. Ease of manufacturing relates to the minimum number of additional devices associated with each force measurement device that needs to be manufactured or purchased to interpret the force measurement. Link assembly feasibility is a selection criterion determining the possibility of attaching the force measurement device to the links. System footprint should be minimized to negate complexities in platform motion interference. Additionally, minimum cost is included in the selection criteria to minimize funding needed by our sponsor.

Force measurement design solutions considered include an electrical strain gauge, optical strain gauge, and load cell. The load cell was arbitrarily chosen as the reference design and therefore the electrical and optical strain gauges were compared accordingly. Based on our force measurement design scoring matrix in Table 6, an electrical strain gauge is the best functional solution. An electrical strain gauge is accurate and precise when adhered to the link properly. Adhering the strain gauge to the link can be done by an expert or by a novice with lots of practice. The electrical strain gauge can move with the links and has a small footprint which will avoid interference with the motion of the platform. Furthermore, although additional devices are required to interpret the force measurement data, our team is able to borrow the devices from the University of Michigan Mechanical Engineering Labs, minimizing cost to our customer.

Table 6: Force Measurement Design Scoring Matrix

Force Measurement Selection Criteria	Electrical Strain Gauge	Optical Strain Gauge	(Ref) Load Cell
Accuracy	o	-	o
Ease of Manufacturing	-	+	o
Link Assembly Feasibility	+	+	o
Minimum Cost	+	-	o
Precision	o	o	o
System Footprint	+	+	o
Sum of +'s	3	3	0
Sum of o's	2	1	6
Sum of -'s	1	2	0
Net Score	2	1	0
Rank	1	2	3

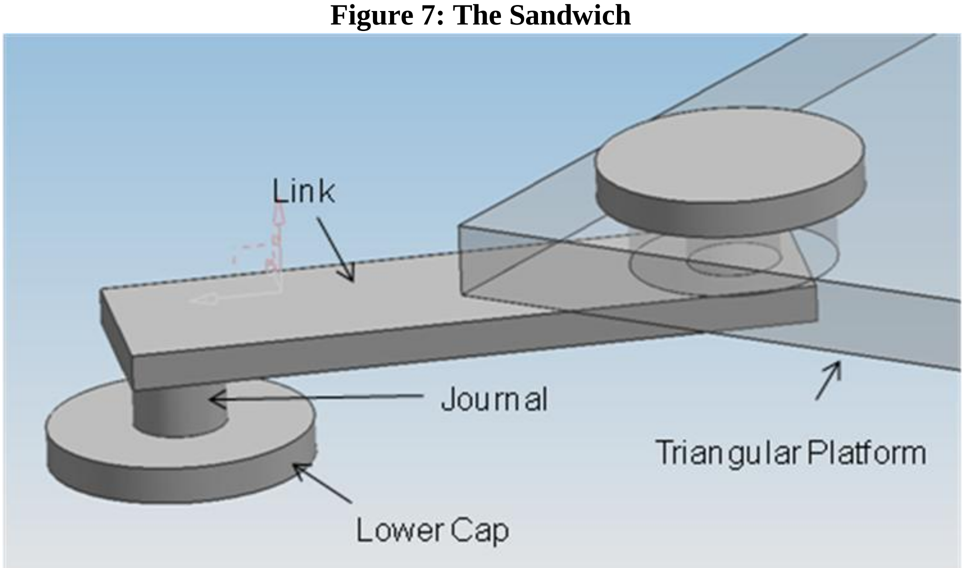
ANTI TILT CONCEPT GENERATION

One customer requirement of this project is to prevent the platform from tilting about the x or y axes during and after repositioning. For this reason, creative methods for anti tilt were explored in order to meet this requirement. At the same time, we were not able to alter the mechanism's geometry, limit or interfere with the motion. In a general brainstorming sense, the basic ideas that we explored were to support the platform from above, below, and from the sides. The following concepts utilize these ideas.

Concept 1 – The Sandwich

The sandwich prevents tilt by providing support from a large surface area both above and below the platform, at each of the three corners. The link provides support from below and the cap provides support from above. Each link and cap is connected by a journal (see Figure 7). The journal is manufactured with precise tolerance (corresponding to the plate thickness) in order to keep the platform flush to both the link and cap from the bottom and top respectively.

Figure 7: The Sandwich



Concept 2 – The Slider

The platform is secured some distance above and parallel to the surface on a set of rigid stilts. These stilts are connected to a larger circular platform which maximizes area while minimizing the footprint (see Figure 8). This larger platform is in constant contact with the surface and acts as a slider since it is only connected to the platform. The slider prevents tilt by providing a counter moment to any off axis force that would otherwise cause rotational motion in the platform (see Figure 9). The contact between the slider and the surface is either lubricated or coated in order to reduce static and kinematic friction.

Figure 8: Counter Moment Visualization

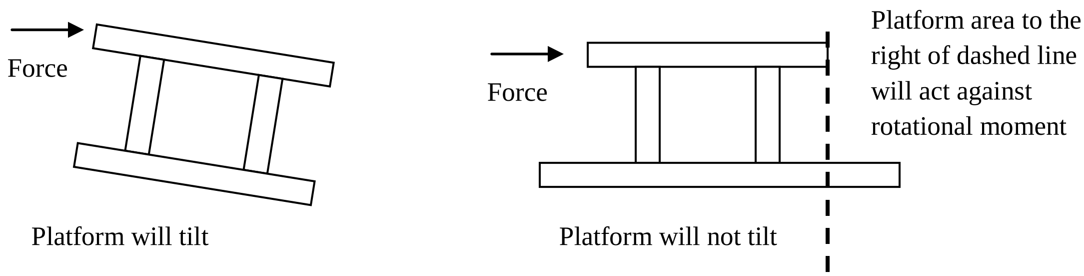
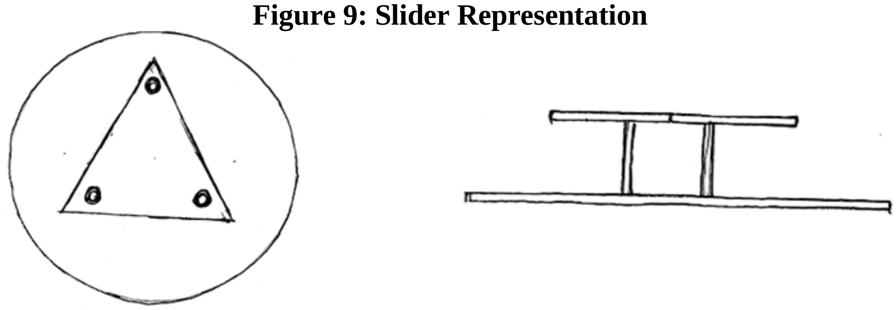


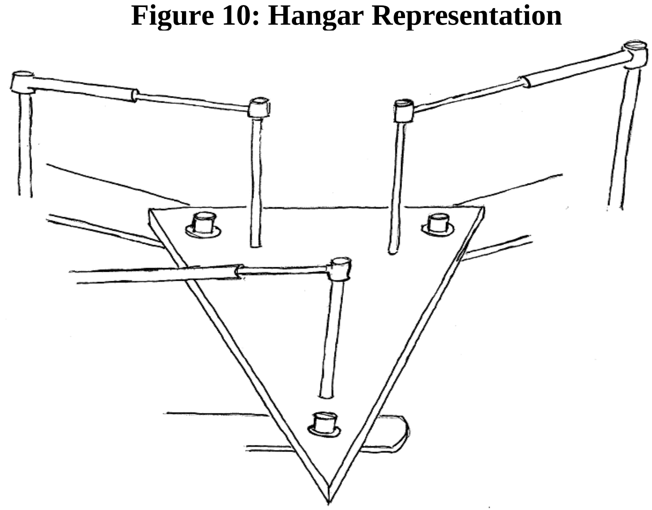
Figure 9: Slider Representation



Concept 3 – The Hangar

The hangar configuration suspends the platform into the air from above. A hanging arm (see Figure 10) extends over each joint and is fixed to the platform at each of the three corners. The hangar composition consists of two vertical rods, one fixed behind the motor joint on the motor mount, and the other fixed inward on the platform from the joint. Atop each of these rods is a cylindrical joint, and these joints simply support a prismatic sliding joint. The movement of the hangar suspension system is directly dependent on the motor, link and eventually platform movement.

Figure 10: Hangar Representation



ANTI TILT CONCEPT SELECTION

In order to evaluate which anti tilt solution would best fit the needs of our project, our team developed a scoring matrix. The scoring matrix lists criteria used to evaluate the designs and uses a reference system to score each design. The slider was arbitrarily selected as the reference and scored with all averages, “o”. The remaining designs are compared to the reference design as either better, “+”, or worse, “-“. Finally, the better (+), average (o), and worse (-) traits are tallied to determine the net design rating and consequently the best functional solution. The scoring matrix for anti tilt designs is shown in Table 7. Based on our Anti Tilt Concept Scoring Matrix, the Sandwich is the best functional solution.

Table 7: Anti Tilt Concept Scoring Matrix

Anti Tilt Design		(Ref)	
Selection Criteria	**Sandwich**	**Slider**	**Hangar**
Minimum Cost	+	o	-
Minimum Weight	+	o	+
Ease of Manufacturing	o	o	-
Minimum Footprint	+	o	+
Minimum Friction	+	o	+
Prevent Tilt	o	o	o
Reconfigurability of Links/Joints	-	o	-
Joint Intricacy	+	o	-
Sum of +'s	5	0	3
Sum of o's	2	8	1
Sum of -'s	1	0	4
Net Score	4	0	-1
Rank	1	2	3

Concept 1 – The Sandwich

The simplicity of the sandwich design provides most of its benefits. There is a minimum amount of added material beyond the required geometry, which includes caps and fasteners for each joint. This means that complex manufacturing and fabrication will be held to a minimum, which will allow us to test the success of the mechanism, not the support structure. Since less material is used, the mechanism is lighter and also has a smaller footprint and less friction. The simplicity also leads to a low cost because it does not introduce additional complex parts. The act of sandwiching the platform between the link and cap may cause friction that would prevent the joint clearances from closing. Lubrication or tolerance adjustments may be required if this issue arises.

Concept 2 – The Slider

The main advantage of this concept is that as long as one places the platform on a level surface it will remain level. Because the links are not fastened to or supporting the platform in any way, they can easily be removed and refitted to simulate different joint scenarios for additional testing without disturbing the rest of the setup. Unfortunately, the slider has a tremendous footprint, and is in direct contact with the surface. In regards to force measurement, the added static friction could reduce and change the maximum link forces, thus preventing the ability to compare them to the static calculation described in the Analysis section. This will lead to the increased material costs involved with adding a large surface plate to reduce the comparatively high amount of friction against the slider. Also, in order to achieve the necessary space for the slider to sit, the motors must be positioned much farther back. This means that the motor mounts will have to extend almost the entire length of the platform, supported from only the motor end. This circumstance would make it difficult to prevent mount deflection and keep the links parallel to the surface, thus providing inaccurate positioning data with respect to the motor location.

Concept 3 – The Hangar

The main drawback from this concept is that it is complicated. There are multiple parts that must be purchased or fabricated and then fastened together with extremely tight tolerances in order to prevent any type of possible instability. This means that there is a higher cost because of the nine additional joints that are required for the configuration, and if each component is not manufactured precisely, then the mechanism will not be able to prevent rotation, or even hold the platform level when stationary. This design also allows for the removal and refitting of joints on links, however in order to access the links, the platform must be disassembled, since the hangars are suspended above the links and hold the platform.

THE “ALPHA DESIGN”

The alpha design incorporates the optimal functional design solutions as well as the design solution to restrict the platform to planar motion. The functional design solutions include rectangular links, a small journal to bearing ratio in the links, a camera and an electrical strain gauge. The sandwich design is incorporated in the alpha design as the anti tilt design solution. These features can be seen in Figures 11 and 12 below. In addition to the functional and anti tilt design solutions, the mechanism dimensions and joint assemblies will be discussed in this section.

Figure 11: The "Alpha Design"

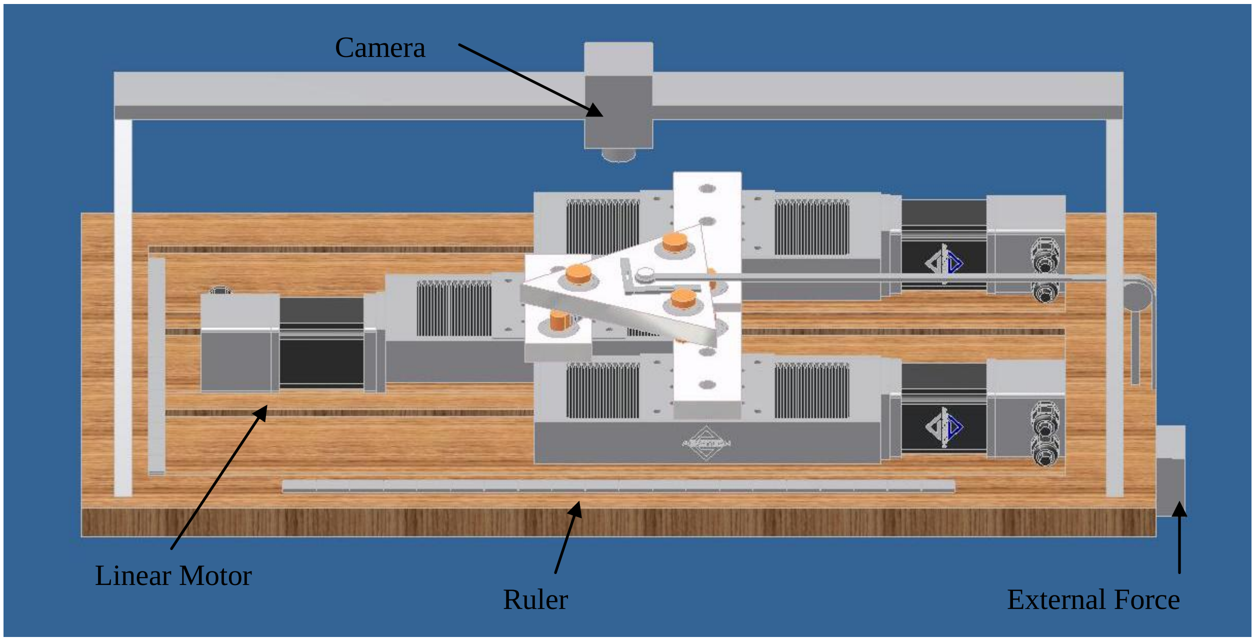
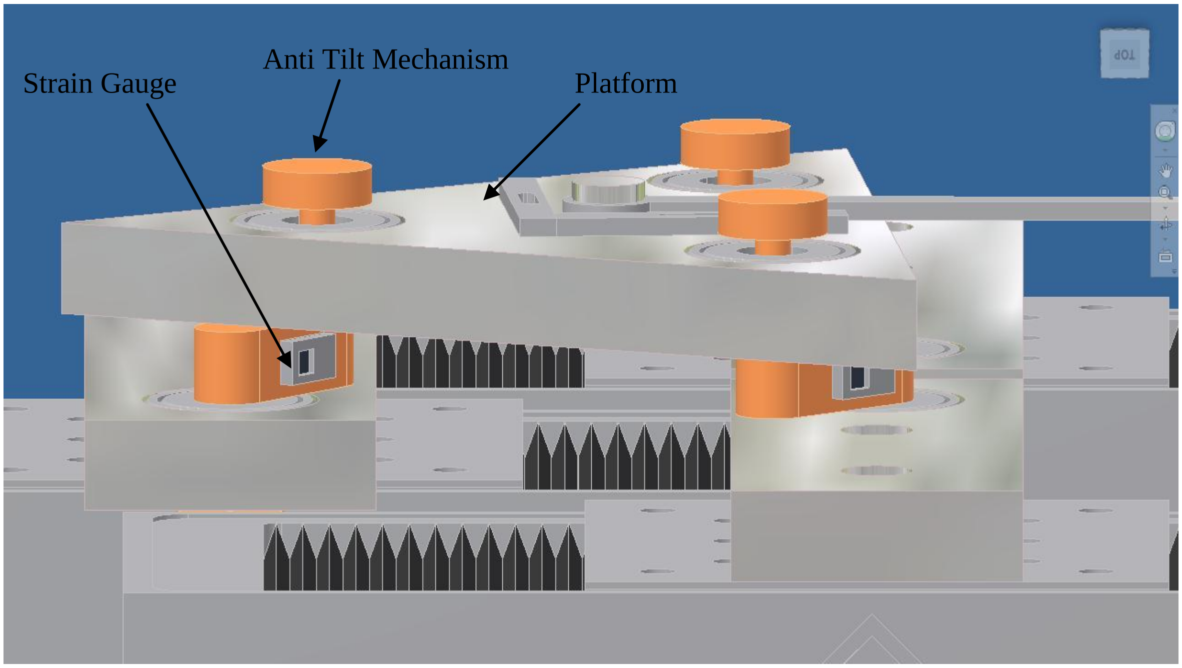


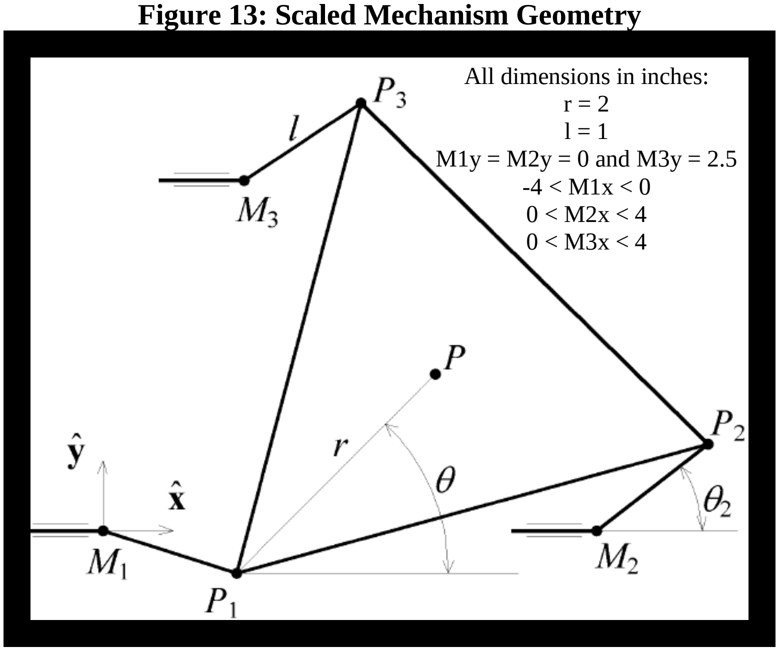
Figure 12: The "Alpha Design"



Scale

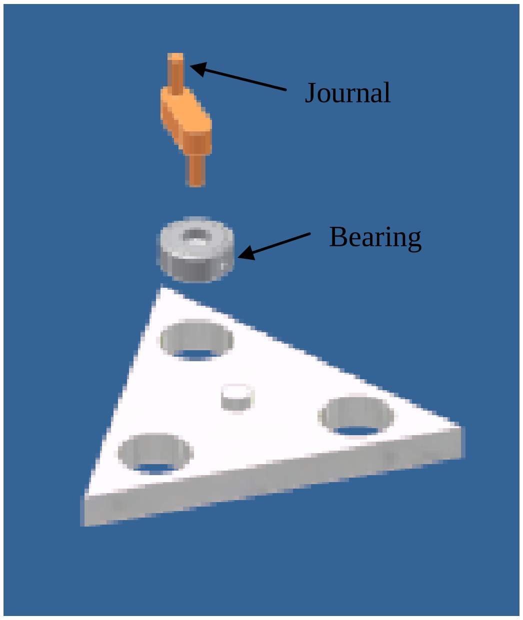
Choosing an appropriate scale for the predetermined geometry (Figure 13) is important because the size of the mechanism will affect the overall material cost and the size and arrangement of functional design components. Through careful inspection, we determined that the scale of our mechanism is constrained by the range of the linear motors provided by our sponsor. The motion of the mechanism will be constricted by the linear motor with the smallest range of motion (4 inches). Because the predetermined geometry specified a motor range of 8, we will scale the dimensions of our mechanism by 1/2.

Figure 13: Scaled Mechanism Geometry



Joint Components and Assembly
The alpha design will incorporate revolute joints with a small journal to bearing ratio. The bearings will be located on the platform and the journals will be connected to the links. The joint clearance between the links and the platform will be doubled so the link to motor joints will not require clearance. With this design, we will be able to manufacture a second platform with bearings that fit tight with the journals on the links. This will allow us to simulate the platform motion without joint clearance to compare the results with the platform motion with joint clearance. This will help to prevent tilting in the links.

Figure 14: Joint Assembly



External Force
The alpha design incorporates an external force between 10-20 kg applied in the y-direction. The forces functions to close the joint clearance prior to moving the mechanism with the motors. We will borrow an engineering weight from a University of Michigan Laboratory. This weight will be suspended by a rotating pulley to allow for platform motion with the motors.

Position Measurement
A webcam will be used to take pictures of the platform between each position specified in the motor program exe files. We will input these pictures into Matlab to analyze the platform's position with respect to the 3 markers located on the surface of the platform. Rulers will be attached to the edges of the base to use as a reference for this analysis.

ENGINEERING ANALYSIS

This section will outline the preliminary analysis our team has completed to ensure feasibility of the strain gauge to measure link forces as well as future analysis that will be conducted to prove the alpha design concepts.

Strain Gauge Feasibility
To ensure the use of a strain gauge is appropriate, we determined the range of forces a typical strain gauge can sense and compared this to the range of potential forces on the links. To calculate the force in the links given the strain as output from the strain gauge, we used Equation 1.

$$F = \varepsilon A E \tag{1}$$

Where F is the link force [N], ε is the link strain, A is the cross section area [m^2], and E is the modulus of elasticity [Pa]. A typical strain gauge can measure strain values between -.002 and .002. These values represent elastic deformation. Assuming a material of Aluminum (E ≈ 70 GPa) and cross sectional dimensions of ¼" by ¼" (A = 4.03×10^{-5} m^2), a typical strain gauge can be used to calculate forces between -11,290 and 11,290 N.

Statics calculations assuming no joint clearances were completed to determine a range of forces expected in the links. See Appendix E for details on the calculations. Based on this analysis, the links will experience forces up to approximately 700 N. Therefore, the range of forces in the links will be well within the range of forces that can be calculated with a strain gauge.

To verify that the resolution of the strain gauges will not be a problem, we determined the resolution of the equipment we will use and the associated error with forces we expect to see in the links. The resolution in strain measurement of the equipment we will use is 2.4×10^{-10}. For a 700 N force, we expect to measure a strain of 1.2×10^{-4} and for a 1 N force, we expect to measure a strain of 1.8×10^{-7}. These values were calculated using Equation (1) above. For a 1 N force, the resolution of the equipment is 1.3% of the expected strain based on theoretical calculations.

The cross sectional area and modulus of elasticity were minimized in this analysis to minimize the range of measurable strains. This will reduce resolution error in the strain gauge readings. A minimal link thinckness was chosen based on strain gauge dimensions. Aluminum was chosen as a material to minimize the modulus of elasticity.

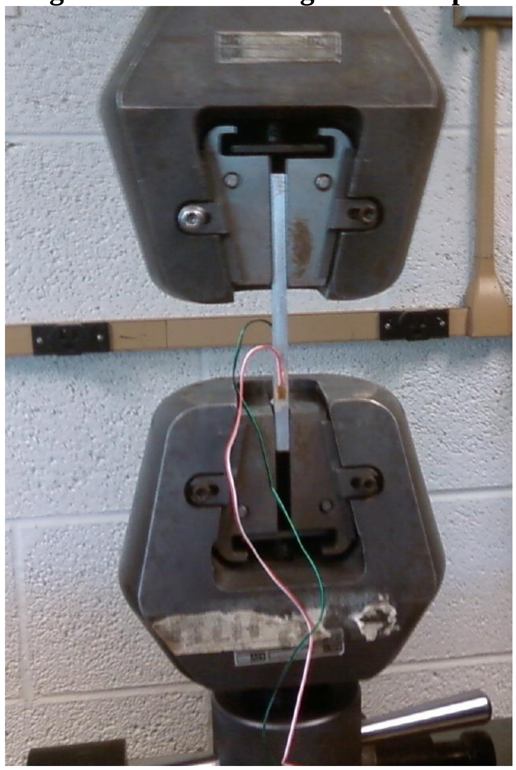
PARAMETER ANALYSIS

In addition to the engineering calculations performed prior to Design Review 2, we conducted more testing and analysis to prove our design. These verifications include strain gauge testing, motor analysis, link bending calculation, and camera verification.

