LUNAR HARVESTER PROTOTYPE

NASA - Corporation 4

"OPERATIONAL READINESS REVIEW"

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1.0 ABSTRACT

The purpose of this senior design project is to develop an excavator for use on a planned lunar base. The harvester will collect regolith for processing into oxygen for use by the lunar settlement. This is a vital component of NASA's eventual plans for the lunar colony, as the cost of transporting enough supplies for the colony would be prohibitively expensive.

Therefore, an in-situ resource collector is a necessity of the colony. This collector has a number of system requirements including:

1. Shall be designed to conduct studies on earth but be able to operate in a Lunar environment
2. Shall interface with Gator utility vehicle
3. Shall be operated remotely
4. Shall collect and hold at least 50 kg soil per hour

These and other requirements will be discussed throughout this report.

The final design is an aluminum frame constructed of 80/20 that supports a linkage subassembly. These linkages are then used to raise and lower a bucket that acts as the digging implement. This bucket has three positions digging, transport, and dumping. An actuator manipulates the linkages and bucket into the required positions. The actuator itself is controlled by the control system designed by the electrical engineering team.

After the Preliminary Design Review (PDR) and Critical Design Review (CDR) were completed, work was begun on the steps necessary for an Operational Readiness Review (ORR). The purpose of the ORR is to document the testing stage of the design process and provide results that the design chosen will meet the project requirements. The cost of correcting any design flaw will be magnified greatly in the post fabrication phase so it was vital to catch all design errors before fabrication begins. All constraints not already specified in the CDR such as bearing and actuator sizes have been selected. The complete set of correctly dimensioned engineering drawings was used to fabricate and assemble a finished prototype. Extensive testing was conducted to test out all the different operations of the Lunar Harvester Prototype. The Operational Readiness Review (ORR) will document the finished prototype complete with validation of requirements and documentation and results of prototype testing.

The bearing selected for use throughout the design was a Dry slide self-lubricating bearing with PTFE coating produced by Daemar Bearings Incorporated. A plunger-type linear actuator was selected instead of a slider linear actuator due to its better cost versus performance ratio. There were no mechanical problems during testing, and the harvester prototype fulfilled the mission requirements exactly as designed. The prototype interfaced with a Gator Utility vehicle, was controlled from a wireless ground station, and scraped up 50 – 80 lbs per test run of pseudo-regolith and dumped it at a designated location in under two minutes.

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4.0 INTRODUCTION

In looking to establish a base on the moon, there is considerable research and development being aimed at building and sustaining such a base. One of the immediate needs that arise is the need for oxygen. Constantly shipping oxygen from the earth would raise the cost of the base significantly and may even make it unfeasible. However, research has shown that due to various oxides in the composition of the regolith, the moon is approximately 45% oxygen by mass. NASA hopes to be able harvest this oxygen by collecting loose regolith and heating it in an hydrogen-rich environment, thus allowing the hydrogen to replace the oxygen in the chemical bonds. Much of the oxygen will then join with excess hydrogen and form water molecules. These will be sent through an electrolysis process, freeing the oxygen for use by the astronauts and recycling the hydrogen to use to extract more oxygen. A team of engineers from Auburn University was chartered to design and build a prototype harvester that would be used to collect the loose regolith found on the lunar surface. This report details the Auburn team's proposed design for a lunar harvester to meet the demands of a NASA regolith processing unit. This design has been broken into electrical and mechanical subsystems according to systems engineering practice, and is presented here for as a final detailed design ready for manufacturing.