Strain Gauge Testing
Testing was conducted on the strain gauges we will use with our final design to ensure the strain can be measured with our link design and data acquisition equipment. This section will discuss the test setup, results, and plans for future strain gauge calibration.

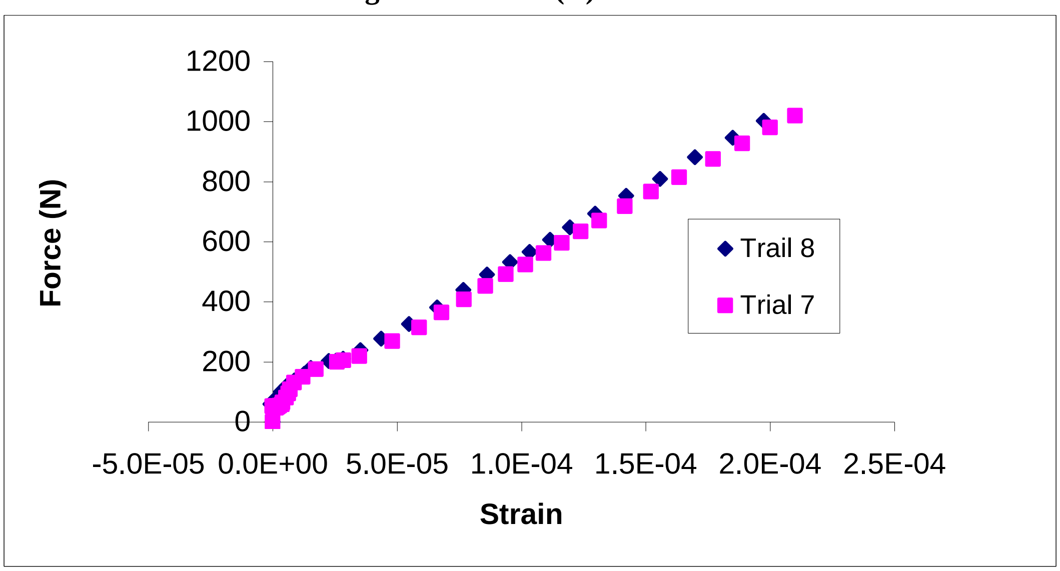
Test setup: We used an aluminum test specimen with the same cross sectional area as our designed link. We attached a strain gauge to each side of the specimen, mounted the specimen in an Instron machine, applied a tensile load up to 1 kN, and measured the resulting strain. Figure 14 displays the test setup.

Figure 14: Strain Gauge Test Setup



Results: Applied force and resulting strain are plotted in Figure 15. It is apparent from this figure that the equipment used to measure strain will be capable of measuring the maximum strain value we expect in our links. However, we were only able to measure forces greater than 269 N with error less than 7% based on theoretical calculations. Due to this, we changed the link design to further minimize the cross sectional area to ¼" x ¼" from ½" x ¼". Based on these theoretical calculations with this reduced area, we will be able to measure forces of 169 N with the same 7% error.

Figure 15: Force (N) vs. Strain

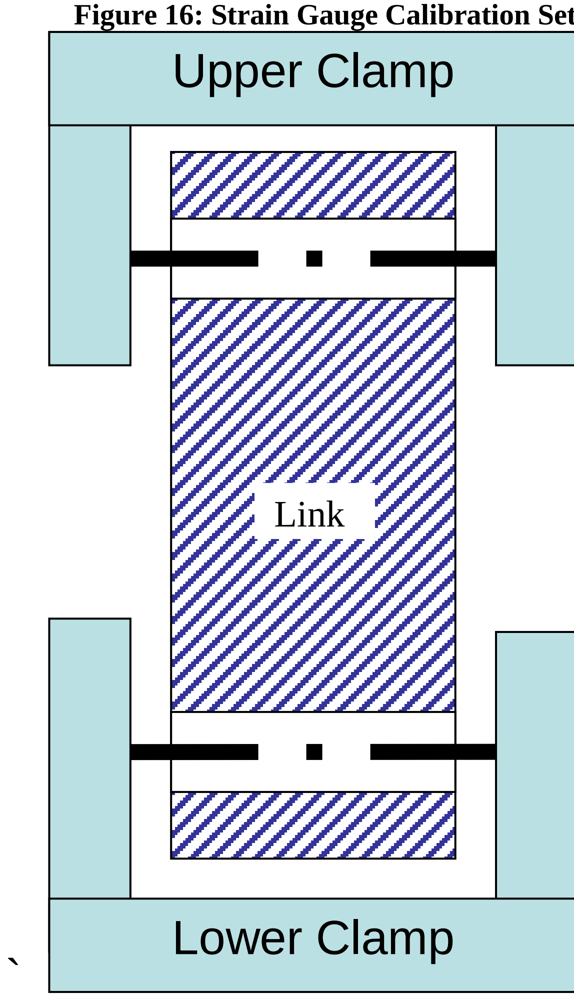


Plans for future strain gauge calibration: During the strain gauge testing, we concluded that the test specimen must be mounted in the centerline of the Instron machine to avoid putting a bending moment on the specimen. When we switched from mounting the specimen in the front of the machine to the middle, we were able to measure more accurate strain values based on theoretical calculations. We also concluded that it would be ideal to maximize the distance from the strain gauge to the mounting location of the specimen to avoid measuring strain associated with the machine's clamping force.

In order to account for these testing observations when we calibrate the strain gauges mounted to the actual links, we will mount each link to the machine using a journal placed through the link holes (see Figure 16). This will allow for a more uniform tensile force to be applied to the specimen, which will increase the accuracy of the strain measurements. Also, since the test specimen itself will not be clamped to the machine, the clamping force will not be associated with strain measurement, further reducing the strain measurement error.

We will produce a total of 4 links with 2 strain gauges per link for calibration testing. Although we only need 3 links for the mechanism, we will produce an extra link in case something goes wrong with link manufacturing or strain gauge mounting. We will apply a load up to 1 kN as we did in the initial testing and we will use the strain gauge on each link that results in the most accurate strain measurements based on theoretical calculations for the final mechanism.

Figure 16: Strain Gauge Calibration Setup



Motor Analysis
The linear motors provided by our sponsor can produce a maximum force of 89 N in both the axial and radial directions. Based on static calculations, we expect the links to experience a maximum axial force of -97 N and a maximum radial force of 671 N assuming a 20 kg external weight. Since the linear motors will not have sufficient power to move the mechanism under this external force, we reduced this force to 2.5 kg so the maximum axial and radial forces expected in the links will be under the 89 N available from the motors. Table 8 summarizes the maximum forces expected under varying external weights.

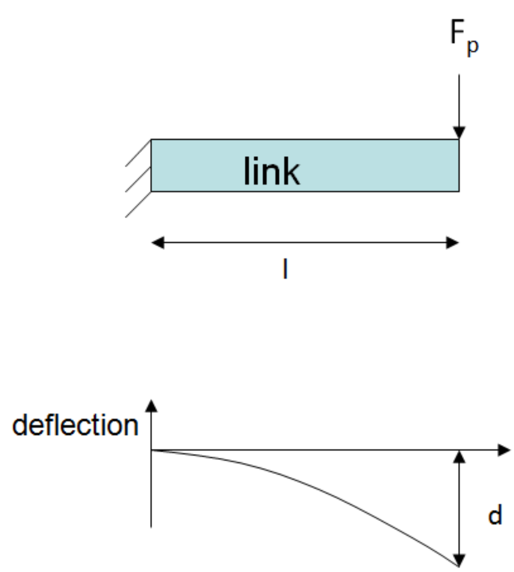
Table 8: Link forces expected under varying external weights

External Weight	**20 kg**	**10 kg**	**5 kg**	**2.5 kg**
Maximum Axial Force	-97 N	-48 N	-24 N	-12 N
Maximum Radial Force	671 N	335 N	168 N	84 N

A range of 10-20 kg for the external weight was originally specified as an appropriate range to close the joint clearances prior to scaling the mechanism by ½. Since the scaled mechanism will not be as large as originally expected, we believe a smaller external force will be sufficient to close the joint clearances in the mechanism. Also, decreasing the external force will decrease the strain in the links, thus increasing the resolution error in the strain gauges. Due to this, we will attempt to use a 5 kg weight. This should not cause problems with the motors since the 89 N axial and radial force limit is very conservative.

Link Bending Calculation
We performed a calculation to ensure the mechanism's links will not bend. We assumed a cantilever beam where one end on the link is fixed to the motor mount and the other end experiences a force due to the weight of the platform. This schematic can be seen in Figure 17 below.

Figure 17: Model for link bending calculation



Using this model, the end of the link will experience a deflection, d, defined in Equation 2 below where F_p is the force exerted on the link due to the weight of the platform (F_p = mp*g), l is the length of the link, E is the modulus of elasticity of aluminum, and I is the moment of inertia about the cross sectional area of the link. Values for these parameters can be seen in Table 9 below.

$$d = \frac{F_p l^3}{3EI} \tag{2}$$

Table 9: Link bending calculation parameters

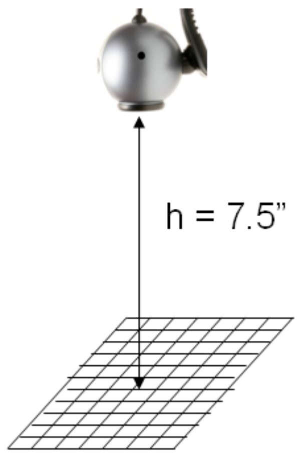
Parameter	Value
F_p	2.1 N
l	0.0254 m
E	70 GPa
I	$1.35*10^{-10}$ m^4

For this analysis, we assumed that the link will experience a force due to the weight of the entire platform even though this force will be equally dispersed between the three links. This assumption will result in a worst case deflection. Using this analysis, we determined the link deflection to be .0012 mm. This deflection can be considered negligible.

Camera Verification

We performed a simple camera test to verify the vertical distance necessary to mount the webcam so we can fully view the platform in every position. We marked out a 5” x 5” field of view to model the position locations we expect and then measured the height of the webcam resulting in this field of view (see Figure 18).

Figure 18: Camera verification setup



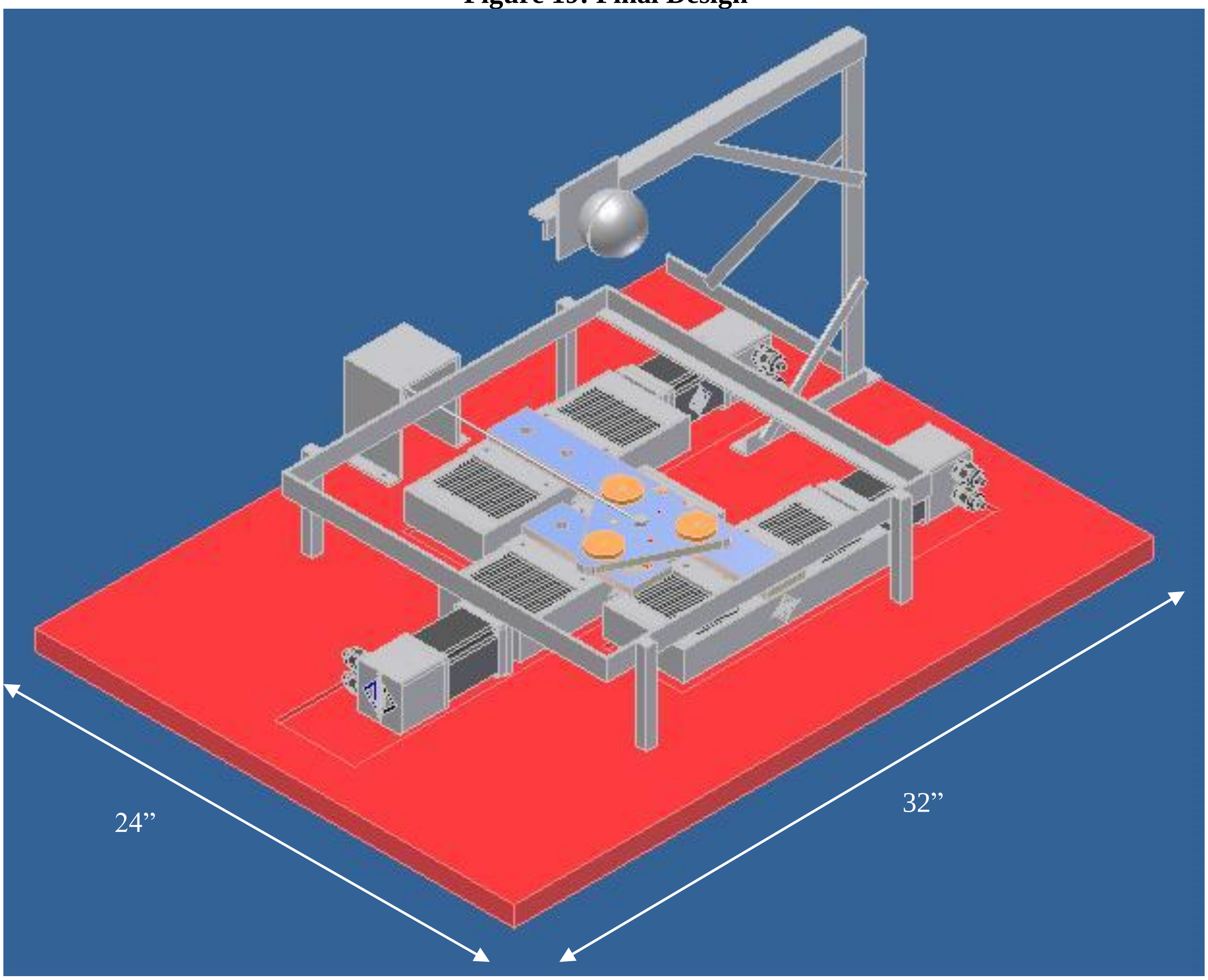
We determined that a height of 7.5” will necessary to view this entire area. Given that the distance from the platform to the camera will be 7.5”, the camera stand will need to be 15” high.

FINAL DESIGN DESCRIPTION

This section will discuss new designs since Design Review 2 as well as the elements of the final design including the base, platform, motors, motor program, caps, links and journals, motor mounts, pulley assembly, grid assembly, and camera assembly, and position program. Subsections my include figures when appropriate to clarify which part is being discussed. Appendix G includes detailed dimensioned CAD drawings of each critical part.

Designs Changes
Since Design Review 2, our team made three changes to our design. These include the use of rulers as a reference for position measurement, link design, and external force application. Instead of using rulers as a reference for the position measurement, we decided to use a grid overlay with 1" increments. Based on the strain gauge testing, we reduced the cross sectional area of the links to ¼" x ¼" from ½" x ¼". Also, our customer specified that the external weight should be applied in the y-direction instead of the x-direction which was specified in the initial design. The magnitude of external force was also reduced based on the motor verification calculation in the Prototype Parameter Analysis section above. Figure 19 shows our final design.

Figure 19: Final Design

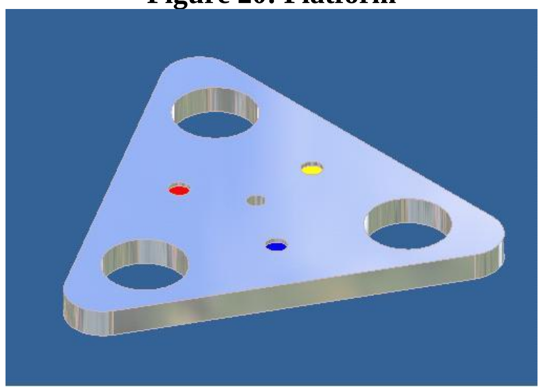


Base
The Base for the entire mechanism is 32"x 24". It is created by a ½" thick PVC sheet. The reason for this decision is that even though it is more costly than 1" thick sanded plywood, it provides rigidity for our construction and it allows us to use the water jet or a mill to accurately machine motor mounting holes. This will keep the linear motors and their tracks parallel with respect to each other. Also, because each motor is mounted on the end of a track and dips below the track mounting points, we will remove the material sections where the motors would otherwise come in contact with the base. Nine 4"x 4"x 1/2" PVC support blocks will be cut from the leftover left over base material and mounted to the bottom of the base, in order to keep the protruding portions of each linear motor from contacting the exterior surface.

Platform

The platform in Figure 20 will be created from 3/8” T6061 aluminum. To reduce the effect of friction and stress at the joint locations, a ½” inner diameter steel ball bearing is press fit into each of the joint locations. These ball bearings are designed to work under, and resist radial loads. The platform will be drilled in 3 locations for marker positioning during initial fabrication to ensure that they are geometrically constrained. Each marker is a 1/8” threaded hole, for a bolt with a colored head. At the center of the platform, there will be a ½”-13 threaded hole for a 1” shoulder bolt that will act as an application point for our external force. This knob will accept the force from the pulley assembly and close the clearance in each joint, which is described in further detail in the Pulley Assembly section.

Figure 20: Platform



Platform without Joint Clearances

We will manufacture a second platform, identical as the first with respect to material and dimension to simulate the platform motion without joint clearances. After the same ½" inner diameter bearing is press fit into each joint location, a ½” diameter journal is press fit into the bearing. This platform will be placed in the same assembly set up and use the same test paths as the joint clearance platform. This platform will determine the percentage of error in movement caused by the joint clearances.

Joints

There are two different types of joints in our system’s design. The first joint (Figure 2, left) is one that has no clearance. In this case, the journal will be press fit into the bearing before the system is assembled. The other joint (Figure 21, middle) has a clearance of 0.125” so that the platform will simply be placed over the journals. For this joint we will apply an external force in the y-direction to close this clearance (Figure 21, right).

Links and Journals

The links are created from T6061 aluminum. They are each 2” long and have a ¼” square cross section. The thickness of the link is constrained to a minimum of ¼” to properly mount strain gauges which are required for force measurement data acquisition. Two types of links will be manufactured; one to simulate joint clearance and another to simulate no joint clearance. Each journal will be created from a steel rod. Using a lathe, the rod can be turned down on the top and bottom by ¼” for the clearance journals and on the bottom by ¼” for the no clearance journals. These distances are determined by the thickness of the links and caps.

In order to fasten the journals and links together, an 8-32 threaded hole will be used. Any larger hole or further reduction in the link’s width would degrade the structural integrity of the mechanism. The bottom ¼” of threads on the clearance and no clearance journals will be screwed into the 8-32 link hole. The top ¼” of threads on the clearance journals will be screwed into the 8-32 cap holes (see cap section below). A notch will be created in the top of the no clearance journals so a screwdriver can be used to screw these journals into the links.

Figure 23: Link (Left) and Journal (Right)

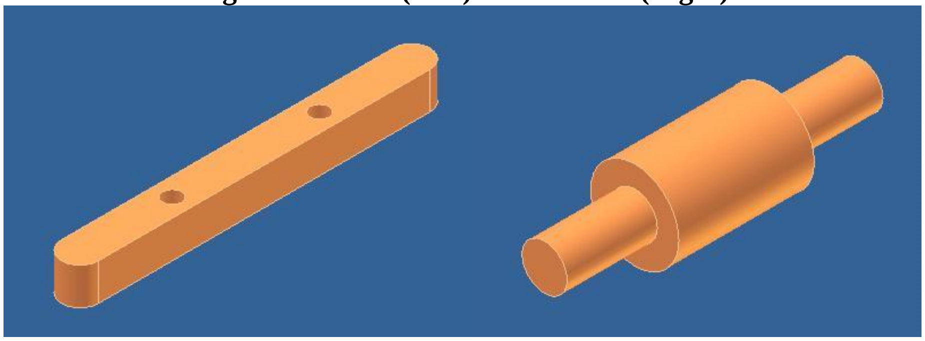
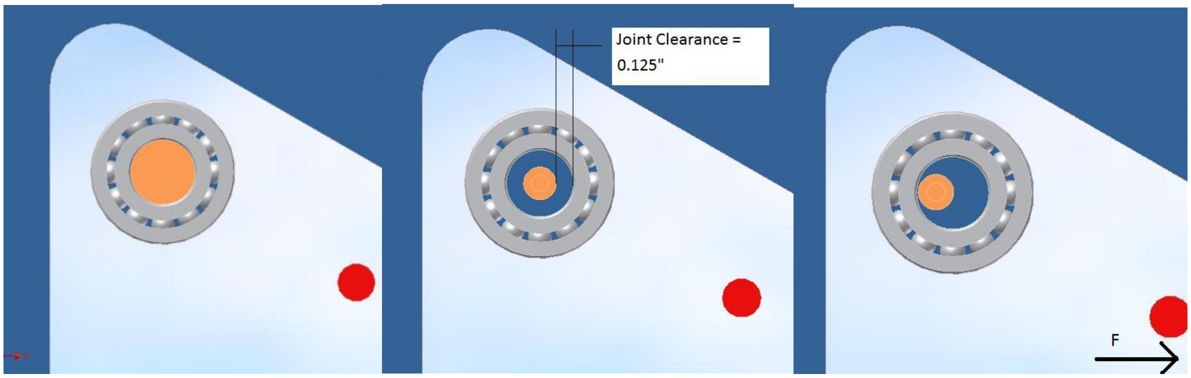


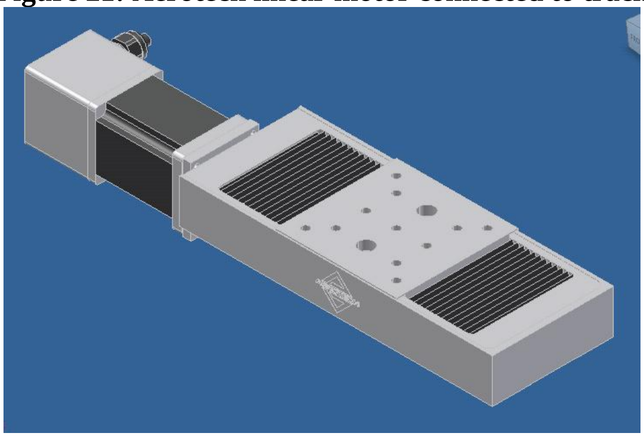
Figure 21: Joint with: no clearance (left), clearance (middle), closed clearance (right)



Motors

The platform will move under the power of 3 linear motors. The motors are provided by the University of Michigan Reconfigurable Manufacturing Lab, and are model number BMS60-A. The Aerotech specification sheet rates the motor for an axial and radial load of 89N. Each motor drives an Aerotech ATS100 track.

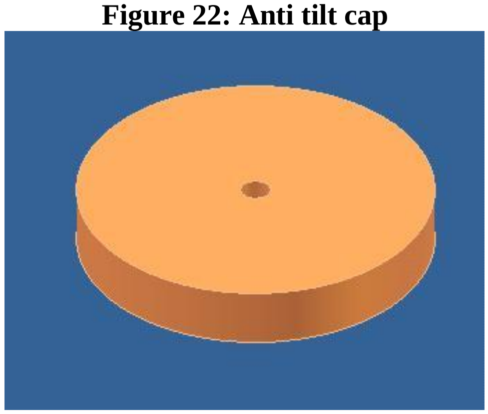
Figure 21: Aerotech linear motor connected to track



Motor Program
Due to assembly modes inherent in parallel mechanisms, a customized motor program must be developed. To start any maneuver, the mechanism with or without joint clearances needs to be disassembled to the motor platform level. This will prevent damage to the mechanism or the motors. Homing commands will then be sent separately to each of the three motors. This home will be the zero of the motor. Next, three sequential commands will be sent to the three motors to move them to their desired location. Once in the correct starting location, the removed portion of the mechanism can be reattached to the motor platforms in the correct assembly mode as indicated by the link's angles given by the customer. An image for position measurement and strain gauge reading will be taken at this starting location. After the mechanism has been configured in the correct assembly mode, tests for that assembly mode can be conducted. Commands will be sent to move the motors. Once the motors are in the correct position, an image for position measurement and a strain gauge reading will be taken. This procedure will continue for each point in the current assembly mode test. When a new assembly mode needs to be tested, the mechanism must be disassembled to the motor platform level to start the process again.