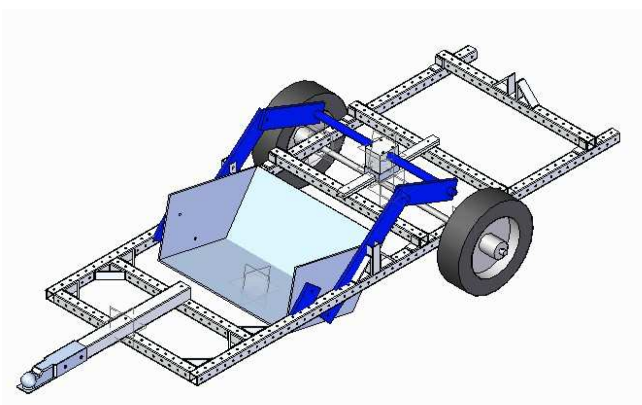


Fig 4.1 Harvester Isometric

Figure 4.2 Harvester Prototype

5.0 PROJECT MANAGEMENT

The Project Manager of the Lunar Harvester Prototype design is responsible for making final decisions, keeps team members on schedule, and keeps track of the cost and budget of the system. Also, the project manager has to interface between the corporate and program mangers and the group members. This includes discovering and defining the stakeholder requirements, as well as keeping the corporate and program managers aware of the design undertakings. To accomplish this, open lines of communication must be maintained. The Project manager is also responsible for managing the work breakdown of the group members, and assigning the Contract of Deliverables (CODs) to achieve the design goals.

The breakdown of the management structure is as follows (Fig 5.1):

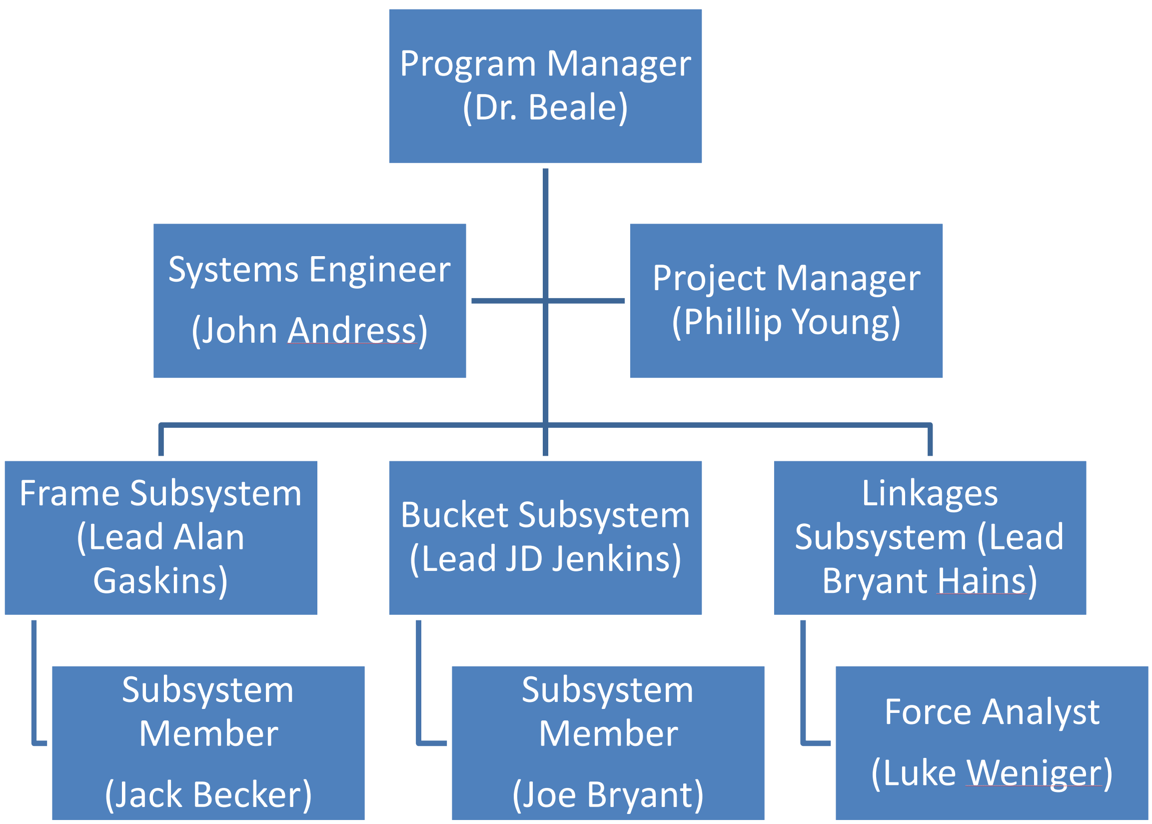
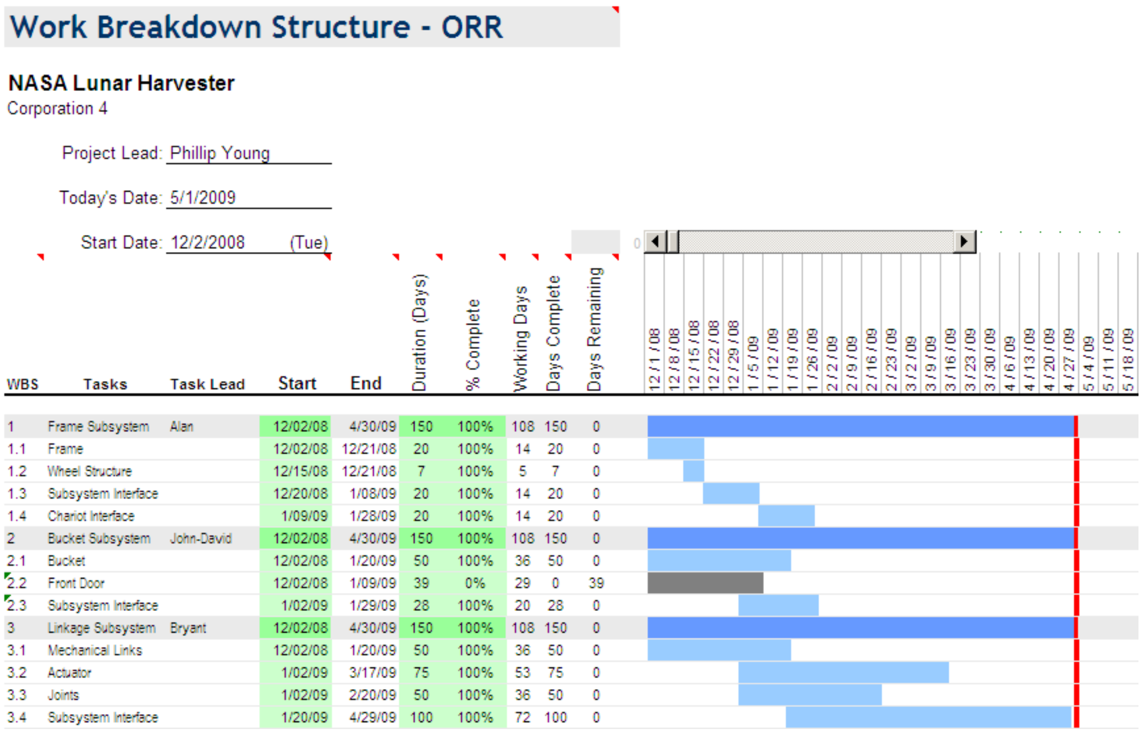