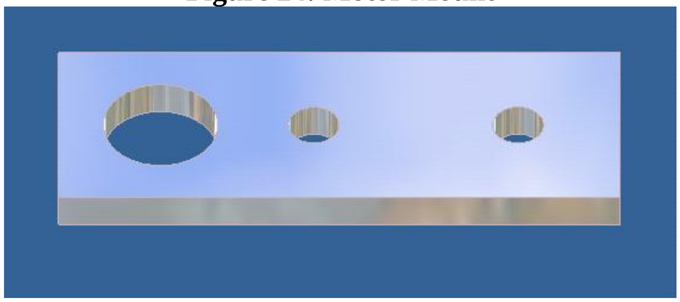
Anti Tilt Caps
The caps employ the sandwich method to prevent tilt. They are created from ¼" thick T6061 aluminum and are 1 ½" in diameter. The center hole will have an 8-32 thread to screw onto the top of the journal. Because contact friction between the platform and cap surfaces has been addressed as a potential concern that might prevent the closing of joint clearances, low friction Teflon tape will be carefully applied to the entire contact surfaces of the cap, platform, and links. The use of an alternative material like Delrin was considered unfavorable since the material had approximately the same coefficients of friction, 0.05, and would cost approximately $32 compared to $15 for the tape. Delrin can also be dangerous to work with, as it emits noxious gas if cut too fast.

Figure 22: Anti tilt cap



Motor Mounts
The motor mounts are created from T6061 aluminum. Because the motor tracks already have a mounting bracket with a pattern of threaded holes on it, simple thru holes can be used to secure the motor mount to the track. The motor mounts are necessary because they allow the motor joints to operate along the same axis since they must be positioned in a staggered arrangement in order to provide a continuous range of motion specified by the predetermined geometry.

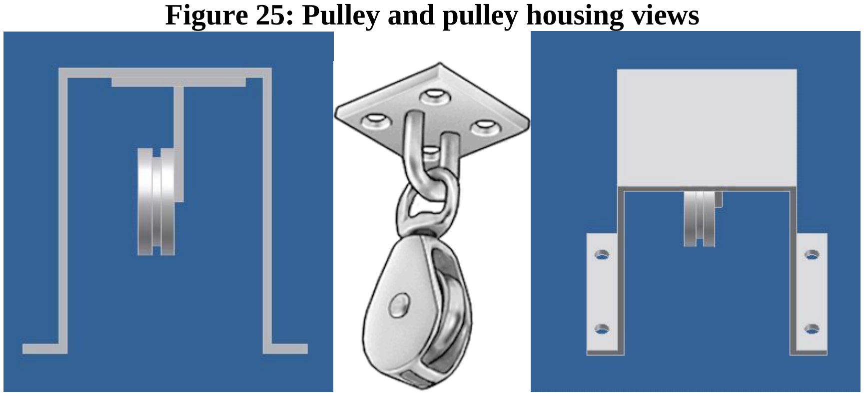
Figure 24: Motor Mount



Pulley Assembly

The pulley assembly is fabricated from 1/8” T6061 aluminum and stands 6” high and 4” wide. The housing is used to suspend a hanging swivel alignment pulley. A 3/8” nylon rope is secured to the platform via the force application knob (shoulder bolt) in the center. The rope is positioned over the pulley and hangs over the edge of the platform in the y-direction, suspending a 5kg mass. The swivel action allows the pulley to rotate as the platform transverses and keeps a constant directional force applied to the platform.

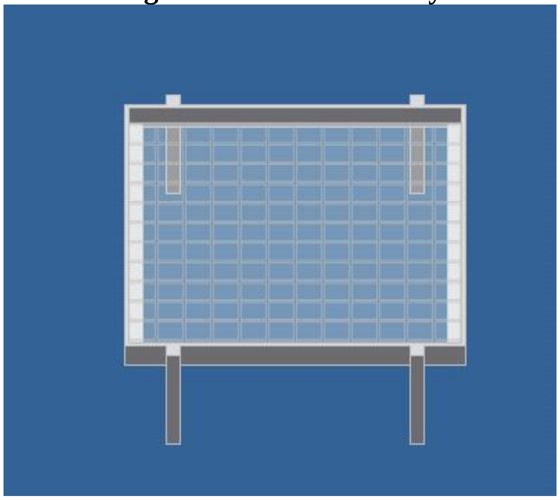
Figure 25: Pulley and pulley housing views



Grid Assembly

The grid assembly provides a reference datum for the users. Technically, the colored markers located on the platform surface are a part of this system, even though they are physically located on the platform. As the platform moves, the camera tracks the motion of the markers and can therefore compute the position of the platform. The grid is suspended over the work area by a 1/8” thick section of plexiglass 12”x12” for rigidity. Plexiglass was chosen over glass because it has better clarity, even at greater thicknesses. The plexiglass holder sits 6” above the plywood surface on 7” long 1”x1/8” legs, and is assembled from 12” length sections of 1” angle bracket. This assembly is fabricated from T6061 aluminum. The bottom of the “L” provides a resting surface for the perimeter of the plexiglass.

Figure 26: Grid Assembly



Camera Assembly
For tracking the marker movements, the Logitech web camera in Figure 27 will be used. The camera will interface the computer via USB connection. Testing the camera's field of view on the required 5”x5” section by varying the height of the camera concluded that the camera should be positioned approximately 7.5” from the platform/marker surface. In order to position the camera at this height over the platform, an aluminum angle bracket assembly will be used with steel supports to increase rigidity. The reason for this choice is that it is simple and inexpensive to construct. The pre-stamped hole pattern in the angle brackets allow for quick and easy camera adjustment, and the stand can be quickly disassembled for transportation. This is important because the stand protrudes out and could be a notable safety concern or become damaged when relocating between buildings. Figure 28 located with the manufacturing plan includes a CAD drawing of this camera structure.

Figure 27: Logitech web camera

Position Program
A Matlab program will be written to accurately determine the platform's position. This program will employ techniques of image tracking. First, we will work with our customer to choose an origin as a reference for all measurements. Then, an image will be inputted into Matlab for platform measurement. Matlab will locate the platform by locating the three different colored platform markers. The x and y location of each marker will be compared to the location of the origin to determine the platform's position. The platform's orientation will be determined through geometrical relations using the x and y

marker positions. After calculating the platform's position and orientation, Matlab will output this data to a text file.

User Safety

As in any design to be manufactured, safety must be considered. There are a few hazards to be aware of during normal user operation including electrical, pinch, and falling object hazards. Due to wires from three linear motors, three strain gauges, and a web camera, special care must be taken to avoid disturbing these paths that could potentially inhibit proper mechanism operation. Additionally, since the parallel mechanism involves moving parts, the user must keep all extremities away from the mechanism while in operation. The plexiglass grid will help remind the user to avoid mechanism contact during procedures. Finally, since a 5 kg weight is suspended from the platform through a pulley toward the ground, the user should take care when walking around the mechanism's base. Walking into the apparatus could lead to inaccuracies in mechanism motion and measurement, and potentially bruise the user.

MANUFACTURING PLAN

The majority of the parts that need to be manufactured will be done so using the water jet machine in the lab. This will leave a relatively rough surface finish, so the first step will be to sand each surface. Next, for the links, the platform, the motor arms, and the anti-tilt caps, we will apply a layer of Teflon tape, to reduce the friction between these parts as they glide across one another. It should be noted that we must manufacture the links first, so that we can send them in to get the strain gauges mounted. There is a one week lead time needed for the strain gauges to be attached before we can perform any testing. For the journals, we will need to lathe down the ends, so that we can thread these parts and allow for them to screw into the links and the anti-tilt caps. The journals with no clearance will be press fit into the bearings with a max/min clearance of 0.0014/0.001". In order to press fit the ball bearings into the platform and motor mounts, there is a max/min clearance of 0.002/0.0002". A detailed list of the parts that we will manufacture and how we will manufacture them is given below in Table 10.

Table 10: List of parts that will be manufactured

Part	Material	Manufacturing Plan
Links	Aluminum with Teflon tape	• Water jet • Drill press • Strain gauge mounting
Platform	Aluminum with Teflon tape	• Water jet • Drill press • Thread tapper
Journals	Steel	• Lathe • Thread tapper
Motor Arms	Aluminum with Teflon tape	• Water jet
Cap	Aluminum with Teflon tape to reduce friction	• Water jet • Drill press • Thread tapper
Camera Stand	Aluminum angle bracket	• Band saw • Drill press

Pulley housing	Aluminum	• Band saw • Drill Press

The rest of the parts in our system do not need to be manufactured. They are listed below in Table 11. The grid that we ordered is too thin to stay up on its own, so we will be placing it into a stand, where it will be held in place with a thin sheet of plexiglass. The pulley will be attached to a housing so that it is allowed to hang from above. This allows us to use a pulley that can swivel as the platform moves. The camera stand will be constructed from an aluminum angle bracket. The motors will be attached to the base from the bottom using four bolts for each motor.

Table 11: List of parts that do not need to be manufactured

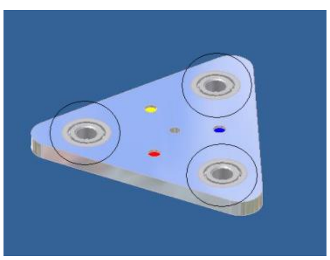
Part	Quantity	Cost ea.	Description
Linear Motor/Track	3	Provided	• Aerotech BSM60-A • Aerotech ATS 100 • Changes position of the platform
Grid	1	$15.00	• Lexan Terrain Mat • One inch grid placed just above platform to determine platform position reference • Held in a stand with a sheet of plexiglass
Engineering weight	1	Provided	• 5 kg hung off of pulley • Applies force to platform in order to close joint clearances
Strain Gauge	10	$4.90	• Omega SGD-1.5/120-LY11 • 2 applied to each link to measure the force in the link
Directional pulley	1	$15.43	• McMaster 3117T6 • Assures that the force from the weight is applied in the correct direction • Attached to a housing so that it can hang down and swivel
Nylon rope	1	$9.42	• McMaster 3819T54 • Priced for 100 ft • Attaches weight to platform through the pulley
Camera	1	Provided	• Logitech V-UW21 • Records the position of the platform
Bubble Level	4	<$5.00	• Measures the tilt of the platform • Purchased at hardware store
Bolt 5/16”-18 standard drive Hex	25	$0.29	• McMaster 92196A581 • Attaches each linear motor to the base • Attaches each motor mount to the motor • 3/4” thread length

Bolt 5/16”-18 Standard drive Hex	25	$0.21	• McMaster 90128A578 • Attaches pulley housing to the base • Attaches camera stand to base • Will be used to assemble camera stand • ½” thread length
Steel Ball Bearing	9	$5.85	• McMaster 6383K34 • Reduces friction at joint locations
Base	1	66.54	• McMaster 8747K162 • 36”x24”x1/2” PVC plate • Provides mounting surface for motors, camera stand, pulley housing and grid

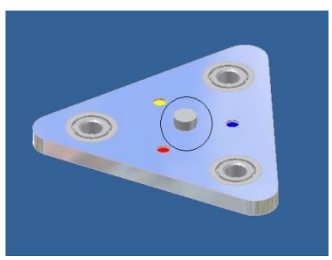
ASSEMBLY PLAN

In order to properly assemble our entire system we must first assemble the following three sub-assemblies: the platform, the motor arms, the links, and the camera stand. Although parts of the system assembly may be put together prior to the sub-assemblies (such as the motors to use for testing), the majority of the mechanism will be assembled after the sub-assemblies are completed.

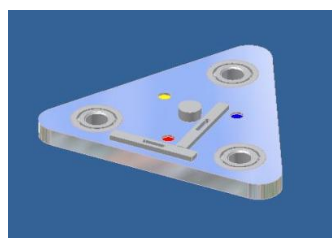
Sub-Assembly: The Platform
To complete the assembly of the platform we must attach the bearings, the knob which holds the weight, and the bubble levels. The sequence of this assembly can be seen in the pictures below.



Step 1: Press fit the bearings into the platform



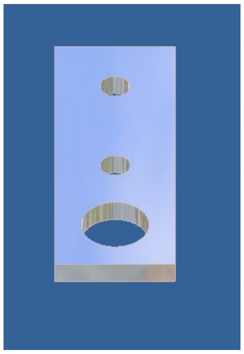
Step 2: Screw in the knob that attaches the weight to the platfor



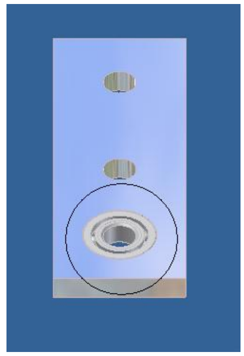
Step 3: Attach the bubble levels

For the platform assembly with no joint clearances (in order to test how the system should work) there will be a step where we press fit the journals into the bearings. This step will come between steps 1 and 2 above.

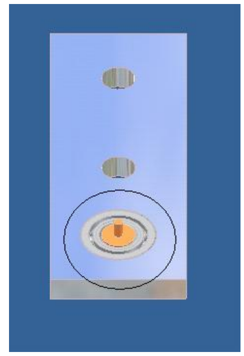
Sub-Assembly: The Motor Arms
To complete the assembly of the motor arms, we must attach the bearings and the journals that provide no clearance to the joint. The sequence of this assembly can be seen in the pictures below.



Motor mount after manufacturing



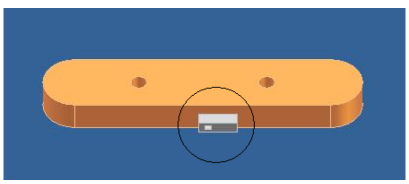
Step 1: Press fit the bearings into the platform



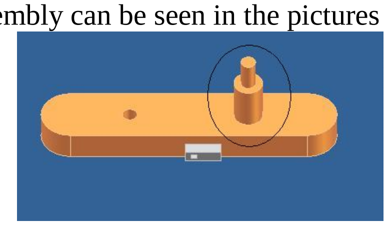
Step 2: Press fit the journals with no clearance into the bearing

Sub-Assembly: The Links

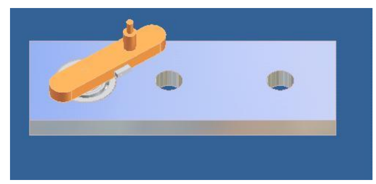
To complete the assembly of the links we must first send them in to have two strain gauges attached to each link. After this is complete, we must attach the journals to the links and then attach the links to the motor arms. The sequence of this assembly can be seen in the pictures below.



Step 1: Attach 2 strain gauges to each link



Step 2: Screw a threaded "clearance" journal into each link

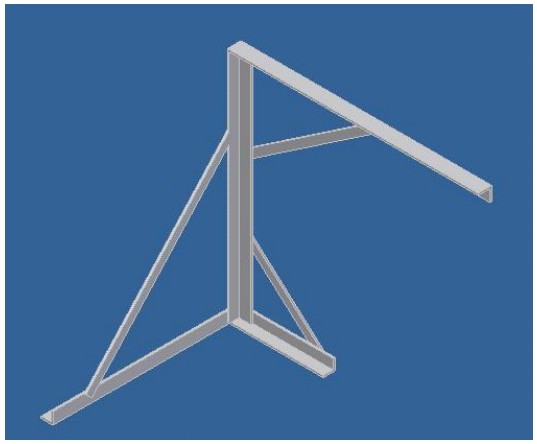


Step 3: Attach each link to one of the motor mounts

Sub-Assembly: Camera Stand

The camera stand is composed of a 16" tall angle bracket positioned vertically, supported by two more angle brackets fastened to the base platform in an L configuration. A ½"x1/8" steel piece will support the vertical bracket from the end of each leg of the L configuration. Another L bracket will extend 10" horizontally from the top of the vertical bracket and will be supported by another angled steel piece. The Logitech camera will be fastened to the end of this horizontal angle bracket to monitor the platform movements. A ¾"x ¾" aluminum angle bracket will be used. This assembly can be seen in Figure 28. Please note that this assembly is portrayed as a 2x4 wood construction in the other figures throughout this report. Due to time constraints, this assembly could not be re-incorporated into each individual screenshot.

Figure 28: Sub-Assembly Camera Stand

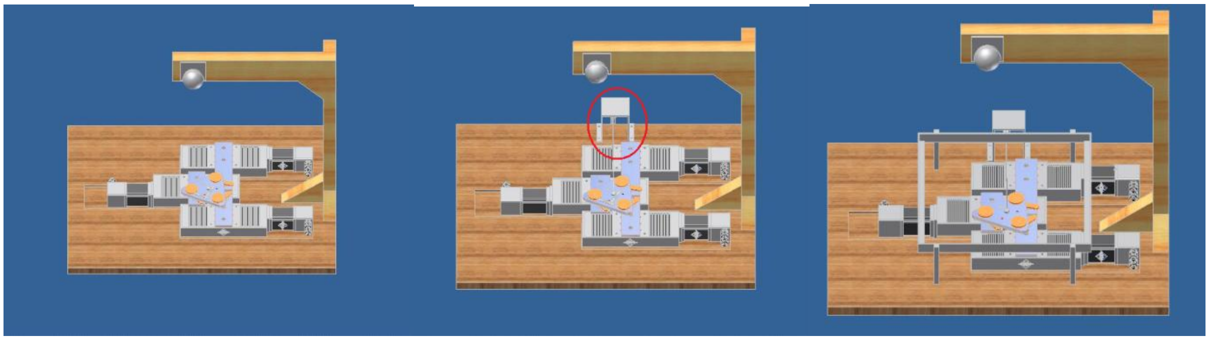


System Assembly
Once all of the sub-assemblies are put together we can begin assembling the system as a whole. The first step is to set the motors into the slots that are cut out of the PVC base. The motors are then bolted onto the PVC base from the bottom. The remaining steps for assembling the system are shown in the pictures below.

Step 3: The motor mount assemblies are bolted on to the motors (above)

Step 4: The platform is set in place by inserting the journals into the bearings (above)

Step 5: The anti-tilt caps are screwed onto the journals (right)



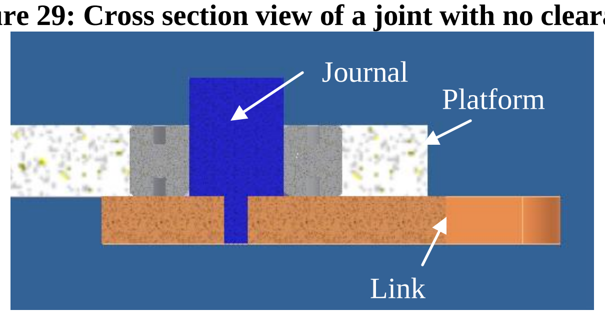
Step 6: The camera stand is set in place over the system

Step 7: The pulley is screwed in place on the PVC base

Step 8: The grid is placed over the system

For the platform that has the journals attached (with no clearance) the journals will be threaded on the bottom so that they can easily screw in and out of the links. This is shown in the cross section view below. This will allow us to easily swap between the platform with joint clearances and the one without clearances.

Figure 29: Cross section view of a joint with no clearance

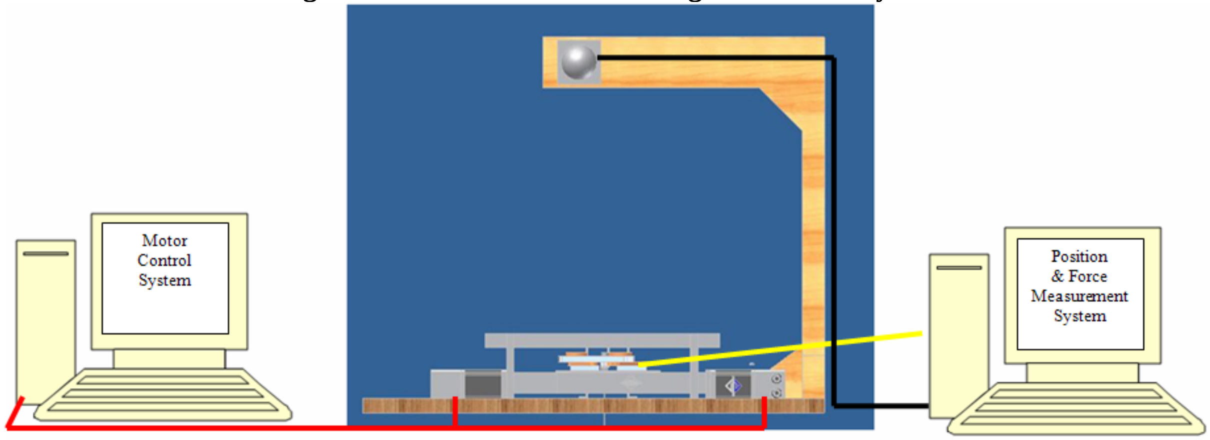


System Integration

In order to control all of the aspects of our system we will need to connect the mechanism to two computers. One computer will be hooked up to the linear motors in order to control the motion of our platform. We will use a program that allows us to input a position that we want our motors to be in and then actually moves the motors to that position. Once they have stopped moving we will use another computer that is hooked up to the camera and the strain gauges. This computer will take a picture of the platform and take measurements for the forces in the links while the system is static. Once these measurements are taken we will input the next position that we want the motors to move to and repeat the above steps. A schematic of the system is shown in Figure 30.

There are a lot of wires associated with the mechanism including motor wires, strain gauge wires, and the camera wires. Special care will be taken during testing to ensure the strain gauge wiring does not get in the way of the mechanism motion. The motor wires will be fed under the base and will therefore not be a problem and the camera wiring will be attached to the camera stand.

Figure 30: Schematic for the integration of the system



VALIDATION APPROACH

In order to demonstrate to the customer that all user needs and engineering specifications have been met, a validation plan was created. Table 12 matches each engineering specification with its designed validation approach.

Table 12: Engineering Specifications and Validation Approaches

Engineering Specifications	Validation Approach
• Joint clearance = 0.125x, x=1,2,3…	• Measure with caliper
• Platform must have 3 DOF (x,y,Ɵ) • Platform must not rotate about x or y axes more than 1/2 degree	• Visual inspection • Two bubble levels instead of dual axis tilt indicator
• Platform position determined to within 2% of the total motor displacement in x,y, and theta	• Compare the platform position with no joint clearances to the platform position with joint clearances
• Link forces determined to within 3% of weight applied	• Compare link forces to static calculation results
• Predetermined dimensions	• Measure with ruler and caliper
• 3 motors required	• Visual inspection
• Force required to close joint clearances	• Visual inspection

In order to verify the predetermined scaled joint clearance, the bearing inner diameter and the journal diameter will be measured with a caliper. The joint clearance is confirmed by matching half the difference between the bearing inner diameter and the journal diameter with the predetermined joint clearance of 0.125.

A visual inspection will be performed to determine the platform's motion in three degrees of freedom in the x, y, and theta directions. The engineering specification requiring that the platform must not rotate about the x or y axes more than a half of degree will be verified using two bubble levels. If the bubbles remain in the middle of the level region during all mechanism motion, the engineering specification will be confirmed. The bubble levels provide a cost and spatial effective way of determining the platforms orientation even though they are unable to measure the exact angle of potential rotation. Tilt indicators were investigated to validate this specification completely. However, tilt indicators do not have the required resolution to measure this specification at a small enough size to be mounted on the platform.

Determining the accuracy of the parallel mechanism is a major priority. In order to determine if the platform's position is within 2% of the total motor displacement, two mechanisms are compared. The first mechanism will be free of joint clearances, while the second mechanism will contain the joint clearance specified in the engineering specifications. The two mechanisms will follow the same robot path and resulting positions will be compared. If the platform position with and without joint clearances match within 2% of each other, accuracy has been achieved.

In order to determine if link forces are within 3% of the applied weight, the forces in the links during mechanism operation will be compared to the expected forces in the links from static calculations. If the difference in these forces is less than or equal to 3%, then accurate force measurement has been achieved.

In order to verify the mechanism matches the predetermined dimensions, a ruler and caliper will be used to verify the dimensions are identical to the predetermined geometry dimensions after scaling.

A visual inspection will be performed to determine the engineering specifications requiring 3 motors and a force required to close the joint clearances. If 3 motors are present and used in operating the mechanism, then the engineering specification requiring 3 motors has been met. If an external force closes the joint clearances through visual inspection prior to mounting the caps, then the corresponding engineering specification has also been met.

VALIDATION RESULTS

This section will give the results of the validation approaches discussed in the section above. Results will include validations of the following design specifications:

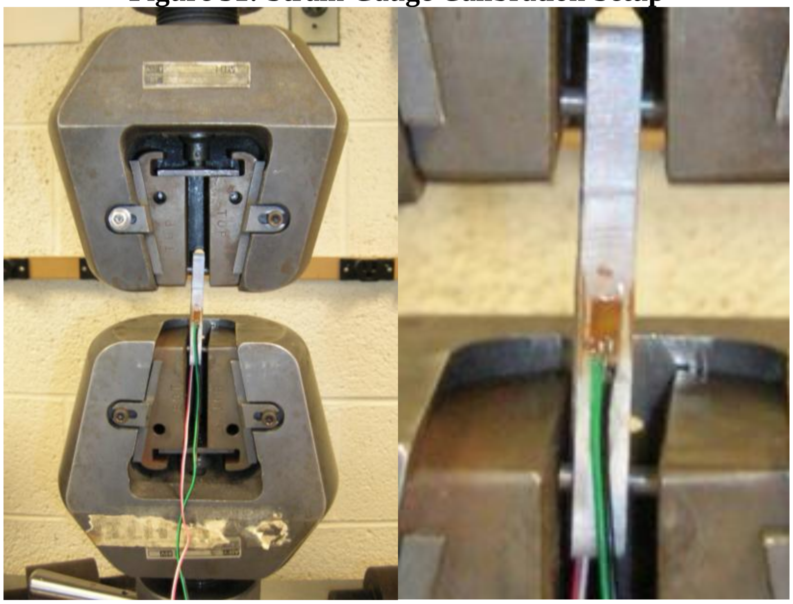
- Platform has three degrees of freedom
- Three linear motors are used
- External force closes the joint clearances
- Predetermined dimensions achieved
- Platform does not tilt more than ½ degree
- Link forces determined to within 3% of weight applied
- Platform position determined to within 2% of total motor displacement

Prior to testing, calipers were used to validate that the predetermined dimensions including joint clearance were achieved. We validated by visual inspection during mechanism testing that the platform is able to move with three degrees of freedom, that three linear motors are used, and that the external force closes the joint clearances. Also during testing, we visually inspected the bubble levels attached to each platform and verified that the bubbles remained in the same location throughout the entirety of each test we ran.

This can be seen by examining the series of pictures taken by the camera for each motor position in the program.

Link Forces
To determine the link forces using strain gauges, we performed calibration testing for each of the four links using an Instron machine with the setup shown in Figure 31. During initial calibration testing, two of the four links broke due to equipment failures. Because we only manufactured one extra link, we had to order more strain gauges, manufacture additional links, remount the strain gauges, and repeat this calibration testing. Five additional links with two strain gauges each were made to account for any additional issues with the testing. Before beginning the second round of calibrations, we tested a link lacking strain gauges to ensure the equipment and program were functioning properly before testing the strain gauge links as a preventative measure.

Figure 31: Strain Gauge Calibration Setup



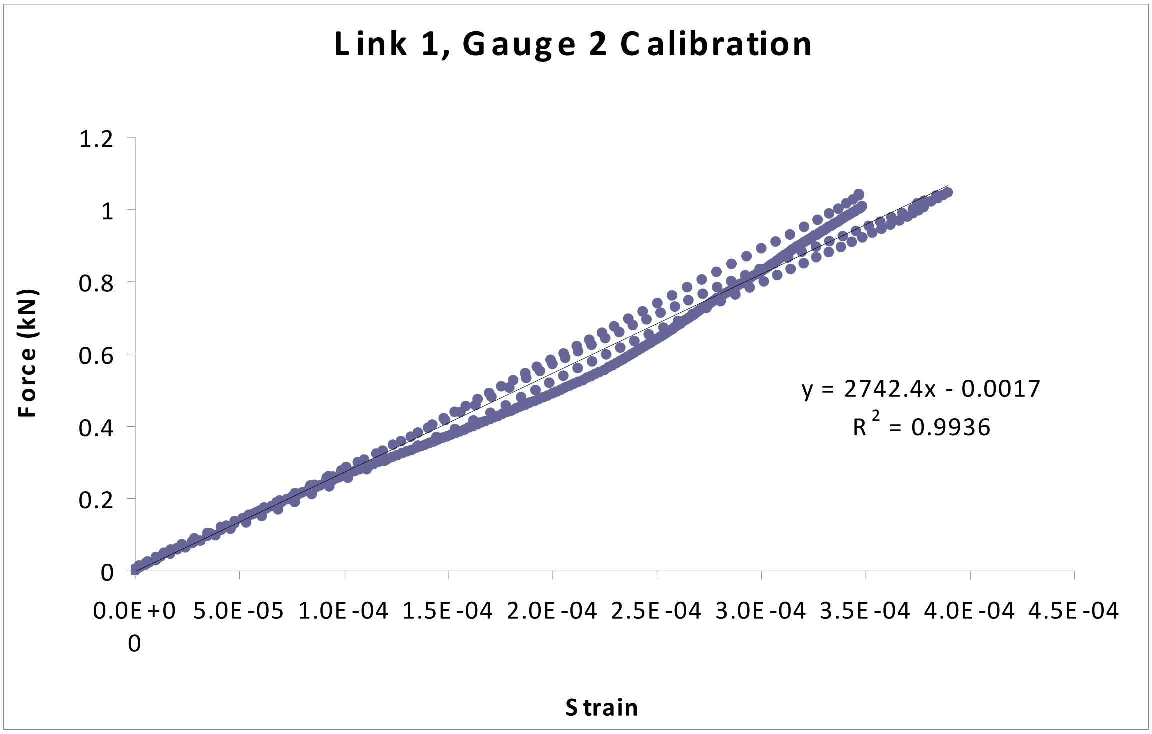
In order to determine which links and strain gauges to use for our mechanism testing, we calculated the error of the link forces based on the difference between the actual force measurement and the expected strain. The table below summarizes the errors for each link.

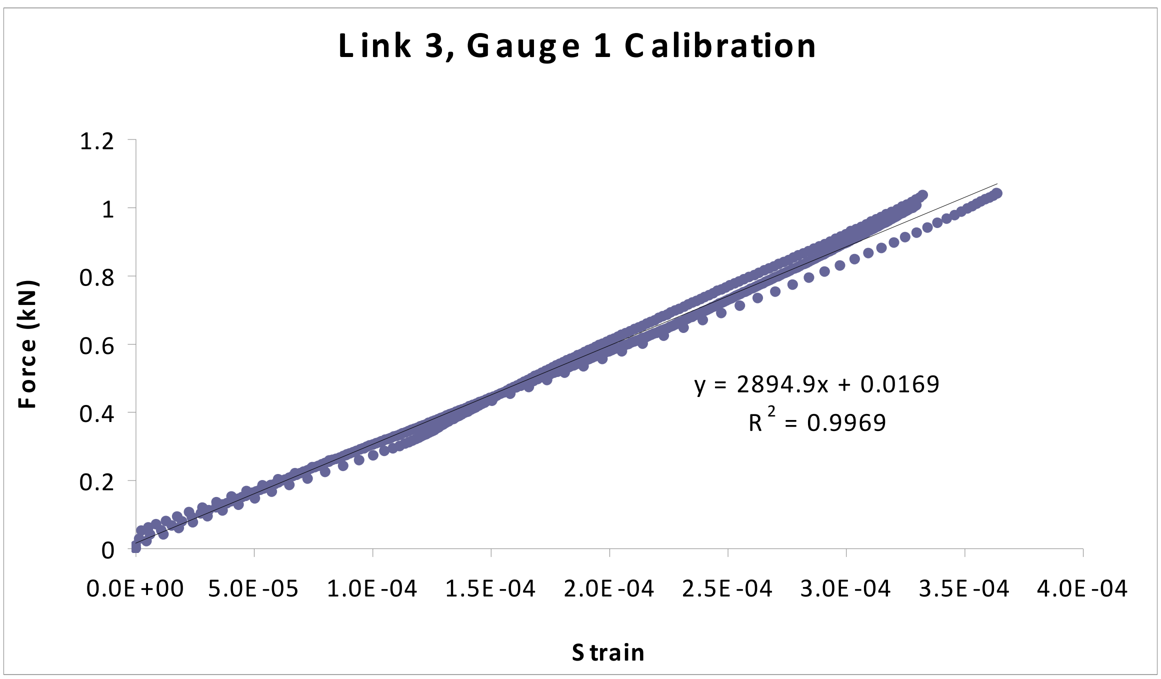
Table 13: Summary of Strain Gauge Error

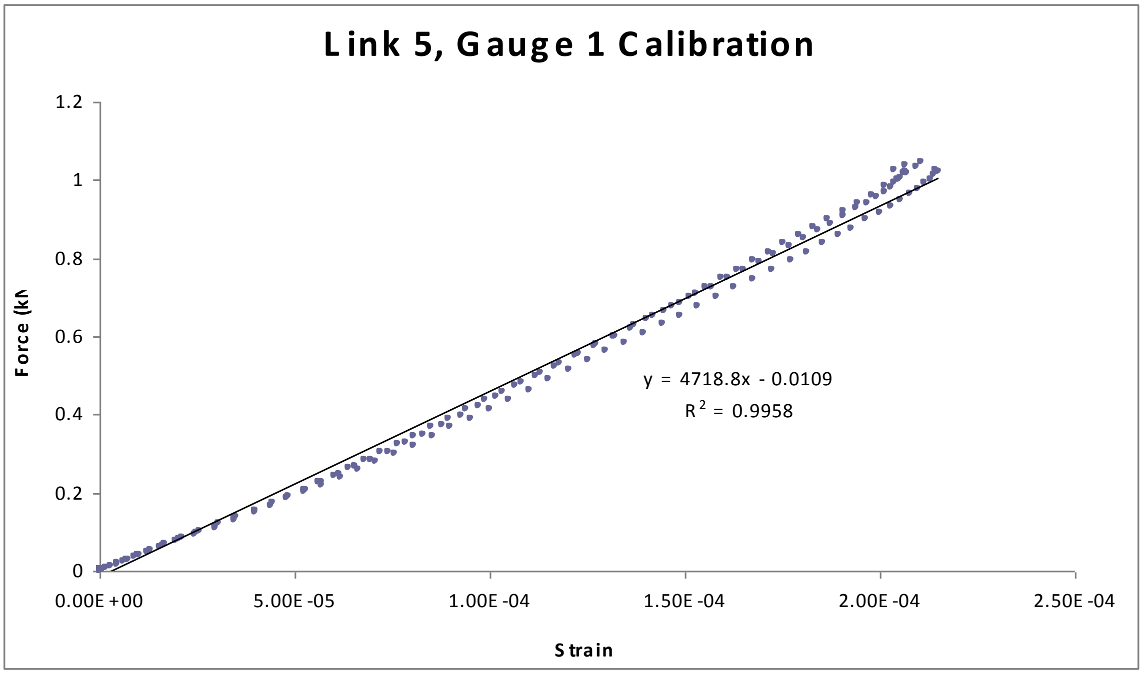
	ERROR (based on theoretical value)			
	Strain 1 (%)		Strain 2 (%)	
Link	F=1kN	F=0.2kN	F=1kN	F=0.2kN
1	3.4	12.9	4.7	5.1
2	10.3	11.9	51.6	119
3	6.5	9.6	16	22.2
4	25.2	30.9	34.6	30.9
5	42.7	27.9	5.4	6.6

To minimize these errors, we chose to use links 1, 3 and 5 and strain gauges 2, 1 and 2, respectively for the mechanism. It should be noted that during the mechanism testing, strain gauge 2 on link 5 did not zero properly. Therefore, we use strain gauge 1 on link 5 instead. Although the error for strain gauge 1 is much larger than strain gauge 2, the calibration curve is linear meaning the link still exhibits strain as expected. Calibration curves for each link and associated strain gauge can be seen in the figures below.

Figures 32 – 34: Calibration Curves for Strain Gauges with Minimal Error

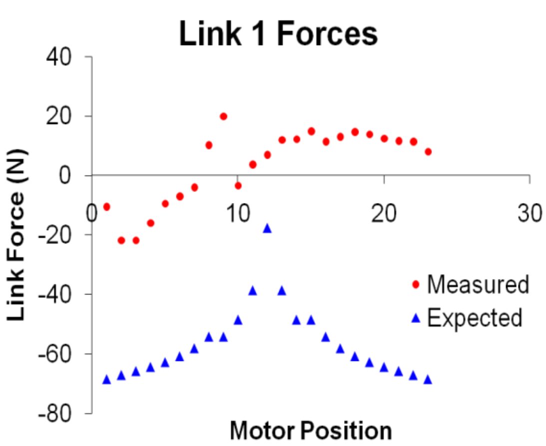


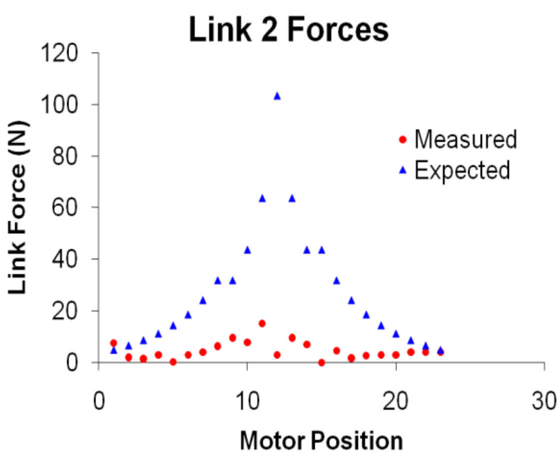


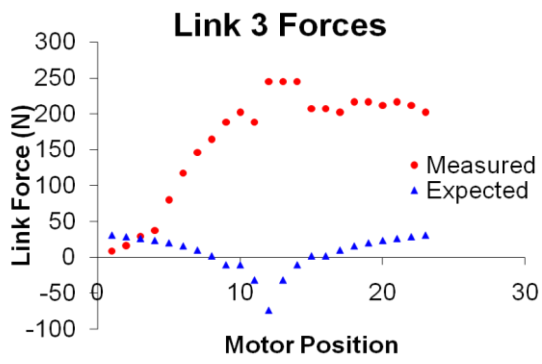


During mechanism testing, we recorded the strain on each link for each motor position in the program. We then used the calibration curves above to determine the forces in each link. Next, we graphed forces seen in each link with the expected forces based on static calculations and found errors that far exceeded what we set out in our engineering specification. Note that Links 1, 2 and 3 refer to the links attached to motors 1, 2 and 3, respectively.

Figures 35-37: Measured vs. Expected Link Forces







The forces measured in the links during testing were not accurately determined because of strain gauge sensitivity at the low forces we expected to see. We decreased the external weight on the mechanism to account for both the maximum producible force in the linear motors as well as the small scale of the mechanism. Due to this, the forces we expected to see were reduced and the strain gauges were not capable of measuring forces of this magnitude. This inaccuracy can also be partially attributed to the static calculations assuming no joint clearance; however, this error is much less significant than the error associated with strain gauge sensitivity.

Platform Positioning

In order to validate that the parallel mechanism with large joint clearances was able to achieve precise motion, we designed, built, and tested a parallel mechanism with large joint clearances and a parallel mechanism without joint clearances. We compared the resulting platform position and orientation of these mechanisms after conducting tests designed to produce identical paths of motion. A total of 8 tests were conducted with varying paths of motion, external force direction, and platform type (clearance versus no clearance). Table 14 identifies the 8 tests conducted.

Table 14: 8 Primary Tests Conducted

Test Number	Platform Type	External Force Direction	Path
1	No Clearance	-y	Path1_no_clearance.bat
2	No Clearance	-y	Path3_no_clearance.bat
3	With Clearance	-y	Path1_clearance_minus.bat
4	With Clearance	-y	Path3_clearance.bat
5	No Clearance	+y	Path1_no_clearance.bat
6	No Clearance	+y	Path3_no_clearance.bat
7	With Clearance	+y	Path1_clearance_plus.bat
8	With Clearance	+y	Path3_clearance.bat

During each test, the three linear motors were moved to their pre-determined locations using the specified path file. A photograph of the platform was taken at each position during the path. After completing all tests, the photographs were run through a Matlab marker identification program. This program outputs the corresponding x and y pixel locations of the three red markers in each photograph. After manually entering the x and y pixel locations into an excel spread sheet, the spread sheet uses a variety of pre-determined functions to calculate and output platform position and orientation. More details on the pre-determined functions used for calculating position can be found in Appendix H.

For the purposes of our project, we were asked to pick one path and external force direction to validate the results of platform position accuracy. We chose to compare tests number 5 and 7. These tests explore the accuracy of platform position using Path 1 and the external force in the positive y direction. We plotted the platform distance traveled in the y direction versus the platform distance traveled in the x direction and the platform rotation versus the platform distance traveled in the y direction for the clearance and no clearance platform. The resulting plots can be seen in Figure 38 and Figure 39

Figure 38: Accuracy in platform position despite large joint clearances

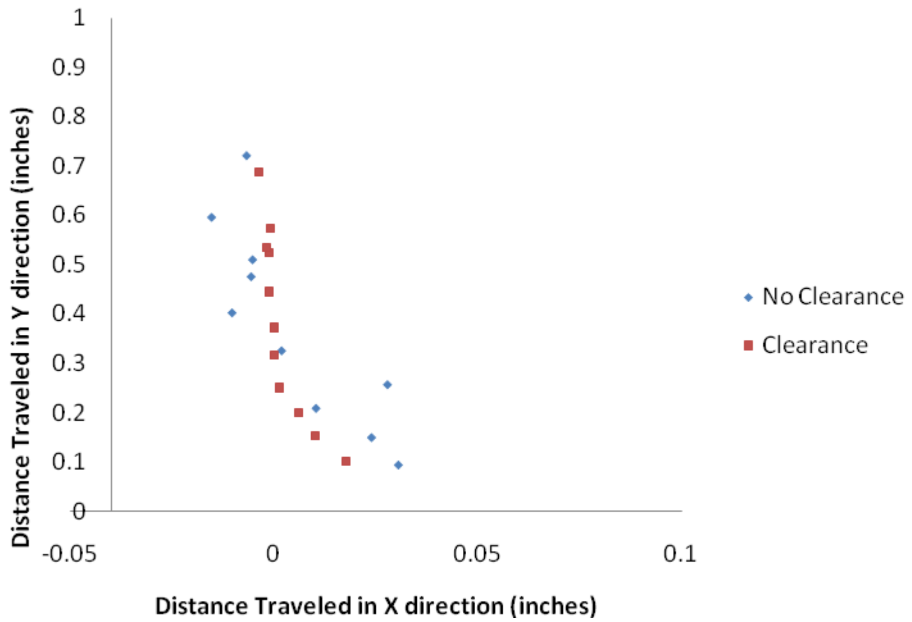
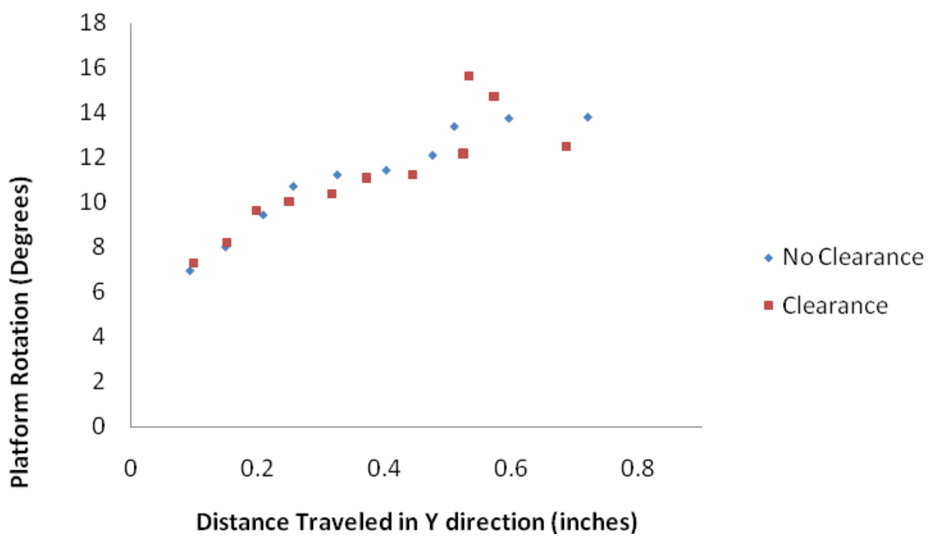


Figure 39: Accuracy in platform orientation despite large joint clearances



From these graphs, we found that the platform with joint clearances was able to perform the same maneuvers as the platform without joint clearances with an average error of 4.1% in the x direction, an average error of 3.0% in the y direction and an average error of 6.4% for Ɵ, rotation about the Z axis. Individual errors varied from 0.7-10.7%, however average error is more indicative of accuracy because motor program 5 completed its maneuver in 21 steps and program 7 used 23 steps. Therefore only a sample of the points could be used for error analysis. The error was calculated by comparing the difference in distance traveled between each point to the total distance traveled by the farthest moving motor over the course of the entire maneuver.

Originally, the customer requirement was to achieve an accuracy of 2% of total motor displacement for platform movement. A later discussion with our sponsor revealed that operating with an error of 2% was not necessary for the project scope and might be unattainable. It was described that the key to validating this requirement was still to achieve high accuracy; however a replacement validation percentage was not provided as this is a research project and there is no particular benchmark. For this reason, we determined that achieving between 5 and 8% accuracy would be sufficient to satisfy this requirement. Because we achieved errors of 3.0% for y, 4.1% for x and 6.4% for Ɵ we feel that we have achieved accurate platform movement despite large joint clearances.

DISCUSSION

In reflection over the project, there are several aspects of the design our team would change. First and most importantly, we would use different linear motors with a larger range of motion. This would allow the size of the mechanism to be scaled larger which would reduce the difficulty in manufacturing and allow for a more robust design. A major concern for our project was maintaining strength despite the small scale. For example, we would not have needed to thread the journals if the project were larger in size. Instead, we could have counter-sunk the links and used a bolt to attach the journals. This would have significantly reduced manufacturing time and it would also increase the journal strength when the external force is applied. The increase strength of the links and journals would also allow us to apply a larger external force which in turn would produce a more accurate strain reading and force measurement.

In addition to the reduced manufacturing complexity and increased mechanism strength, having a larger mechanism would also allow for the use of tilt indicators (instead of bubble levels) on the platform. With tilt indicators, we would be able to more accurately validate the anti tilt engineering specification.

Other, less significant changes include making clear caps so the joint clearance is visible during mechanism motion (we plan to accomplish this for our sponsor before the end of the semester), using a pulley that does not allow the weight to rotate it inward, making a cleaner looking camera and pulley stand, and making multiple extra links initially for testing and calibration to avoid remanufacturing and testing under such a short timeline.

RECOMMENDATIONS

In order to determine accurate link forces, our team recommends repeating the mechanism testing using a strain gauge reading program that is better suited to the purpose of this project. We were unable to do this due to time and budget constraints. As an alternative, a larger weight could be used to decrease the strain gauge sensitivity and attempt to more accurately measure the link strain; however, there is a risk that the journals may break in doing this. We would only suggest resorting to this solution after all additional mechanism testing is completed. If these solutions do not produce the desired results, it may be necessary to build an additional mechanism on a larger scale.

CONCLUSIONS

To design and manufacture a parallel mechanism that effectively demonstrates accurate positioning with large joint clearances, we developed engineering specifications, performed initial static calculations and developed design concepts to satisfy the specifications. We then developed an evaluation process to select the optimal design solutions and incorporated these into the final design. We performed engineering analysis to prove the functionality of the design concepts and validation activities to ensure the design specifications were met.

Using a web camera and three position markers on the platform, we achieved accurate positioning with 4.1% average error in the x direction, 3.0% average error in the y direction, and 6.4% average error for Θ, rotation about the Z axis. We were unable to accurately determine the link forces due to strain gauge sensitivity. The remaining engineering specifications including anti tilt, three degrees of freedom using three linear motors, maintaining a predetermined geometry, and applying an external force to close the joint clearances were also satisfied.

ACKNOWLEDGEMENTS

Our team would like thank Professor Yoram Koren , Hagay Bamberger , and Dan Johnson for their continuous support throughout the project. We would also like to thank Tom Bress and Todd Wilber for their help with strain gauge implementation, Saikrishnan Ramachandran for his help with our positioning program, Julie DeFilippo for purchasing support, and Bob Coury, Marv Cressey and Steve Erskine for their help with manufacturing.