Figure 5.1 Management Breakdown Structure

The Subsystem Leads report directly to the Project Manager, and are responsible for defining the requirements and constraints of their corresponding subsystems. The subsystem leads are responsible for creating CAD modeling of their respective subsystems, as well as coordinating the drafts for manufacturing.

The tasks to be completed are assigned according to subsystem and are broken up to be equal time wise. The Gantt Task Chart showing the progress made on the design up to the date of the Preliminary Design Review is shown by the following figure (Fig 5.2):



Work Breakdown Structure - ORR

NASA Lunar Harvester
Corporation 4

Project Lead: Phillip Young

Today's Date: 5/1/2009

Start Date: 12/2/2008 (Tue)

WBS	Tasks	Task Lead	Start	End	Duration (Days)	% Complete	Working Days	Days Complete	Days Remaining
1	Frame Subsystem	Alan	12/02/08	4/30/09	150	100%	108	150	0
1.1	Frame		12/02/08	12/21/08	20	100%	14	20	0
1.2	Wheel Structure		12/15/08	12/21/08	7	100%	5	7	0
1.3	Subsystem Interface		12/20/08	1/08/09	20	100%	14	20	0
1.4	Chariot Interface		1/09/09	1/28/09	20	100%	14	20	0
2	Bucket Subsystem	John-David	12/02/08	4/30/09	150	100%	108	150	0
2.1	Bucket		12/02/08	1/20/09	50	100%	36	50	0
2.2	Front Door		12/02/08	1/09/09	39	0%	29	0	39
2.3	Subsystem Interface		1/02/09	1/29/09	28	100%	20	28	0
3	Linkage Subsystem	Bryant	12/02/08	4/30/09	150	100%	108	150	0
3.1	Mechanical Links		12/02/08	1/20/09	50	100%	36	50	0
3.2	Actuator		1/02/09	3/17/09	75	100%	53	75	0
3.3	Joints		1/02/09	2/20/09	50	100%	36	50	0
3.4	Subsystem Interface		1/20/09	4/29/09	100	100%	72	100	0

Figure 5.2 Gantt Task Chart

At this point, all tasks have been completed for the Operational Readiness Review. All tasks that were deemed necessary to complete the project have been fulfilled. Components that were removed from the final design, such as the Front Door in the Bucket Subsystem were discontinued at an early date. This allowed for maximum effort to be concentrated on mission-critical components like the completed Bucket, the actuator, and all of the subsystem interfaces.

Included in Project Management are the setting of Long-term milestones and design schedules so that group members can see what is expected of them and in what time frame. It is important that the subsystem leads and group members have ample time to plan how they will accomplish the tasks and CODS that are presented to them. A second semester Milestone and Phases schedule can be viewed in the following figure:

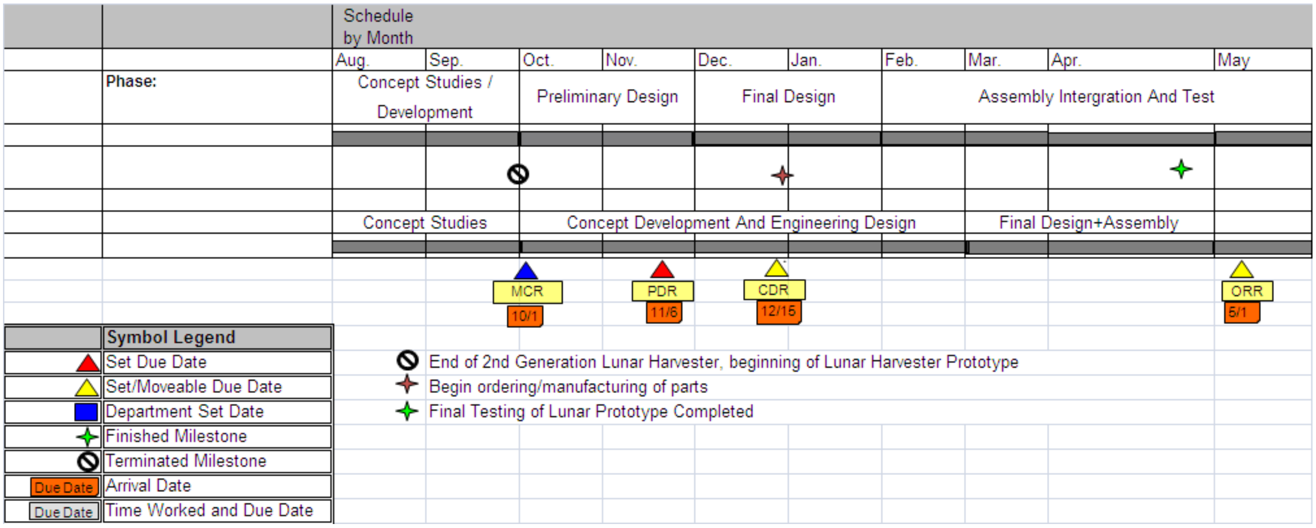


Figure 5.3 Milestone and Phases Schedule

Figure 5.4 Testing Group at USDA Facility

6.0 SYSTEMS ENGINEERING

6.1 Introduction into Systems Engineering

Corporation 4 is in the Operational Readiness Review portion of the Systems Engineering approach, also known as Phase D. The purpose of this phase is to assemble parts and components to create subsystems, integrate these subsystems to make the complete system, and then to test this system to verify that it meets the system requirements laid out in earlier Systems Engineering Phases. The system will be demonstrated to show that it meets performance and functional requirements in the various testing phases.