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APPENDIX A: BILL OF MATERIALS

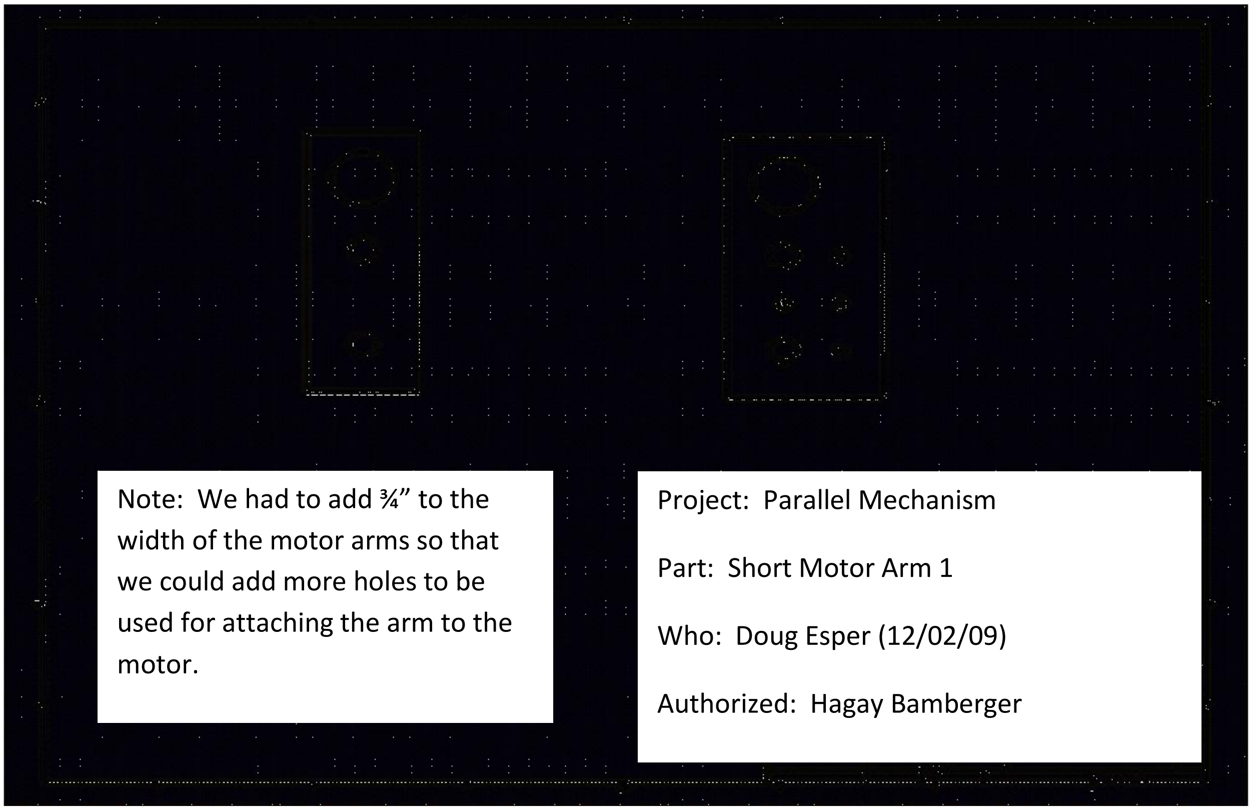
Item	Quantity	Source	Catalog Number	Cost (total)	Contact	Notes
Linear Motor	3	RMS Lab	None	None	Aerotech.com	
5.46 kg Weight	1	RMS Lab	None	None	Steve Erskin	
Strain Gauge	10	Omega	SGD-1.5/120-LY11	$4.90	Omega.com	
Directional Pulley	1	McMaster-Carr	3117T6	$15.43	mcmaster.com	
¼” Diameter Nylon Rope	1	Home Depot	386523	$1.05	homedepot.com	5 Feet
Camera	1	Logitech	V-UW21	None	Logitech.com	
Bubble Level	4	Home Depot	LNLVL2PKGLIM	$5.14	homedepot.com	
Bolt 5/16”-18 standard drive Hex	25	McMaster-Carr	92196A581	$7.25	mcmaster.com	¾” thread
Bolt 5/16”-18 Standard drive Hex	25	McMaster-Carr	90128A578	$5.25	mcmaster.com	½” thread
Steel Ball Bearing	9	McMaster-Carr	6383K34	$52.65	Mcmaster.com	
5/8” Hex Bolt	2	Home Depot	655473	$1.68	homedepot.com	

5/8” Cut Washer	2	Home Depot	668192	$0.62	homedepot.com	
PVC Base	1	McMaster-Carr	8747K162	$66.54	mcmaster.com	36”x24”x1/2”
Links	3	Bob’s Shop	None	None	None	¼” Thick Aluminum
Platform	2	Bob’s Shop	None	None	None	3/8” Thick Aluminum
½” Diameter 12” Steel Rod	1	McMaster-Carr	8893K451	$4.65	mcmaster.com	Cut into 6 no-clearance journals
¼” Diameter 12” Steel Rod	1	McMaster-Carr	4347T41	$15.73	mcmaster.com	Cut into 3 clearance journals
Motor Arms	3	Bob’s Shop	None	None	None	3/8” Thick Aluminum
Caps	3	Bob’s Shop	None	None	None	¼” Thick Aluminum
Camera Stand	1	Bob’s Shop	None	None	None	Built from Steel Angle Brackets
Pulley Housing	1	Bob’s Shop	None	None	None	Built from 1/8” Aluminum and Steel Angle Brackets
Teflon Tape	1 Roll	McMaster-Carr	7801A14	$15.20	mcmaster.com	15 – 2” x2” squares
¼” Bolt	25	Bob’s Shop	None	None	None	Used in camera stand and Pulley

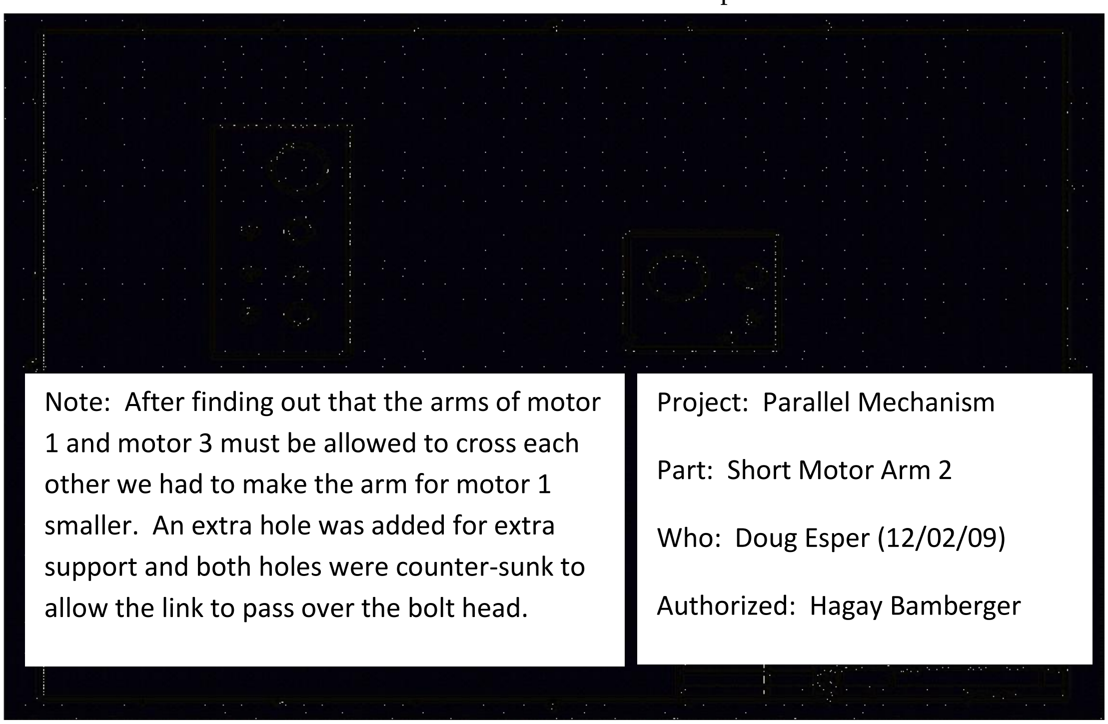
¼" Washer	25	Bob's Shop	None	None	None	Used in camera stand and Pulley
¼" Nut	25	Bob's Shop	None	None	None	Used in camera stand and Pulley
Washer	9	Bob's Shop	None	None	None	Used as shims for motor arms
Bolt	3	Bob's Shop	None	None	None	Used for position markers
Red Paint	1	Bob's Shop	None	None	None	Used for position markers
1/4" Screw	12	Bob's Shop	None	None	None	Secured camera stand and pulley to base

APPENDIX B: ENGINEERING CHANGES SINCE DESIGN REVIEW #3

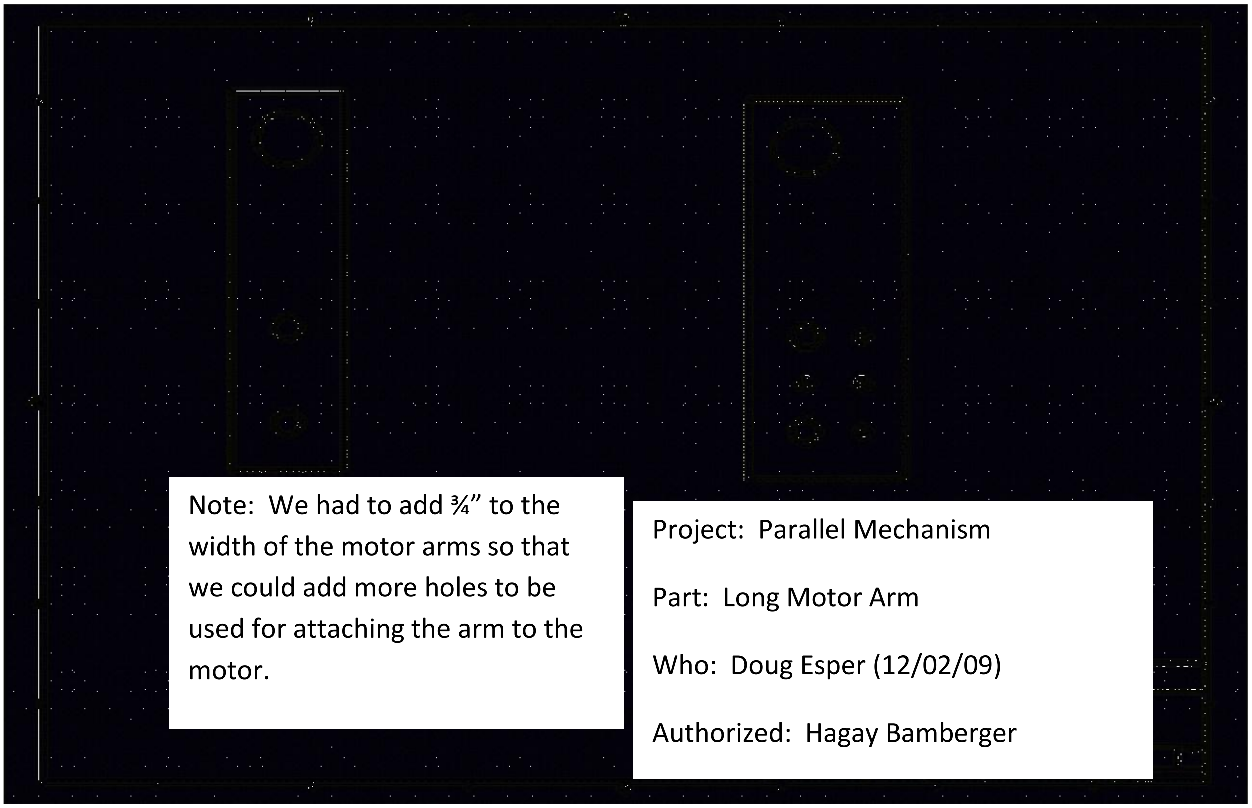
B.1 Short motor arm changed to allow for better attachment to motor



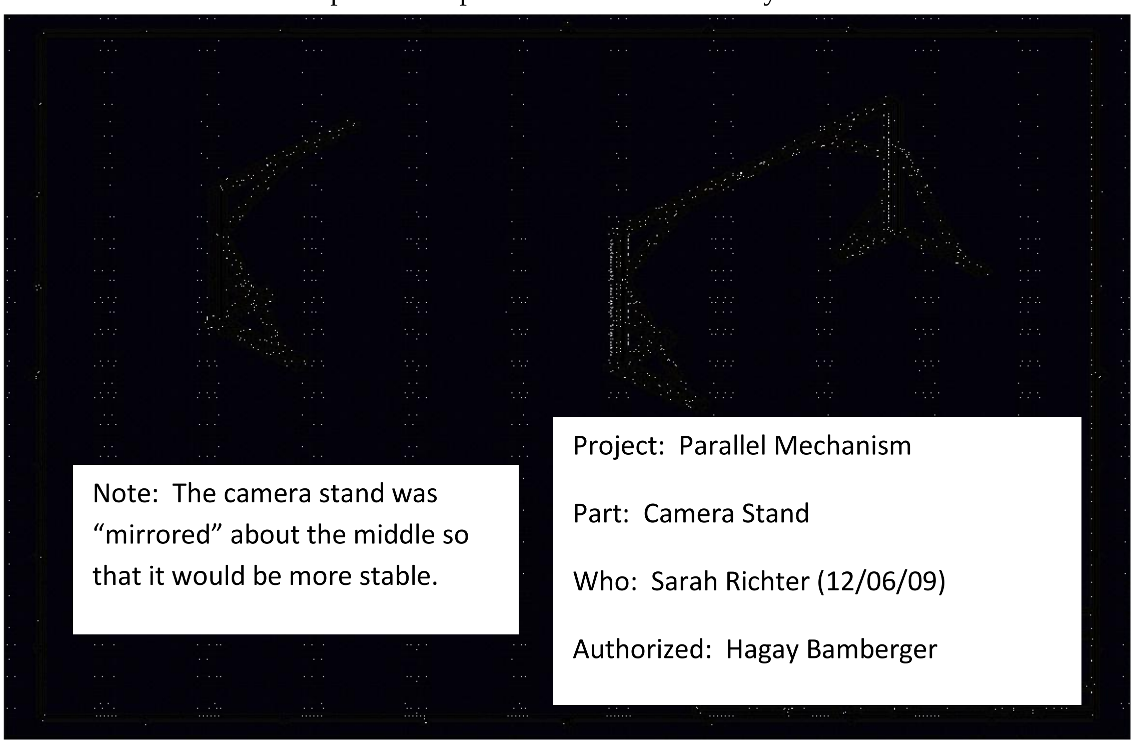
B.2 Short motor arm for motor 1 cut to allow motor 3 arm to pass



B.3 Long motor arm changed to allow for better attachment to motor



B.4 Camera stand made to span whole platform for better stability



B.5 Shims added to motor arms

In order to allow the motor arms to easily pass over the other motors during movement, we added washer between the motor and the arms. These washers acted as shims, raising the arms above the motors enough for them to clear. This change impacted all of the motor arms in our parallel mechanism and was done by Doug Esper on 12/04/09. The change was authorized by Hagay Bamberger.

B.6 Markers added as bolts

Originally the three markers on our platform were round pieces of pink paper that were taped over the holes in the platform. We changed these to be small bolts that could be threaded into those platform holes. The heads of these bolts were painted red so that the platform position program could recognize them. This change made the positioning of the platform more precise. This change impacted the platform in our parallel mechanism and was done by Beth Kovacic on 12/06/09. The change was authorized by Hagay Bamberger.

B.7 Middle bolt added

At the time of design review #3 we had not determined yet what type of bolt we were going to use to attach the weight to the platform. We used a 5/8" bolt that was threaded into a hole in the center of the platform, and kept in place with a 5/8" nut. This change impacted the platform in our parallel mechanism and was done by Brandon DeMars on 12/06/09. The change was authorized by Hagay Bamberger.

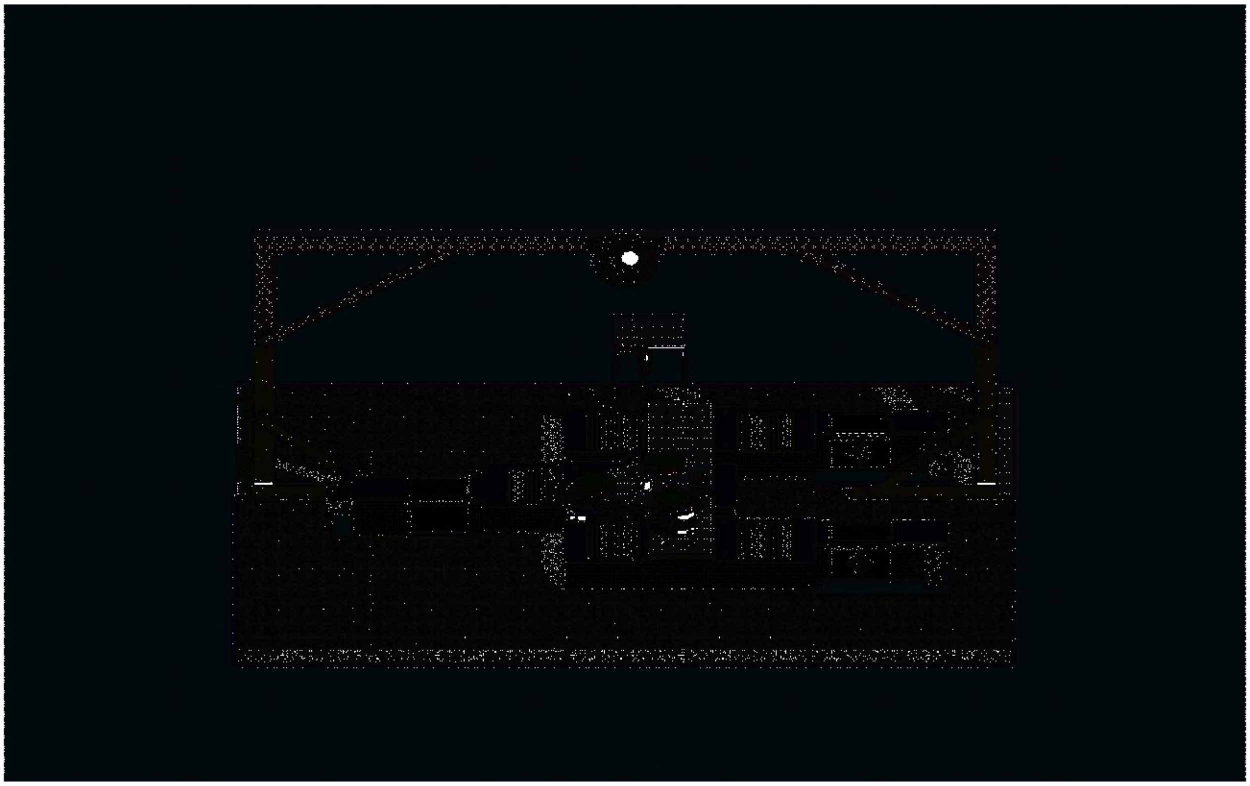
B.8 Grid Removed

During design review #3 we were not sure how the platform positioning program would work. Originally we were going to place a grid over the system as a reference for the movement of the platform. We later found out that the position program worked fine without the grid, so we removed it to make the system simpler. This change impacted our parallel mechanism as a whole and was done by Doug Esper on 12/06/09. The change was authorized by Hagay Bamberger.

B.9 Caps changed to a clear material

After performing our tests we realized that it was difficult to see what was happening in the joint with clearances when our anti-tilt caps were in place. In order to allow us to see the joint we decided to change the cap material to plexiglass. They will be held on completely by the nut, as opposed to them being threaded as in the original design. This change impacted the anti-tilt caps in our parallel mechanism and will be done by Sarah Richter on 12/15/09. The change was authorized by Hagay Bamberger.

B.10 Final design after implemented changes



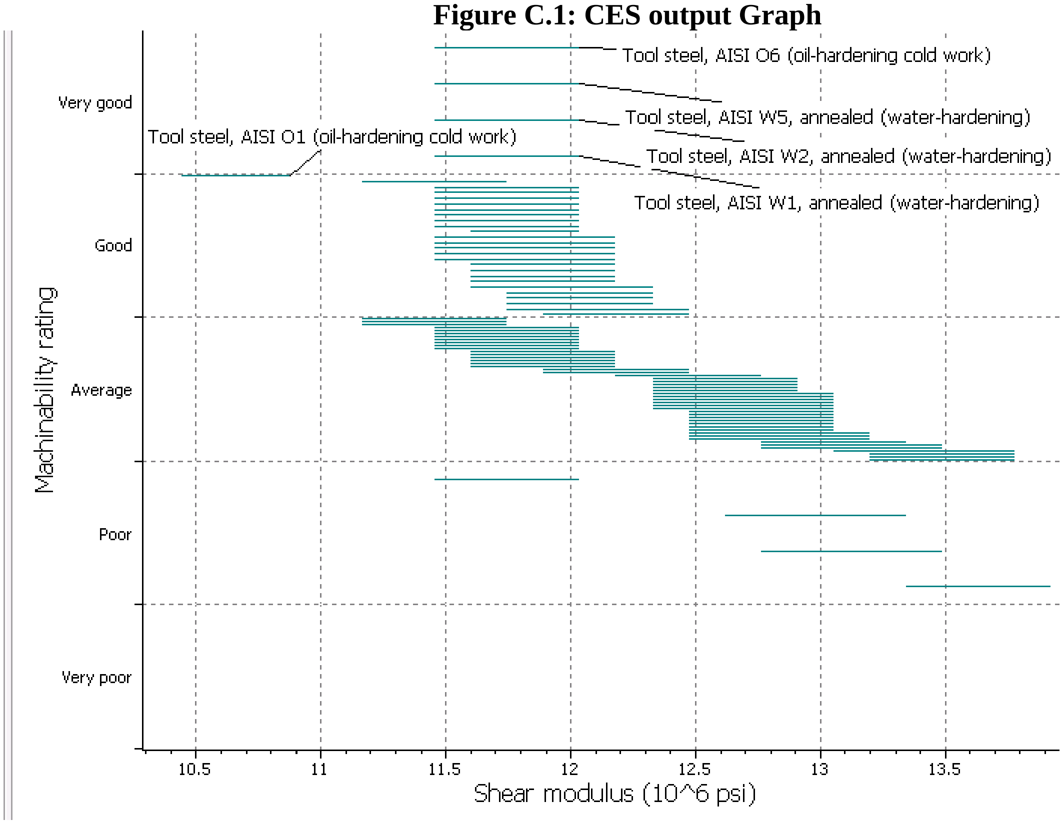
APPENDIX C: DESIGN ANALYSIS

Material selection for Link and Journal

Journal
Function: To facilitate revolute joint rotation
Objective: High strength, good machineability, low cost
Constraints: Fixed Diameter, cannot fail due to shear force or become deformed
Material Indices: Shear Modulus and Machineability Rating

Top 5 Materials
Tool steels;
1. AISI 06, (oil-hardening cold work)
2. AISI W5, annealed (water-hardening)
3. AISI W2, annealed (water-hardening)
4. AISI W1, annealed (water-hardening)
5. AISI O1, (oil-hardening cold work)

Figure C.1: CES output Graph

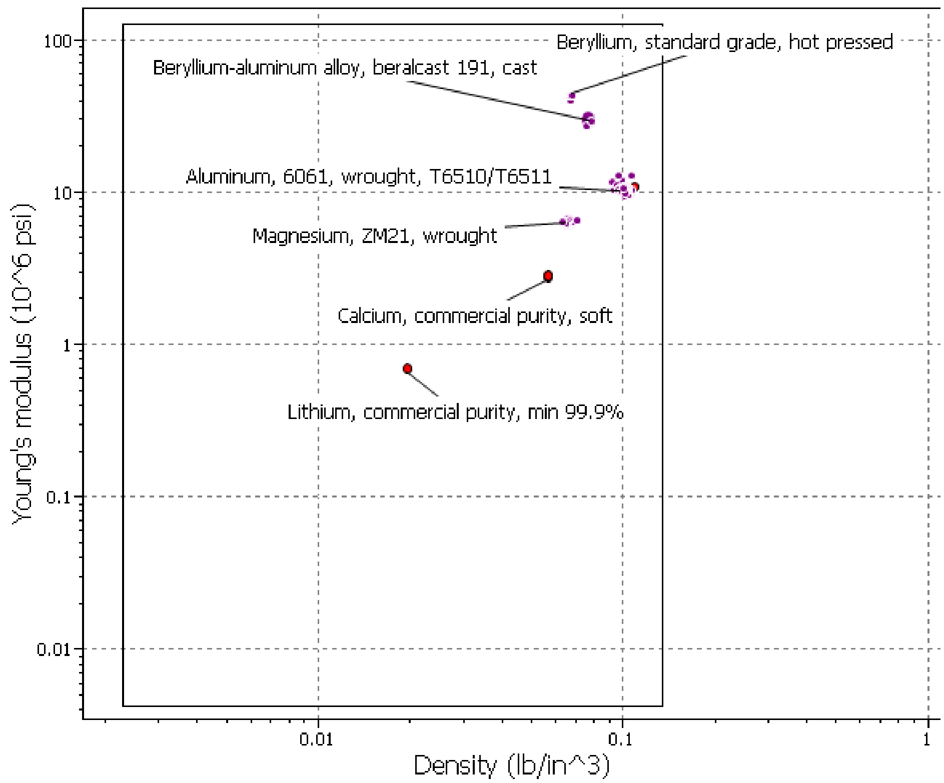


Journal Choice and Justification
For the Journals we chose to use multipurpose oil hardened O1 Tool steel C60-62. The reason is that at the time of order it was readily available to order from McMaster in both 0.5" and 0.25" diameter stock and lead time was our greatest concern. Although the Shear modulus is lower when compared to the other top candidates, 11 MPa for shear modulus is more than adequate assurance for our needs.

Link
Function: Translates the motor movements into platform motion, and provides a location to collect strain data
Objective: Capable of high resolution strain readings, machineable, low cost, lightweight
Constraints: Fixed length, must be capable of having strain gauges mounted on it
Material Indices: Young's Modulus and density

Top 5 Materials
1. Lithium $30-50/lb Max service temperature 80.6 deg F
2. Calcium $4.20-5.60/lb max tensile strength 7.25 ksi compressive 2.32 ksi Highly flamable
3. Magnesium $2.32-2.52/lb
4. Aluminum $0.71-0.79/lb
5. Beryllium or Beryllium-Aluminum Alloy $221-243/lb

Figure C.2: CES output graph



Link Choice and Justification
For the links we chose to use aluminum. Of the comparable materials, Lithium has a maximum service temperature of 80.6 deg F. Calcium has a low maximum tensile strength and an even lower compressive strength which is not preferred for a component that will be under both tensile and compressive loads, it is also highly flammable. Beryllium/Beryllium-Aluminum alloys, and magnesium both offer similar to better properties when compared with aluminum, however they are 342 and 3.6 times more expensive, respectively. Also, the strain gauge choice we had initially made noted either aluminum or steel were the preferred mounting materials.

SimaPro exercise

Figure C.3: Relative Impacts in Disaggregated Damage Categories

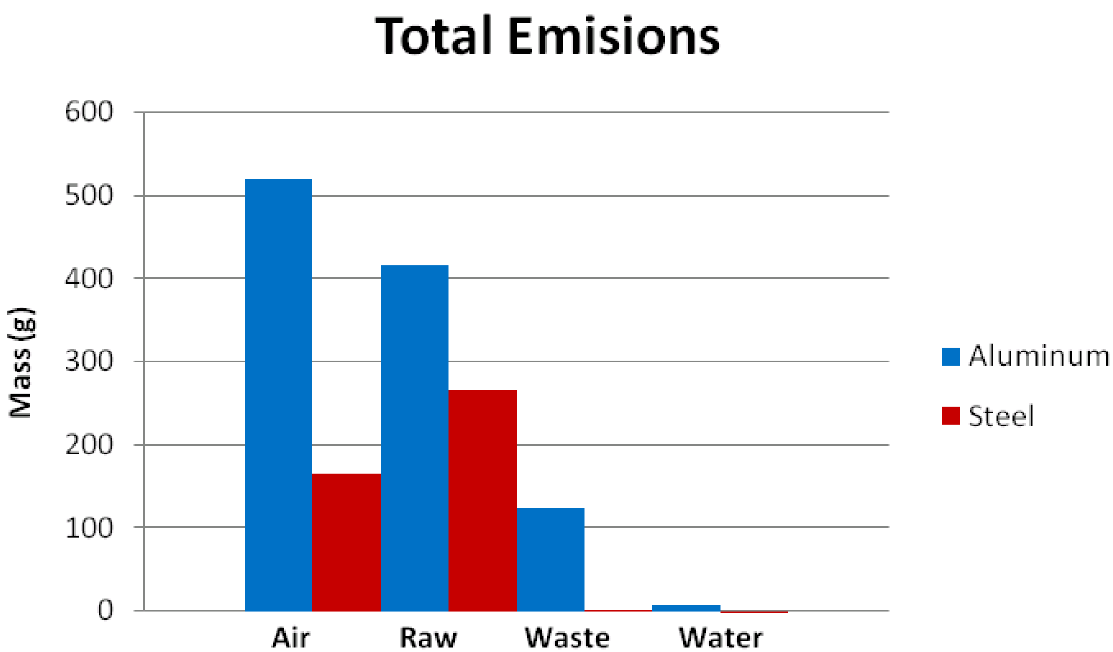


Figure C.4: Normalized Score in Human Health, Eco-Toxicity, and Resource Categories

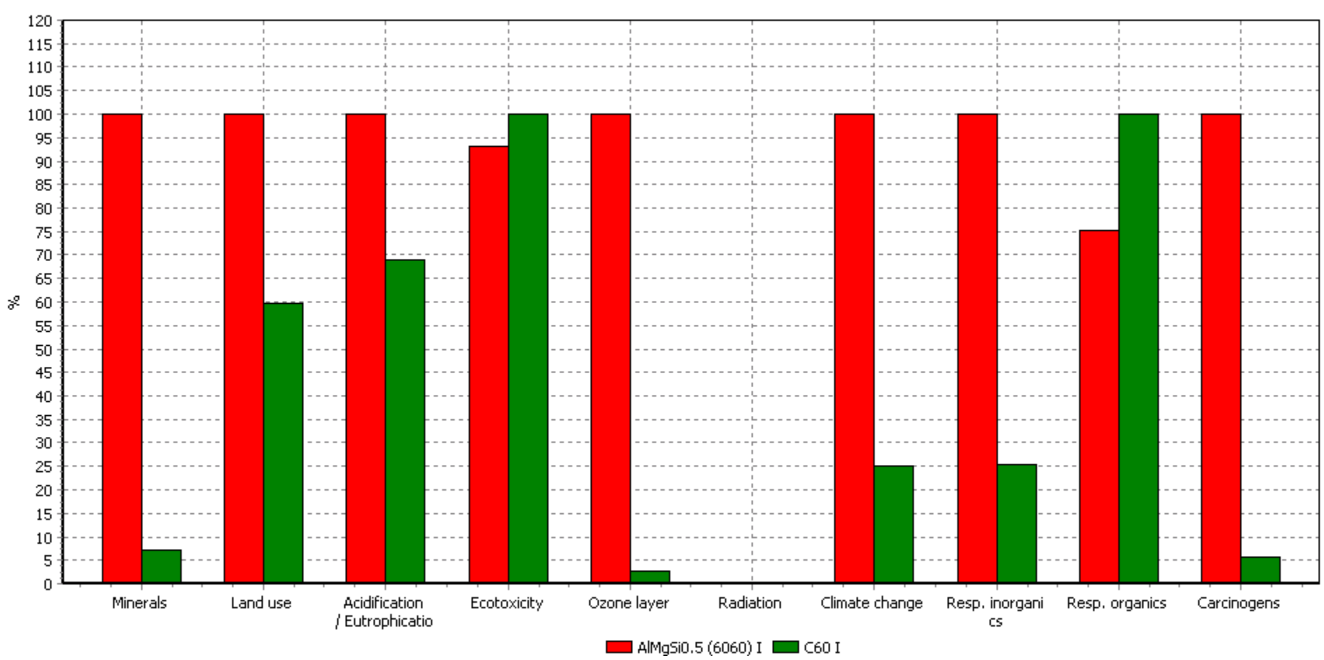
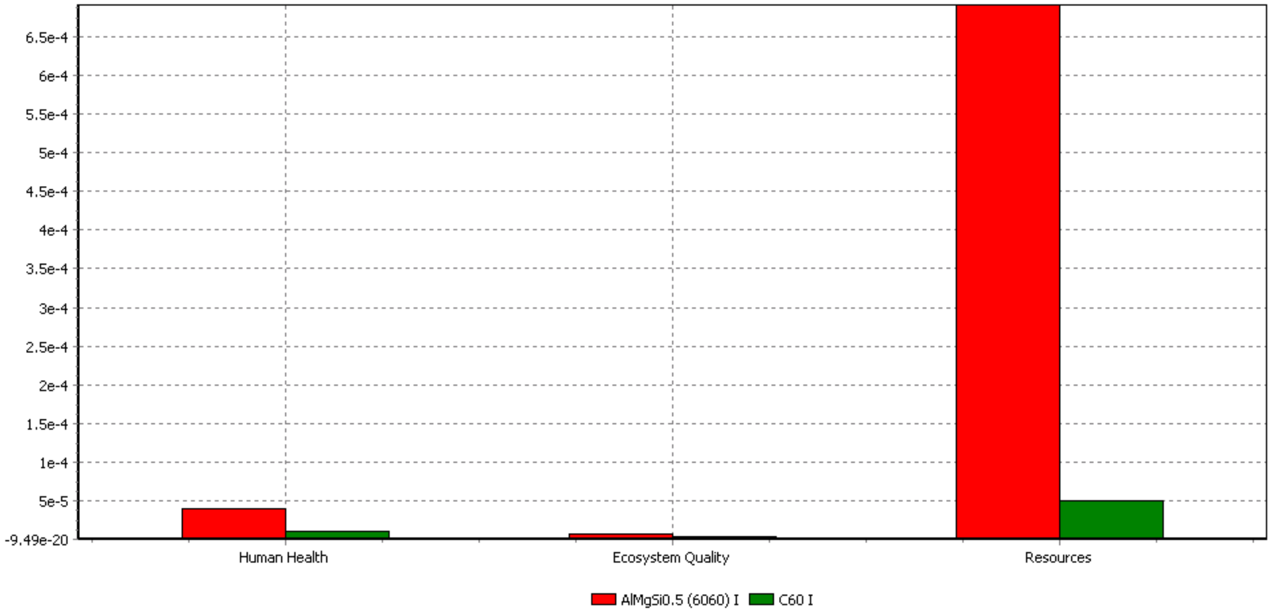
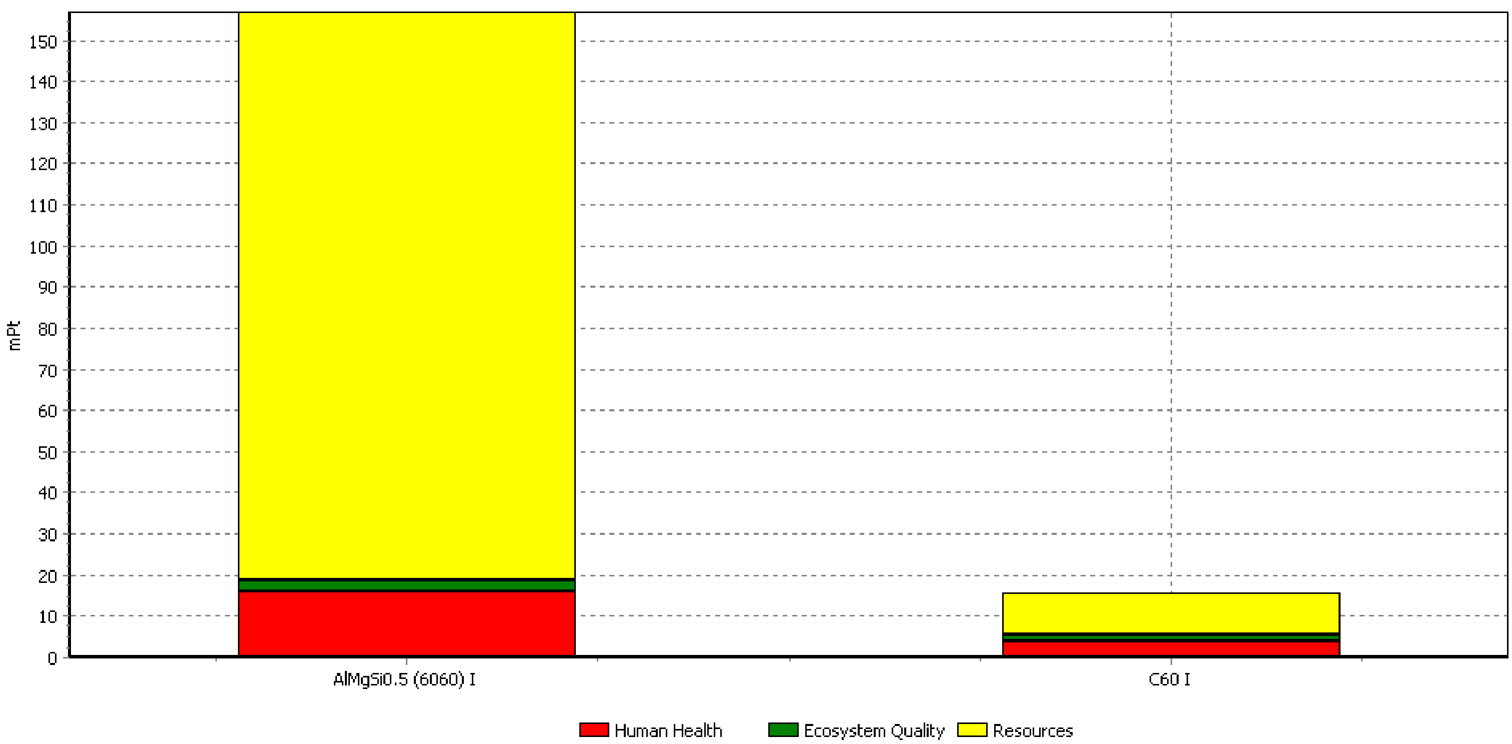


Figure C.5 and C.6: Single Score Comparison in "Points"





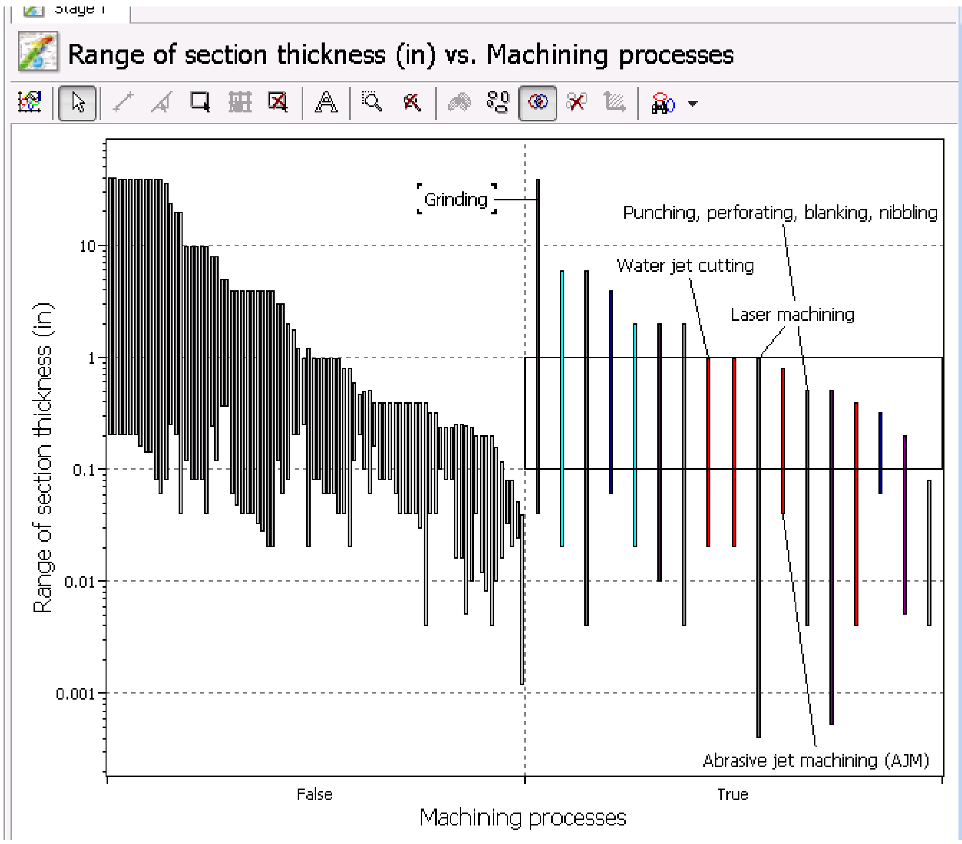
Manufacturing Process Selection

1. Due to the fact that the intended use of this parallel mechanism is for research purposes, it will not go into mass production. However, it is very possible that other top mechanical engineering and manufacturing research facilities around the world would be interested in testing and working with a device like this. For this reason, one could assume that anywhere between 10-15 universities would like to receive either 1 or 2 parallel mechanisms with large joint clearances for their own research. With this in mind, a safe estimate would produce 20 total mechanisms.

20*3 = 60 Total Links
20*9 = 180 Journals

2-A. Link Selection

The most important attributes for the link are the cross section size ¼” x ¼” and thru hole size of 1/8” diameter. The process must also be able to create a shape that is at least 2.75” long. The process must be able to achieve high accuracy and precision, as precise geometry is one of the key elements in the mechanism’s success. Using section thickness and machining processes as the CES search elements, we were able to narrow down our choices with respect to the material thickness of ¼”. Even though there are multiple adequate processes, water jet cutting has a short set up time, is low cost to operate and can create many parts at once. It can also perform cutting and drilling operations in one run. This is important for achieving high accuracy components and we are still maintaining a low cost.



2-B. Journal Selection

The most important attribute for the journals is that they are circular prismatic. For this reason, we used this as one of the two inputs in CES selector. The other comparable attribute included on the x axis was cutting processes available. CES returned 8 possibilities for manufacturing the geometry.

1. Abrasive Jet Machining – can only create simple geometry by cutting shapes, leaves taper on part edges
2. Broaching- high economic batch size (minimum of 1,000 and high tooling cost)
3. Chemical machining – High environmental impact, scrap cannot be recycled, and disposal of chemicals is costly and can be dangerous and hazardous

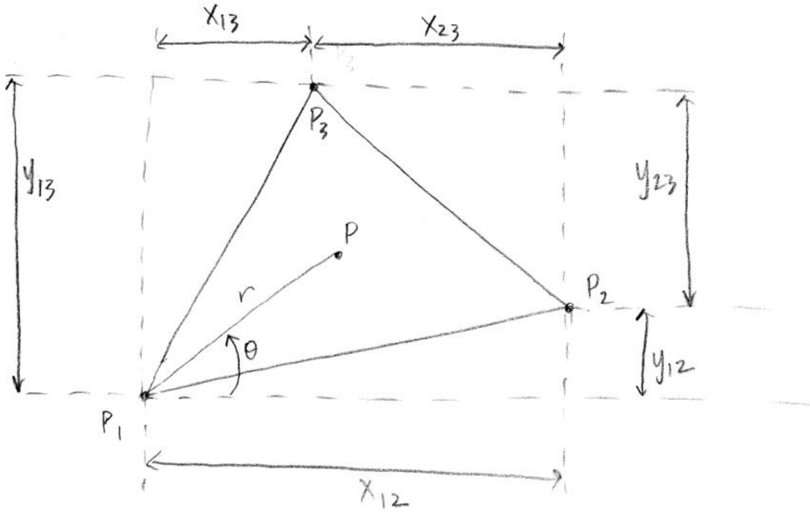
4. Electrical discharge wire cutting – This would an adequate choice because of correct range for batch size, good tolerances, and low tooling cost. However, high equipment cost (not available) and high labor intensity eliminate this possibility
5. Hot wire cutting – Not used for aluminum or steel, only foam or thin plastics
6. Plasma arc cutting – Used to create simple geometrical shapes from sheets, and has a high equipment cost – not available to us.
7. Turning, boring, parting – Good for batches between 1 and 10,000,000 and can be automated or done manually with precision. Secondary "in set-up" processes like threading can also be performed which is optimal for the manufacturing requirement of the journals. Ideally this will be performed on a CNC lathe which will be automated and provide extreme accuracy and precision.
8. Ultrasonic machining – can create complex parts and is a versatile tool. However, some tapering occurs as distance from discharge nozzle increases. Journals are required to be precisely toleranced.

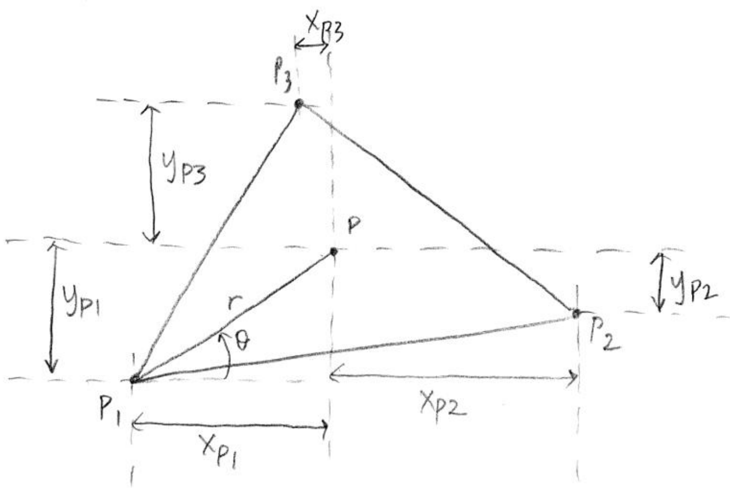
APPENDIX D: FUNCTIONAL DESIGN SOLUTIONS

	Feasibility	Technology	Go/nogo
Motor interp			
Labview	yes	yes	yes
Platform Movement			
Linear Motor	yes	yes	yes
Linkages			
Cylindrical	yes	yes	yes
Rectangular	yes	yes	yes
Triangular	no	-	-
Force measure			
Force meter	no	-	-
Electrical strain gauge	yes	yes	yes
Load Cell	yes	yes	yes
Pressure Transducer	no	-	-
Optical Strain Gauge	yes	yes	yes
Joints			
spherical	no	-	-
revolute	yes	yes	yes
prismatic	yes	yes	no
cylindrical	no	-	-
helical	no	-	-
plane pair	no	-	-
gear pair	no	-	-
cam pair	no	-	-
Position measure			
accelerometer	yes	yes	no
fiber optics	yes	yes	yes
lasers	no	-	-
camera'	yes	yes	yes
physical measurement	yes	yes	yes
pencil tracer	yes	yes	yes
Applied force			
human applied load	yes	yes	no
suspended weight	yes	yes	yes
spring load	no	-	-
Platform/links material selection			
steel	yes	yes	yes
aluminum	yes	yes	yes
plexiglass	yes	yes	yes
wood	yes	yes	yes
posterboard	no	-	-
corkboard	no	-	-
carbon fiber	yes	yes	yes

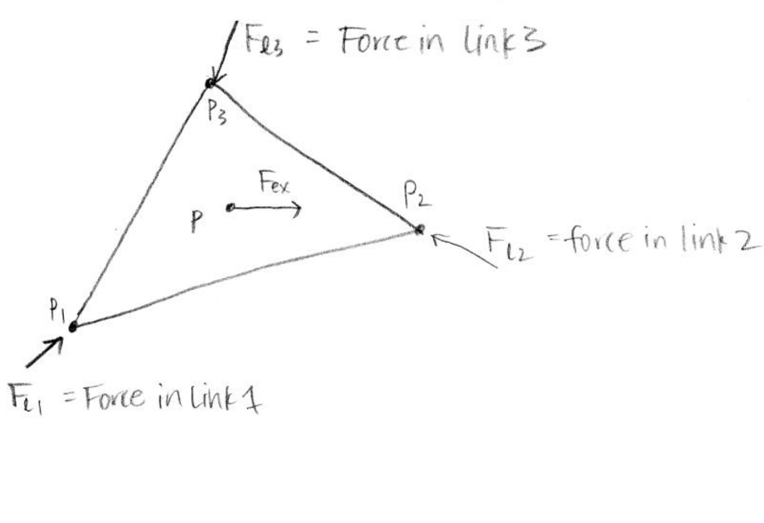
APPENDIX E: STATIC CALCULATIONS

PARALLEL MECHANISM GEOMETRY (NO JOINT CLEARANCES)

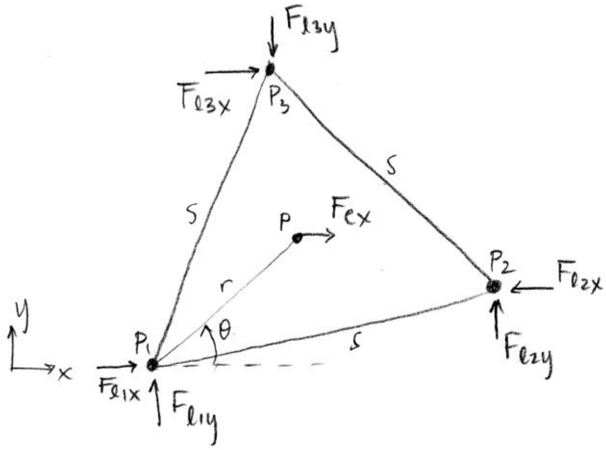




PARALLEL MECHANISM FORCES



PARALLEL MECHANISM FREE BODY DIAGRAM (FOR LINK FORCE DETERMINATION)



FORCES

$$\Sigma F_x = F_{ex} + F_{l1x} - F_{l2x} + F_{l3x} = 0$$

$$\Sigma F_y = F_{l1y} + F_{l2y} - F_{l3y} = 0$$

MOMENTS

$$P_1: \ \circlearrowleft+ \Sigma M_1 = -F_{ex} y_{P1} + F_{l2x} y_{12} - F_{l3x} y_{13} + F_{l2y} x_{12} - F_{l3y} x_{13} = 0$$

$$P_2: \ \circlearrowleft+ \Sigma M_2 = -F_{ex} y_{P2} + F_{l1x} y_{12} - F_{l3x} y_{23} - F_{l1y} x_{12} + F_{l3y} x_{23} = 0$$

$$P_3: \ \circlearrowleft+ \Sigma M_3 = F_{ex} y_{P3} - F_{l2x} y_{23} + F_{l1x} y_{13} - F_{l1y} x_{13} + F_{l2y} x_{23} = 0$$

$$P: \ \circlearrowleft+ \Sigma M_P = -F_{l1y} x_{P1} + F_{l1x} y_{P1} - F_{l3x} y_{P3} + F_{l3y} x_{P3} + F_{l2y} x_{P2} - F_{l2x} y_{P2} = 0$$

GEOMETRY

$x_{P1} = r\cos\theta \quad x_{P2} = r\cos(60-\theta) \quad x_{P3} = r\cos(120-\theta)$

$y_{P1} = r\sin\theta \quad y_{P2} = r\sin(60-\theta) \quad y_{P3} = r\sin(120-\theta)$

$x_{12} = s\cos(\theta-30) \quad x_{13} = s\sin(150-\theta) \quad x_{23} = s\sin\theta$

$y_{12} = s\sin(\theta-30) \quad y_{13} = s\cos(150-\theta) \quad y_{23} = s\cos\theta$

PARAMETERS

$r = 2$ in.