6.2 Mission Objectives and Requirements

A mission statement was developed to clearly define the goal and the expectations of the stakeholder of this design project.

> "Create a tele-operated lunar harvester prototype targeting less than 150 W power usage and weighing less than 100 kg for studies on the earth fulfilling environmental requirements of the moon."

Also developed were mission level requirements and subsystem level requirements. These derived requirements have evolved through the systems engineering process as new concepts were realized and enacted, trade studies with bucket analysis, and realization of stakeholder expectations. These requirements are either measures of performance (MOPs) or measures of effectiveness (MOEs), and were further classified as either functional or performance requirements in Phase B. As stated before, our mission level requirements are:

1. Shall be designed to conduct studies on earth but be able to operate in a Lunar environment (MOE – functional)
2. Shall interface with Gator utility vehicle (MOE – functional)
3. Shall be operated remotely (MOE – functional)
4. Shall collect and hold at least 50 kg soil per hour (MOP – performance)
5. Shall be designed to integrate Electrical Engineering subsystems into the mechanical design

The requirements become more detailed and specific at the subsystem level which will be addressed in the main body of the report.

6.3 Concept of Operations

The Harvester Prototype design has to operate in adverse environmental conditions in a precise manner. The Concept of Operations describes how the design will accomplish the mission and meet stakeholder expectations. The Concept of Operations for the Harvester Prototype is detailed in time-ordered sequence of events form, as well as graphical form.

Time-Ordered Sequence of Events:

i) Soil pan to harvest position, "scrape" soil from behind chariot/Gator Utility Vehicle in lane-like fashion until bucket reaches capacity
ii) Soil pan to transport position, chariot rover returns to collection point with soil pan pulled behind, only surface contact is soil pan wheels
iii) Chariot rover/Gator Utility Vehicle up and over soil ramp to position soil pan over hopper opening
iv) Soil pan to dump position, empty bucket contents into hopper
v) Soil pan to transport position, return to harvest area
vi) Soil pan to harvest position, begin "scrape" process

Graphical Form:

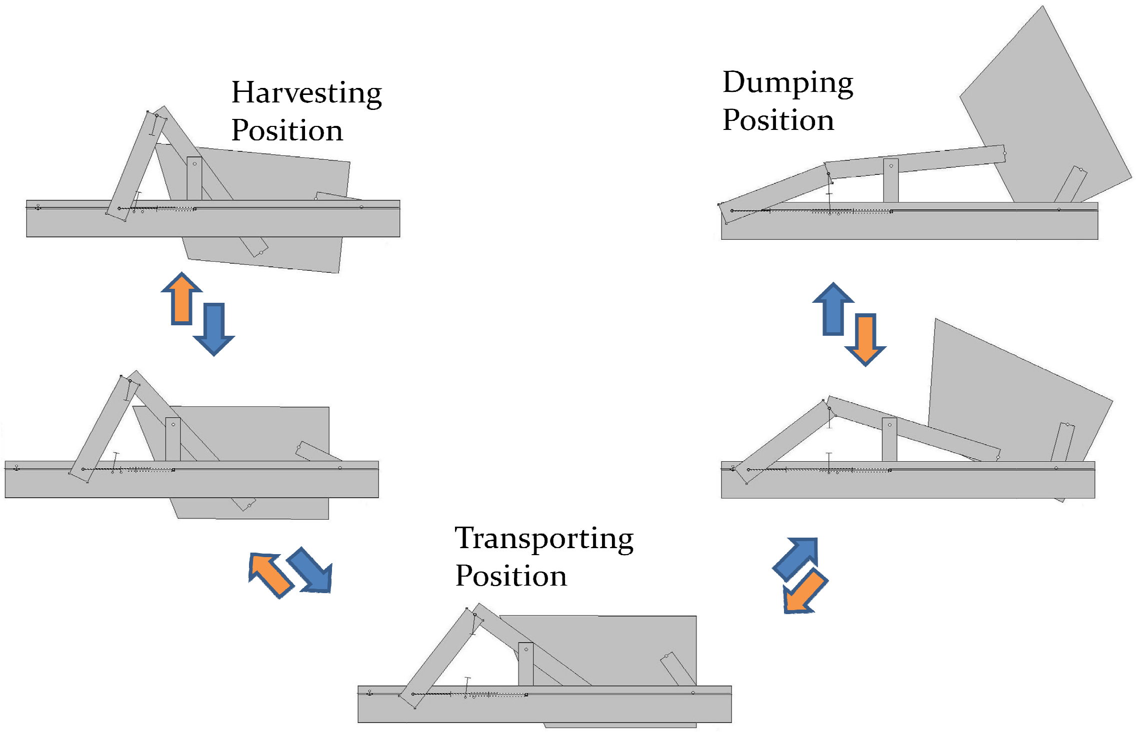


Figure 6.1 Graphical Concept of Operations

6.4 Architecture and Design – Product Breakdown Structure

The Lunar Harvester architecture can be detailed as a block diagram that becomes more detailed each tier. The architecture begins with the system level and progresses into subsystems and finally components. In Phase D, the Operational Readiness Review, the architecture includes all included named components. This structure has been referred to and updated in the current phase to include manufacturing methods as well as interfacing with other components. The architecture demonstrated on the next page serves as a starting point to understanding the outline of the design concept:

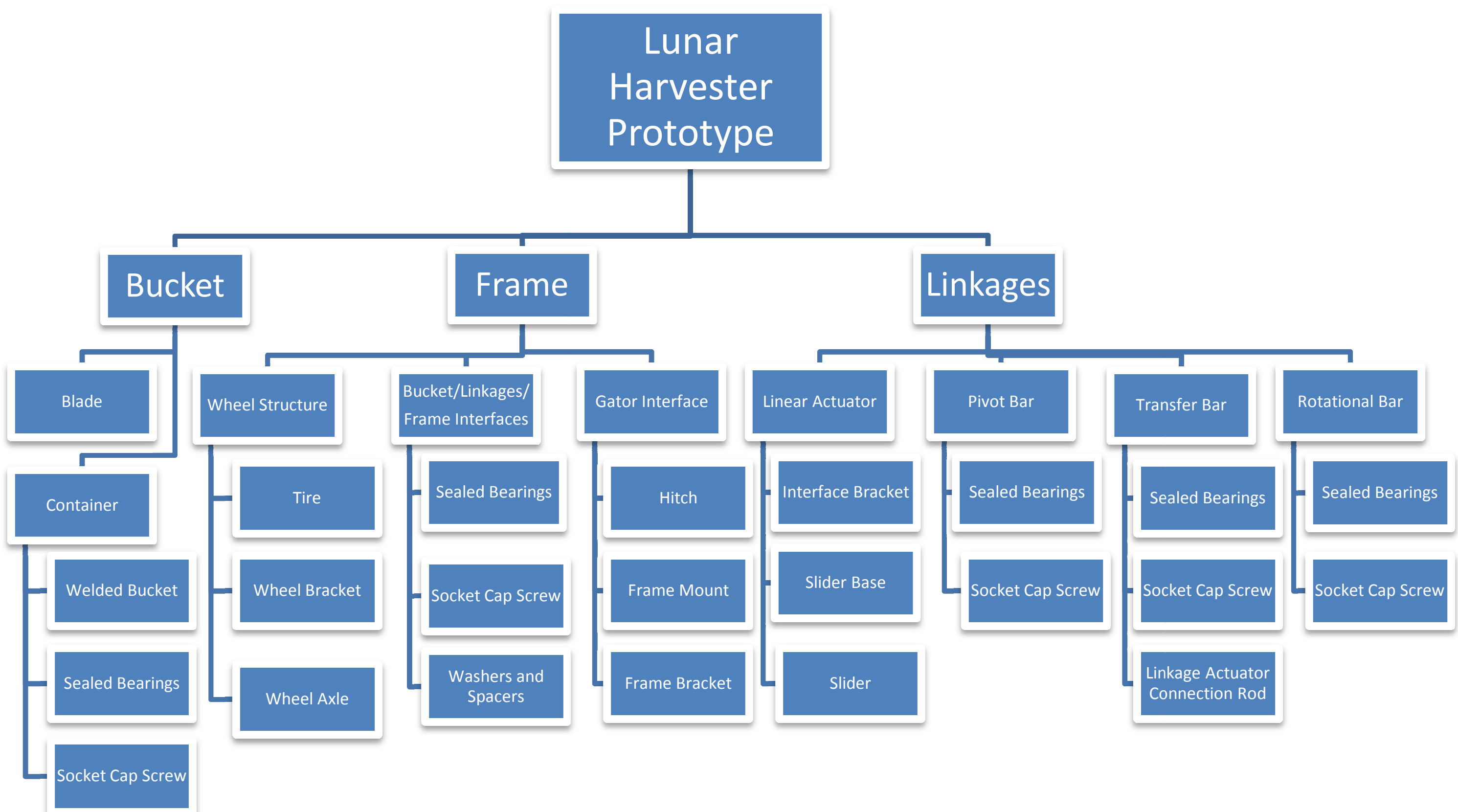


Figure 6.2 System Block Diagram

6.5 Validation and Verification

Throughout the systems engineering process, it is important to continue to make sure that the design is continuing to meet the stakeholder requirements, the derived requirements, and cost/weight budgets. In Phase B, verification of requirements for each subsystem was accomplished through either computer simulation to predict performance, engineering analysis, an inspection or a logical argument. In Phase C, more advanced computer simulation and engineering force analysis was utilized to predict performance.

Phase D Verification requires much more testing for functional and performance requirements. The following are verification tests performed of the subsystems to determine if the subsystems and components are built and interface correctly.

- Linkages assembled separately from system to test effectiveness. Bearings in the linkages must be effective in allowing free rotation. Bolts were tightened slowly and in a step fashion to determine correct amount of torque and to allow for correct amount of compliance.

- The frame was manually loaded to test for deflection and strength of components. The frame was determined to not flex enough to cause performance issues, but would need to be addressed in a moon-ready concept.
- Bucket subsystem was manually loaded to determine if strength of the bucket was going to be a problem. The weight of the bucket was a concern and would need to be addressed in a moon-ready concept. The strength verse weigh ration of the steel used in the bucket is not optimal.
- Tested for environmental conditioning – compared “loose” tolerances versus “tight” tolerances.

Phase D Validation was conducted to ensure that the system met all system requirements and the mission objectives. This is detailed more in Final Testing section of the ORR, which shows that all requirements and objectives were indeed met.

- Assembled total system for manual proof of concept testing. This consisted of manual movement of bucket positions, manual pushing of bucket through pseudo-regolith.
- Actuated through full range of motion (all bucket positions) in project room before testing with actual soil.
- Conducted proof of concept testing at USDA facility using all components and interfacing to the Gator vehicle actuating through full range of motion following the Concept of Operations. This included tests for time/speed, different digging depths, and amount of soil per run.

6.6 Interfaces

Interfaces exist between subsystems and in between components. These boundaries are required to successfully mate and integrate the subsystem/component. Often, the interfaces are needed to perform or limit a function. As a consequence of these technological necessities, interface requirements can be derived. These interfaces have all been addressed in the final prototype. The functional and performance requirements were all demonstrated in the final testing to show operational readiness. Functional and performance interface requirements for the Harvester Prototype design are:

1. Interface between harvester system and chariot rover interface plate shall have horizontal rotational movement (pin joint) to accommodate a turning radius and a raising radius (Functional). The ball hitch that was utilized allows for this “trailing” motion to be accomplished on the completed prototype.