$s = 4\cos 30 = 3.46$ in

$F_{ex} \sim (10 \text{ to } 20 \text{ kg})(9.8 \text{ m/s}^2)$

APPENDIX F: ADDITIONAL ANTI TILT CONCEPTS

The following Anti Tilt concepts are the original ideas that eventually led to the Sandwich concept found on page 14.

Figure B1: Double link sandwich concept supports platform from above and below

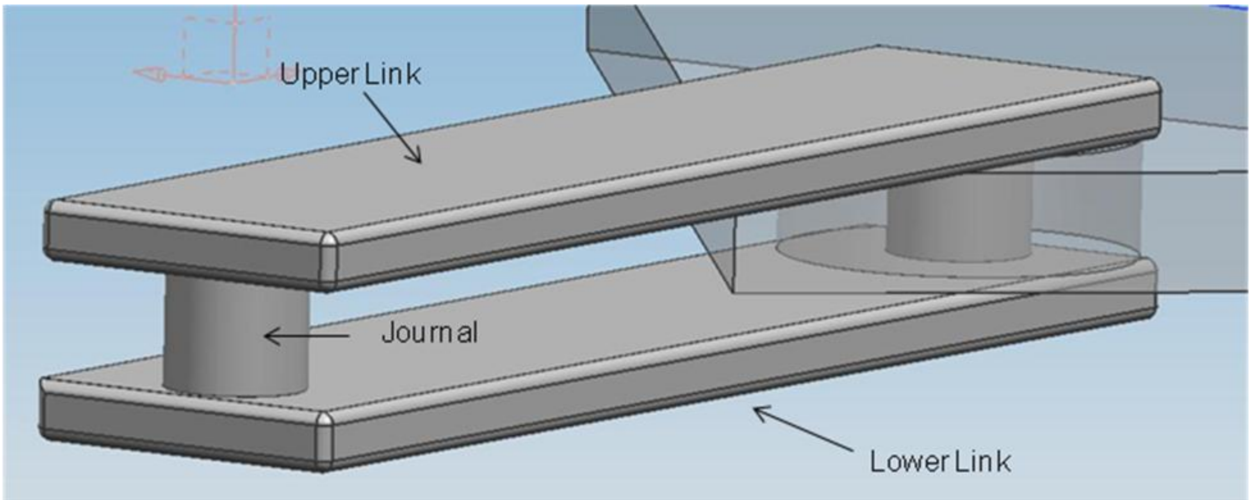
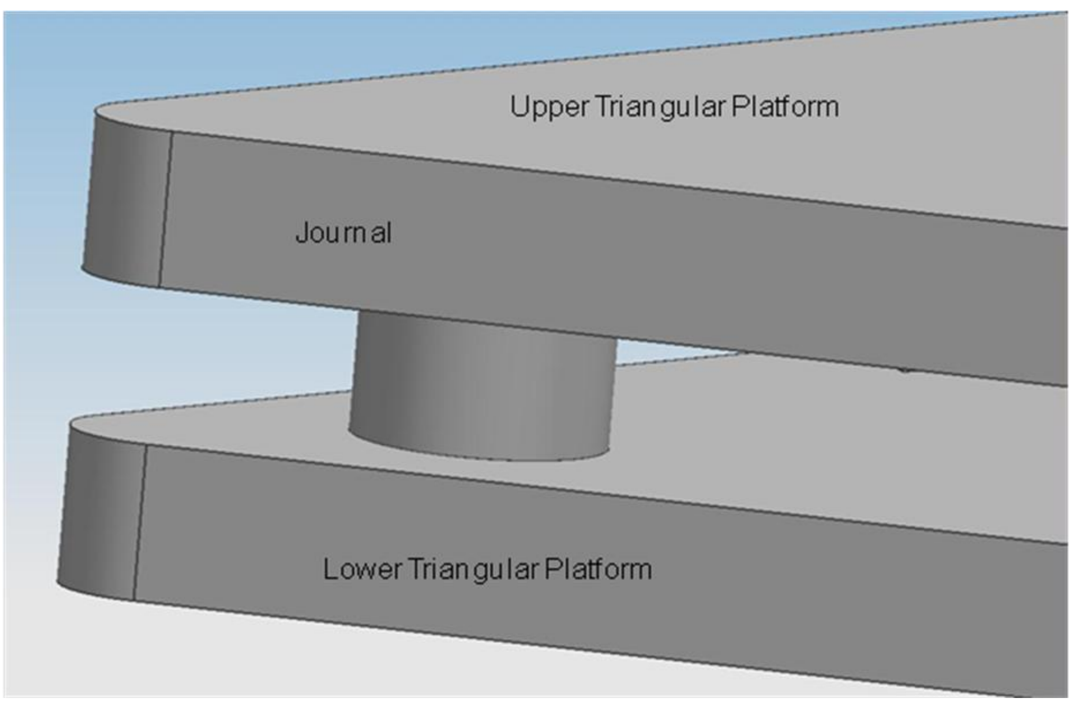
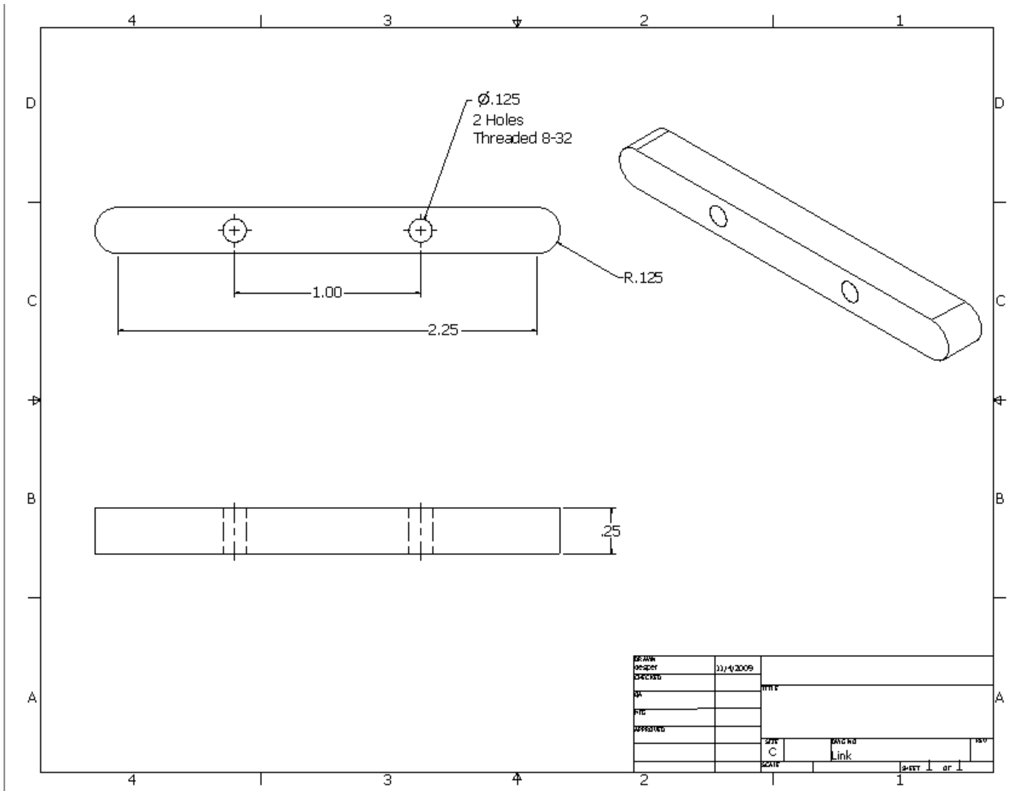
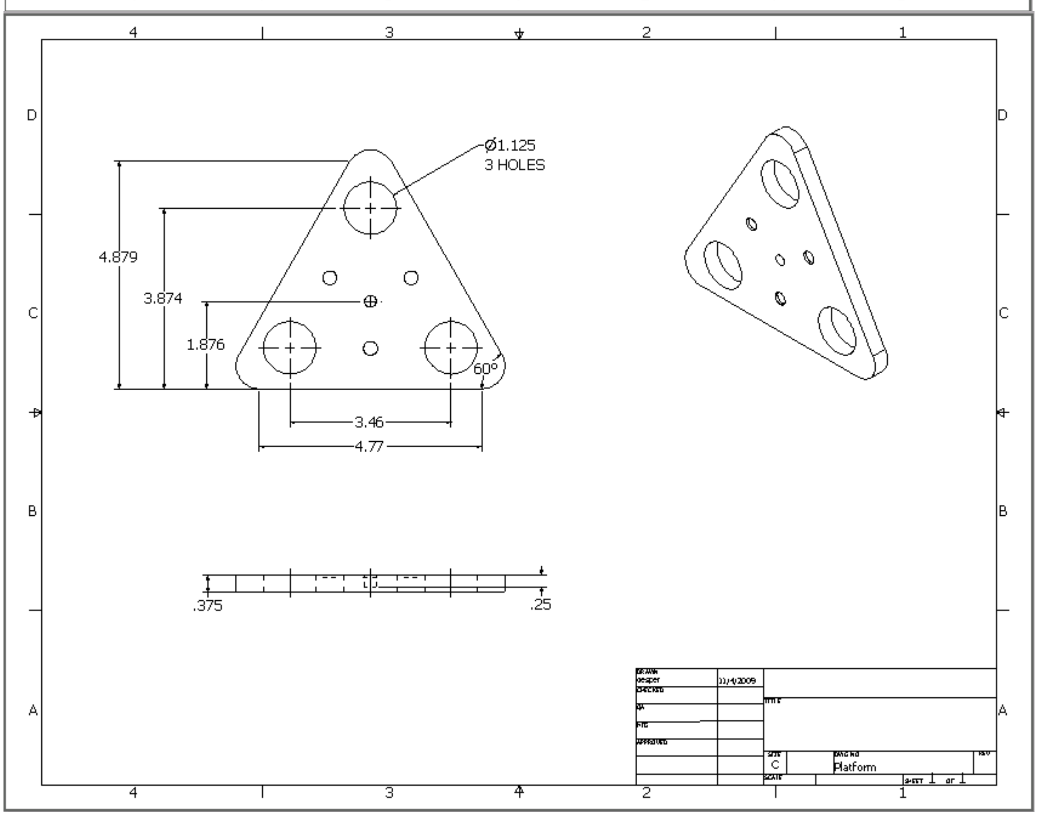


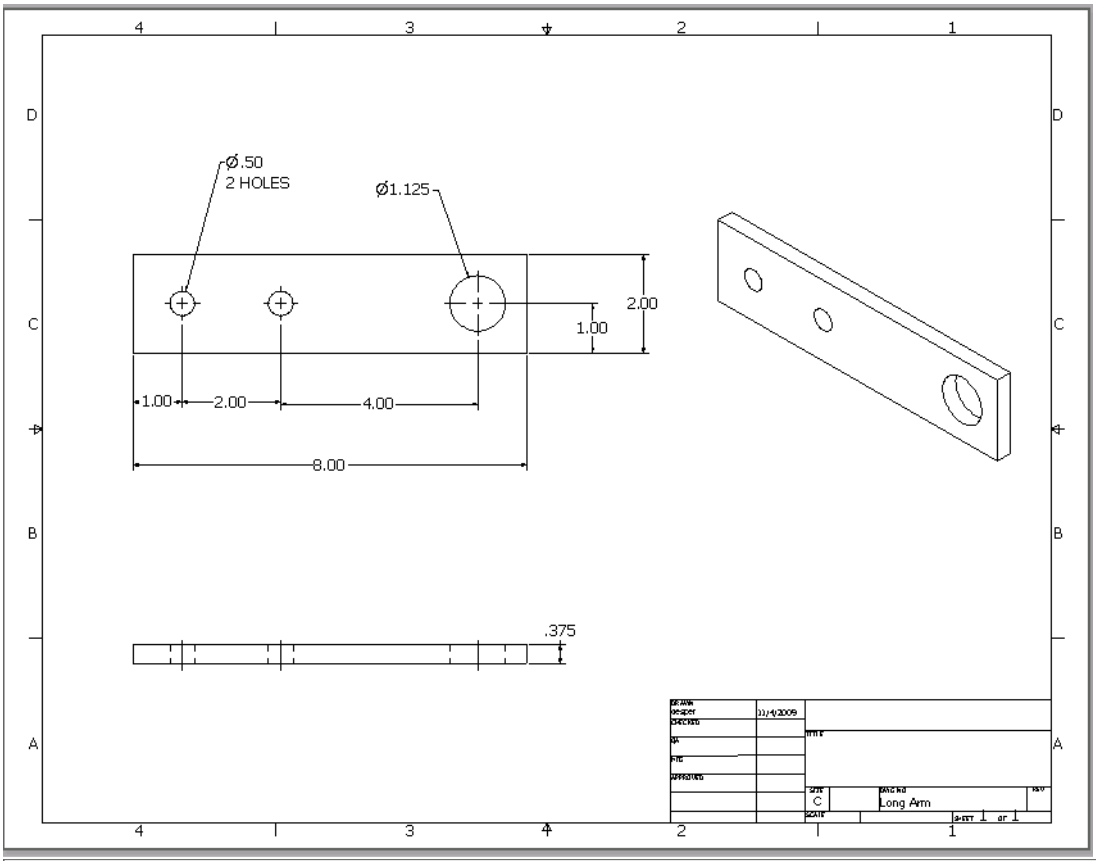
Figure B2: Double Platform sandwich concept constrains platform orientation to that of the link it is connected to

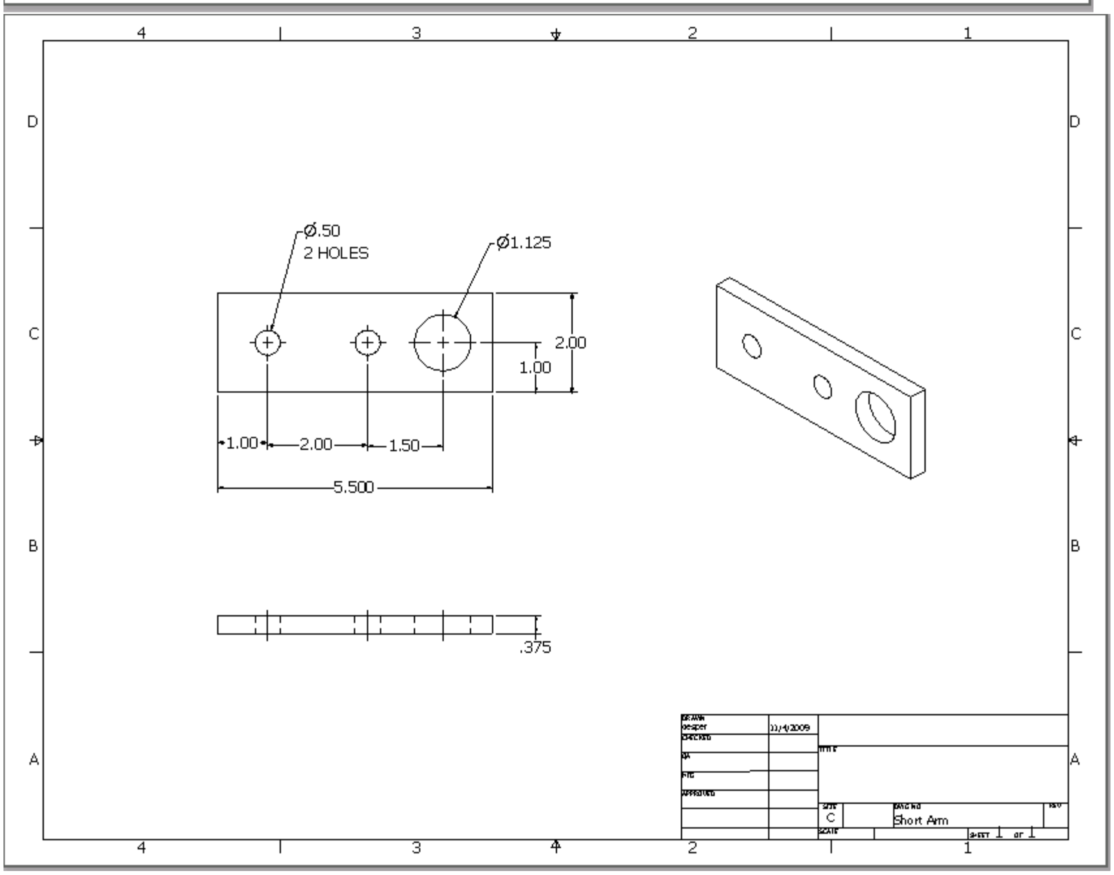


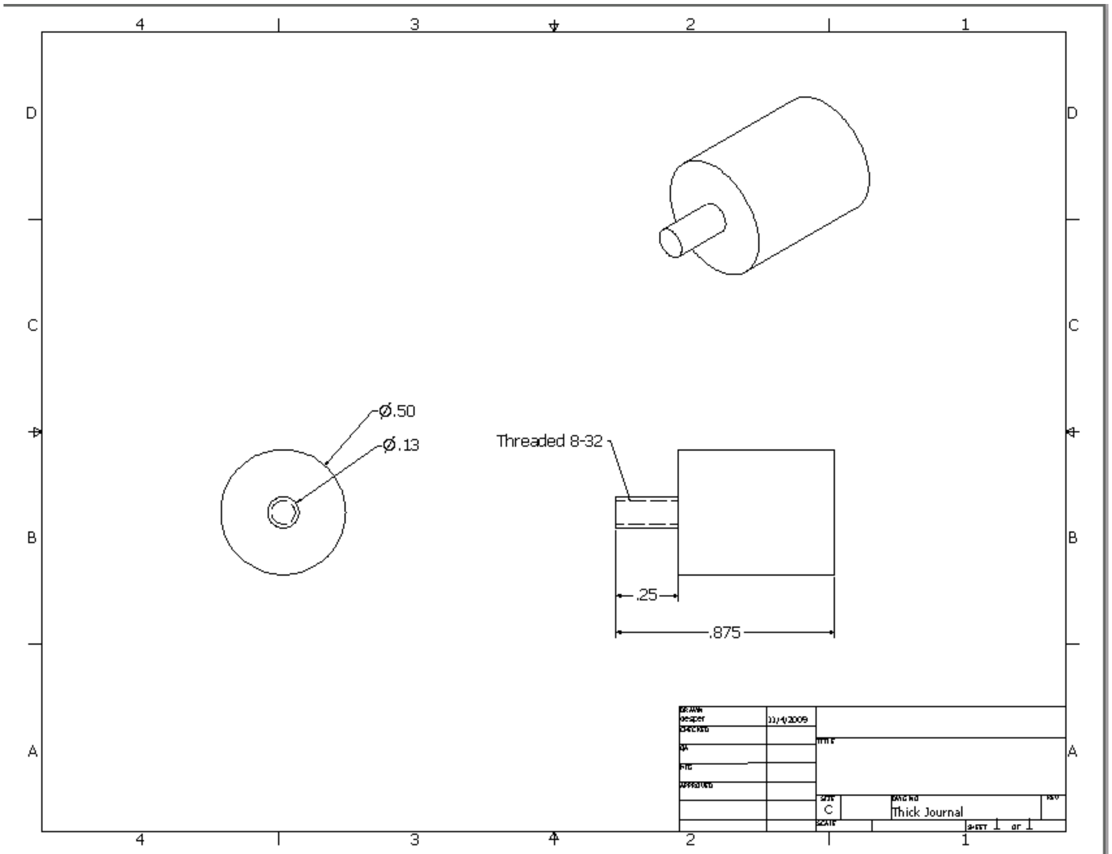
APPENDIX G: CAD DRAWINGS

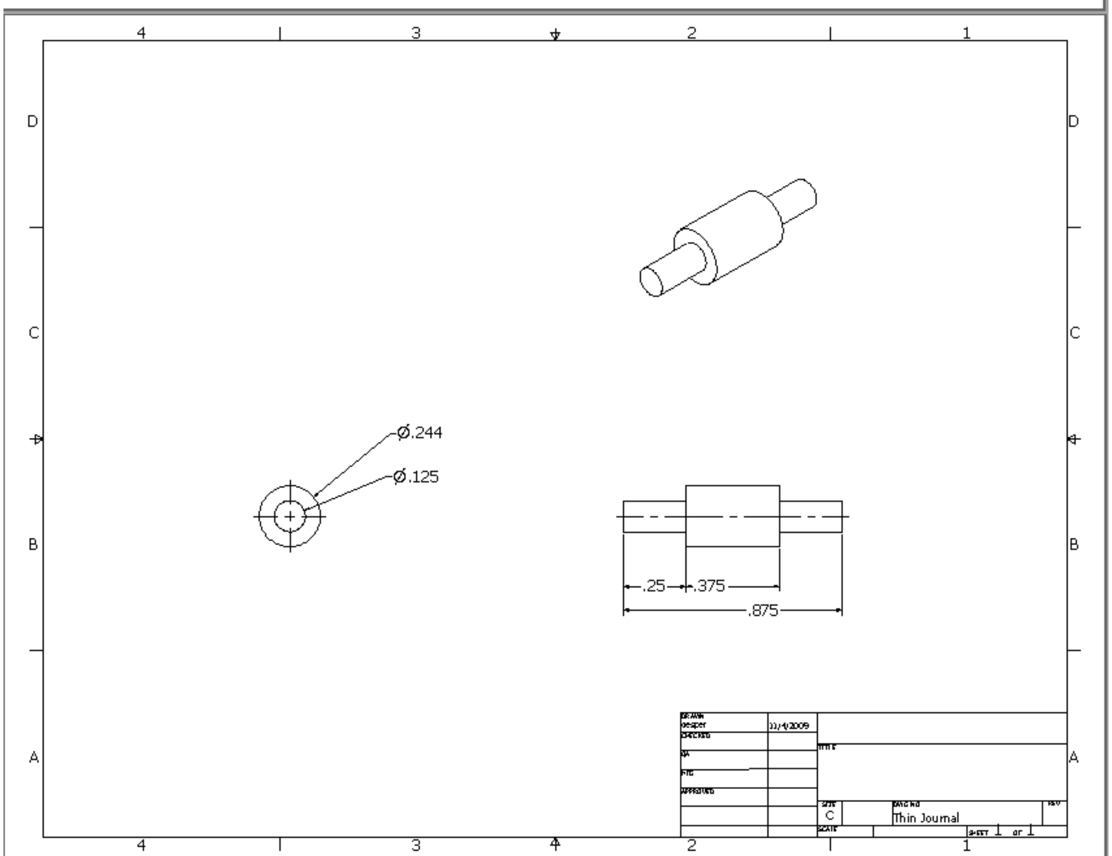


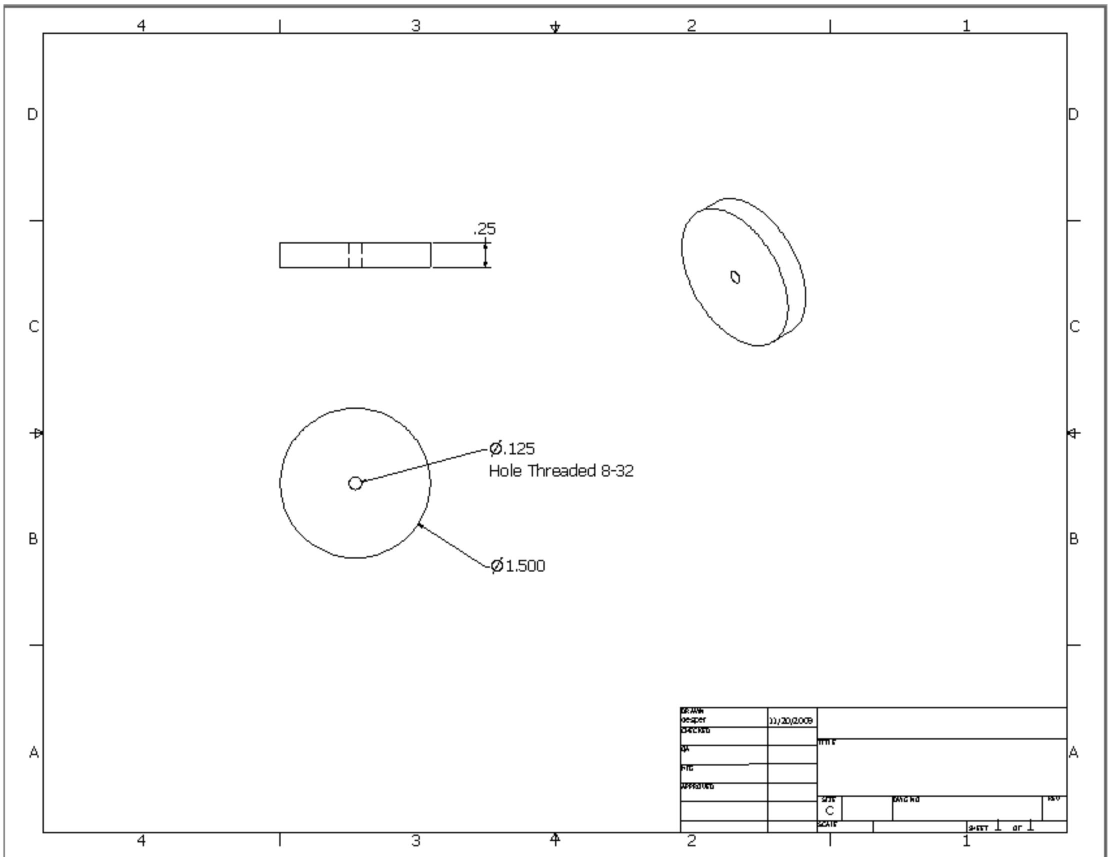


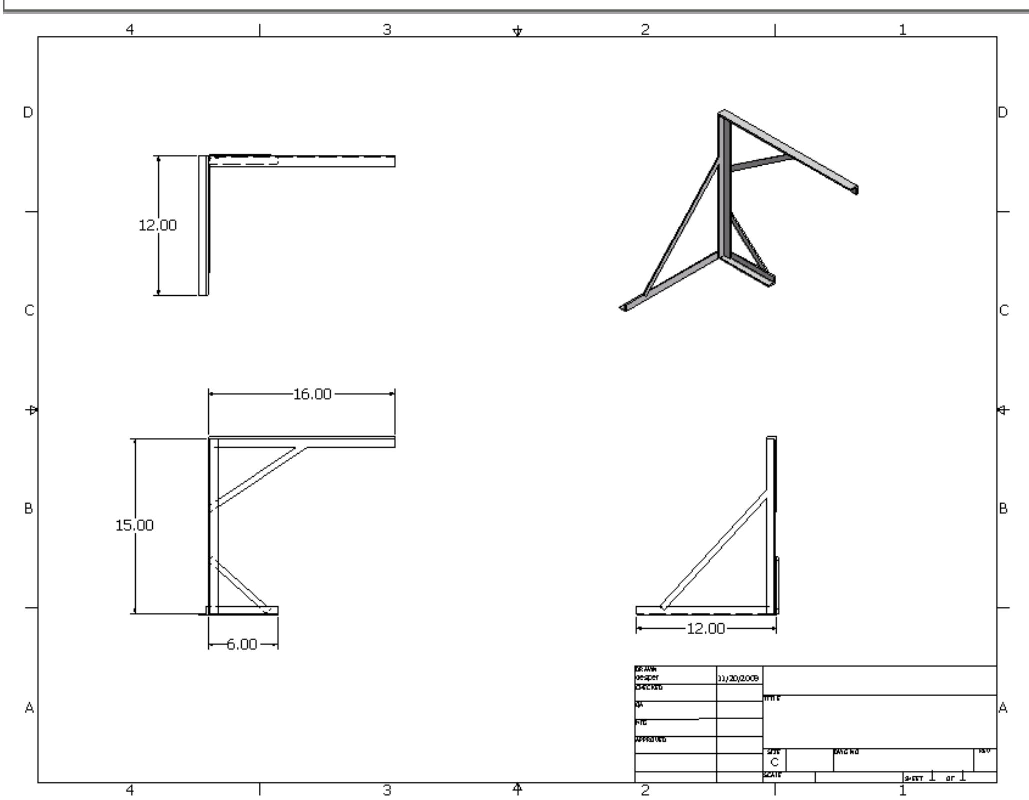

Ø.50
2 HOLES
Ø1.125
2.00
1.00
1.00
2.00
4.00
8.00
.375
11/4/2009
C
Long Arm

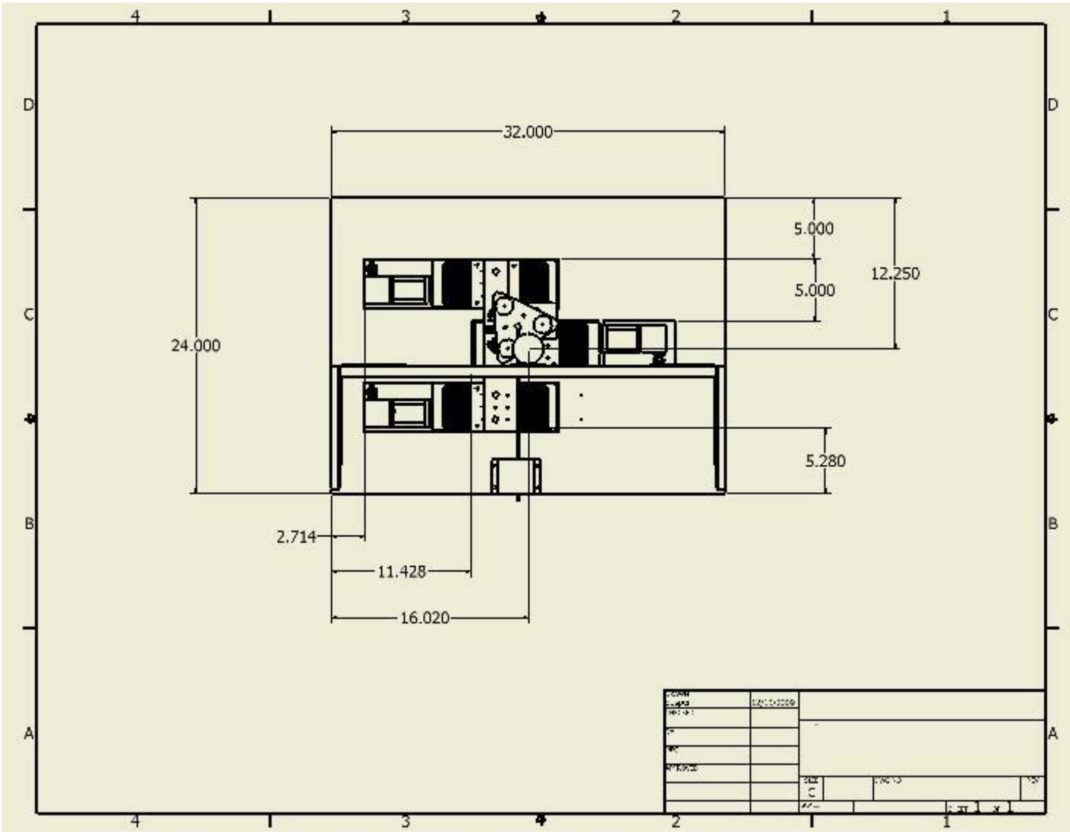

Ø.50
2 HOLES
Ø1.125
2.00
1.00
1.00
2.00
1.50
5.500
.375
11/4/2009
C
Short Arm

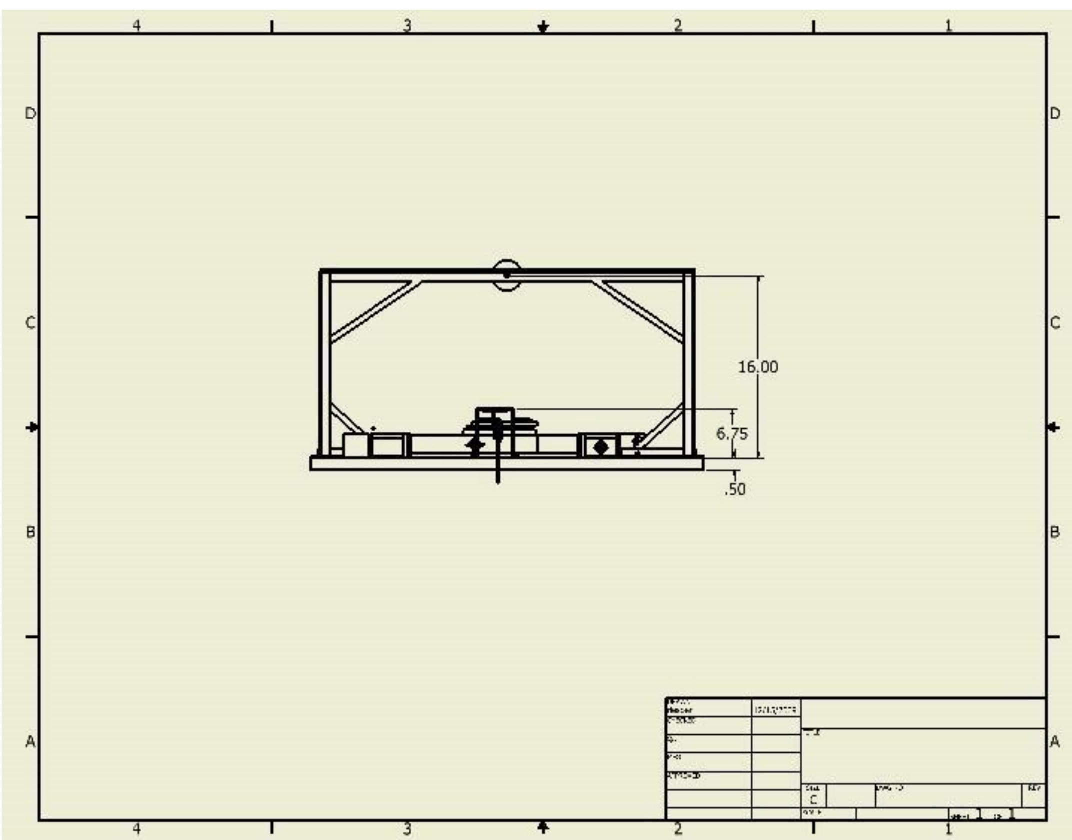

Ø.50
Ø.13
Threaded 8-32
.25
.875
Thick Journal
C


Ø.244
Ø.125
.25
.375
.875
Thin Journal
C


.25
Ø.125
Hole Threaded 8-32
Ø1.500


12.00
16.00
15.00
6.00
12.00


32.000
5.000
5.000
12.250
24.000
5.280
2.714
11.428
16.020


16.00
6.75
.50

APPENDIX H: CAMERA AND POSITIONING PROGRAM

A Logitech V-UW21 web camera and corresponding Logitech QuickCam software was used to obtain photographs for position measurement. The following list depicts the steps necessary to take a photograph:

1. Make sure the web camera is connected to the computer containing the Logitech QuickCam software via its USB connection
2. Select "Quick Capture"
3. Select "Take a Picture"

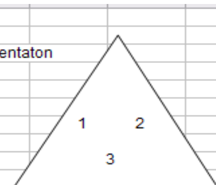
The images are stored and automatically numbered in a folder titled "My Logitech Pictures". To ensure that the photograph numbering begins with Picture 1, make sure this folder is empty prior to conducting a test. At the conclusion of a test, move the test photos to a new folder so that the web camera is ready to start taking pictures from Picture 1 again.

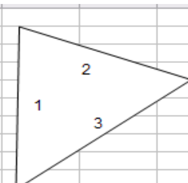
In order to analyze the images from each test to determine platform position, a Matlab program was used to identify the three marker's (X, Y) pixel locations. The Matlab program identifies the colored markers, their corresponding centers, and outputs the pixel X and Y locations of the marker's centers.

In order to convert the marker identification pixel position outputs into final X, Y, Θ positions with respect to the origin, a Microsoft Excel worksheet was used as seen in Figure XXX. First, the origin was determined with respect to the pixel origin referred to by the Matlab marker identification program. This coordinate set was added or subtracted to each data point in order to shift the outputted data to the correct reference frame.

Figure H.1: Excel worksheet used to calculate platform position and orientation

	X	Y	
Origin	310.23	93.92	<- May need to be adjusted
Pixels to " conversions	66.55	64.8	To negative values for orientaton





Make sure that markers are detected as shown in example triangle to the left these correspond P1,P2 and P3 (x,y) in the data table below

	Absolute Pixels						Adjustment (from origin) to inches						Midpoint		Platform Position		
	1		2		3		1		2		3						
Trial No	P1x	P1y	P2x	P2y	P3x	P3y	P1x	P1y	P2x	P2y	P3x	P3y	Ax	Ay	X	Y	Theta(DEG)
1	373	144.9	255.1	158.6	301.5	48.1	-0.943	0.787	0.828	0.998	0.131	-0.707	-0.057	0.892	0.037	0.093	6.939
2	374.1	147.3	256.9	163.1	301.1	51.9	-0.960	0.824	0.801	1.068	0.137	-0.648	-0.079	0.946	0.029	0.149	8.009
3	375.3	149.8	258	168.4	301.5	55.7	-0.978	0.862	0.785	1.149	0.131	-0.590	-0.096	1.006	0.017	0.208	9.440
4	375.2	151.5	258.8	172.6	298.6	58.9	-0.976	0.889	0.773	1.214	0.175	-0.540	-0.102	1.051	0.037	0.255	10.723
5	377.5	155.1	260.6	177.2	299.7	63.8	-1.011	0.944	0.746	1.285	0.158	-0.465	-0.133	1.115	0.013	0.325	11.238
6	378.2	159.6	262.8	182.1	300.4	69.1	-1.021	1.014	0.713	1.361	0.148	-0.383	-0.154	1.187	-0.003	0.402	11.444
7	380	163.5	264.3	187.3	299.7	74	-1.048	1.074	0.690	1.441	0.158	-0.307	-0.179	1.257	-0.010	0.475	12.115
8	379.7	165.9	265	192.2	299.7	74.8	-1.044	1.111	0.680	1.517	0.158	-0.295	-0.182	1.314	-0.012	0.509	13.410
9	379.3	169.1	265.7	196.1	298.6	82.4	-1.038	1.160	0.669	1.577	0.175	-0.178	-0.184	1.369	-0.005	0.595	13.774
10	380.4	177.1	265.8	204.2	299.7	90.5	-1.054	1.284	0.668	1.702	0.158	-0.053	-0.193	1.493	-0.018	0.720	13.826
11	371.9	211.5	254.7	227	299.7	116.8	-0.927	1.815	0.834	2.054	0.158	0.353	-0.046	1.934	0.056	1.144	7.856
12	360.9	224.2	243.3	231.2	295.6	124.9	-0.761	2.010	1.006	2.119	0.220	0.478	0.122	2.065	0.171	1.271	3.539
13	356.1	228.4	238.5	230.9	295.2	126.3	-0.689	2.075	1.078	2.114	0.226	0.500	0.194	2.095	0.210	1.297	1.263
14	354.6	229.8	236.6	229.5	296	126.7	-0.667	2.097	1.106	2.092	0.214	0.506	0.220	2.095	0.217	1.300	-0.152
15	352.8	231.9	235.2	229.4	296.3	127.7	-0.640	2.129	1.127	2.091	0.209	0.521	0.244	2.110	0.227	1.316	-1.263
16	352	232.9	234	227.6	296.7	127.6	-0.628	2.145	1.145	2.063	0.203	0.520	0.259	2.104	0.231	1.312	-2.679
17	350.2	233.9	232.6	227.7	297.8	127.3	-0.601	2.160	1.166	2.065	0.187	0.515	0.283	2.112	0.235	1.314	-3.134
18	349.8	235.8	231.5	227	298.2	129.1	-0.595	2.190	1.183	2.054	0.181	0.543	0.294	2.122	0.237	1.332	-4.451
19	352.4	235.1	234.4	225.2	301.5	127	-0.634	2.179	1.139	2.026	0.131	0.510	0.253	2.102	0.192	1.306	-5.008
20	354.6	234.4	236.3	222.8	304.8	125.9	-0.667	2.168	1.111	1.989	0.082	0.494	0.222	2.078	0.152	1.286	-5.871
21	356.4	233.6	239.2	221.7	308.1	124.2	-0.694	2.156	1.067	1.972	0.032	0.467	0.187	2.064	0.109	1.266	-6.024
22							4.662	-1.449	4.662	-1.449	4.662	-1.449	4.662	-1.449	4.662	-1.449	0.000
23													0.000	0.000	0.000	0.000	0.000

Next, using the X and Y reference images, the data points were converted from pixels to inches by determining the ratio of pixels per inch in the x and y direction. The reference images are shown in Figure H.2 and Figure H.3.

Figure H.2: Horizontal Position Reference Image

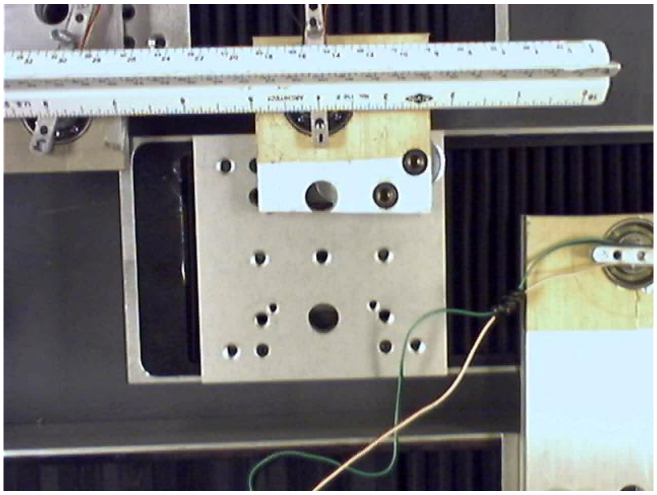
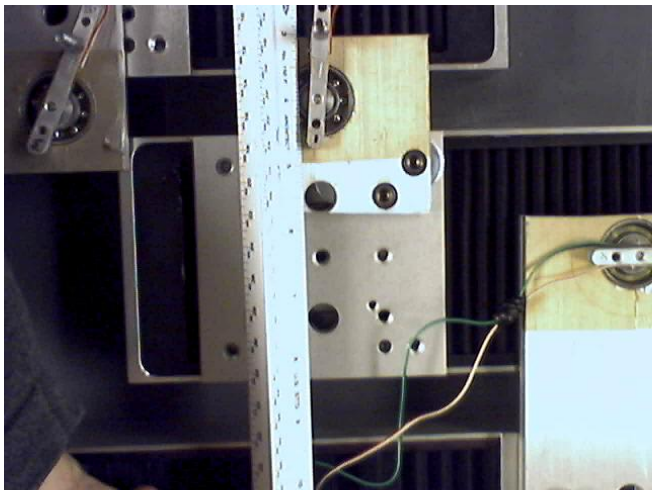


Figure H.3: Vertical Position Reference Image



The center of the platform was determined by using a set of midpoint calculations based on the geometry. The midpoint of P1 and P2 was found to be A (Ax, Ay). Then the center of the platform was determined as the midpoint between A and P3.

In order to determine the angle of rotation, Θ, from the x axis Equation XXX was used where P1y and P2y are the adjustment (from origin) to inches values from the excel worksheet.

Equation H.1: Platform Orientation

$$\theta = \frac{180}{\pi} \sin^{-1} \frac{(P2y - P1y)}{1.75}$$

The excel worksheet for calculating platform position, Matlab marker identifier program, and Logitech QuickCam software can be found on the backup software CD.

APPENDIX I: MOTOR OPERATION

For correct motor operation, the motors are defined as follows:

- Motor 1 = X
- Motor 2 = Y
- Motor 3 = Z

In order to move each motor to the x = 0 location the following steps must be taken (please note that motors 1 and 2 cannot occupy this position at the same time):

1. Home Y
2. Home X and then move 101mm in the X+ direction
3. Home Z and then move 125 mm in the Z+ direction

A list of initial motor positioning steps was developed for all 8 tests. The primary 8 tests are defined in Table I.1. The following sections outline these steps so that the mechanism can be assembled with the correct motor locations and assembly modes. Please note that when using the avi files, the green link donates a long link and the black link denotes a short link for cases with joint clearances. Additionally, the prepared batch files call upon the function Scan.exe.

Table I.1: Primary 8 Tests

Test Number	Platform Type	External Force Direction	Path
1	No Clearance	-y	Path1_no_clearance.bat
2	No Clearance	-y	Path3_no_clearance.bat
3	With Clearance	-y	Path1_clearance_minus.bat
4	With Clearance	-y	Path3_clearance.bat
5	No Clearance	+y	Path1_no_clearance.bat
6	No Clearance	+y	Path3_no_clearance.bat
7	With Clearance	+y	Path1_clearance_plus.bat
8	With Clearance	+y	Path3_clearance.bat

For Paths 1 and 5:

1. Home X
2. Home Y
3. Home Z
4. Move Y+ -24.73
5. Move X+ 32.22
6. Move Z+ 145.83
7. Look at AVI file for link/angle positions: Path1_no_clearance.avi
8. Assemble platform
9. Run corresponding motor program: Path1_no_clearance.bat
10. Read strain gauge values and take photograph at each position

For Path 3:

1. Home X
2. Home Y
3. Home Z
4. Move Y+ -20.25
5. Move X+ 35.83

6. Move Z+ 142.63
7. Look at AVI file for link/angle positions: Path1_clearance_minus.avi
8. Assemble platform
9. Run corresponding motor program: Path1_clearance_minus.bat
10. Read strain gauge values and take photograph at each position

For Path 7:

1. Home X
2. Home Y
3. Home Z
4. Move Y+ -30.05
5. Move X+ 28.72
6. Move Z+ 149.02
7. Look at AVI file for link/angle positions: Path1_clearance_plus.avi
8. Assemble platform
9. Run corresponding motor program: Path1_clearance_plus.bat
10. Read strain gauge values and take photograph at each position

For Paths 2 and 6:

1. Home X
2. Home Y
3. Home Z
4. Move Y+ -64.56
5. Move X+ 71.58
6. Move Z+ 106.16
7. Look at AVI file for link/angle positions: Path3_no_clearance.avi
8. Assemble platform
9. Run corresponding motor program: Path3_no_clearance.bat
10. Read strain gauge values and take photograph at each position

For Paths 4 and 8:

1. Home X
2. Home Y
3. Home Z
4. Move Y+ -64.56
5. Move X+ 71.58
6. Move Z+ 106.16
7. Look at AVI file for link/angle positions: Path3_clearance.avi
8. Assemble platform
9. Run corresponding motor program: Path3_clearance.bat
10. Read strain gauge values and take photograph at each position