2. Interface between bucket subsystem and frame subsystem shall be constrained to 1 DOF by revolute joint (Performance). The bucket fits with a very tight tolerance into the frame subsystem. Bolt heads barely "kiss" the side of the bucket as it rotates into its different functional positions.
3. Interface between linkage components shall be constrained to 1 DOF by revolute joint (Performance). Some compliance is allowed to ensure that unnecessary forces to not act on these joints.
4. Interface between actuator and frame shall be defined by 2 points and constrained to vertical motion only (Performance). The interface block and slider rail system allows this interface to function correctly. The block takes all of the load from the actuator in a horizontal direction and the load from the linkage interface bar in the vertical direction. The slider block allows unrestricted horizontal motion and constrains all motion in the vertical direction.
5. Interfaces shall be designed to accommodate lunar environmental conditions (Functional). The system was implemented using sealed components and "loose" tolerances, preventing dirt from affecting performance.

7.0 BUCKET SUBSYSTEM

7.1 Bucket Subsystem Requirements and Engineering Analysis

When generating the bucket subsystem specifications and constraints, manufacturability issues and the following functional and performance requirements were the primary criteria that guided the design of the subsystem.

- Functional Requirements

  1) Shall be designed to accommodate flow of regolith during dumping

  2) Shall provide a method of keeping regolith from spilling during transport

  3) Shall have a angled back wall to aid in harvesting and dumping

- Performance Requirements

  1) Shall hold 50 kg of soil ($V = \frac{m}{\rho} = 1.36\ ft^3\ using\ \rho = 1.3\ {}^{kg}\!/_{ft^3}$)

  2) Shall be able to accommodate a cutting blade mounted on the front edge of the bucket

To accommodate the angled wall requirement, a simple right trapezoid became the side view of the bucket. (Fig. 7.1)

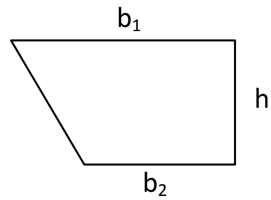


Fig. 7.1 Bucket

This side profile, along with a wall thickness and an open front for the entering regolith, yielded a shape for the entire bucket. (Fig. 7.2)

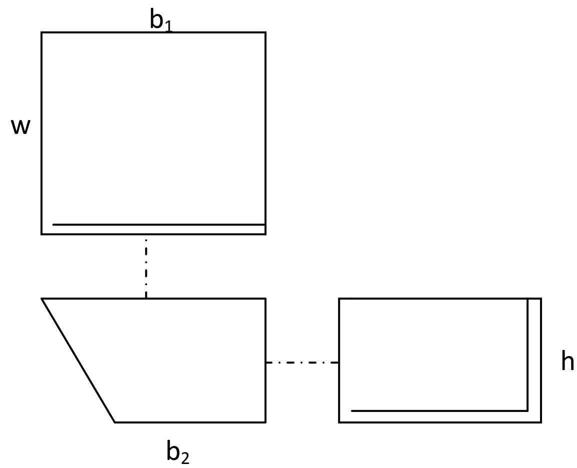
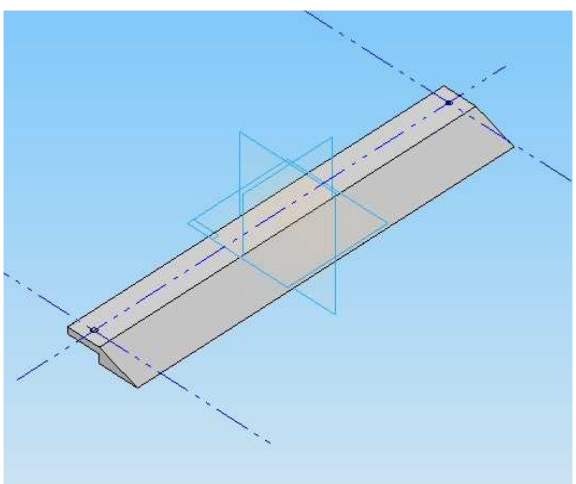
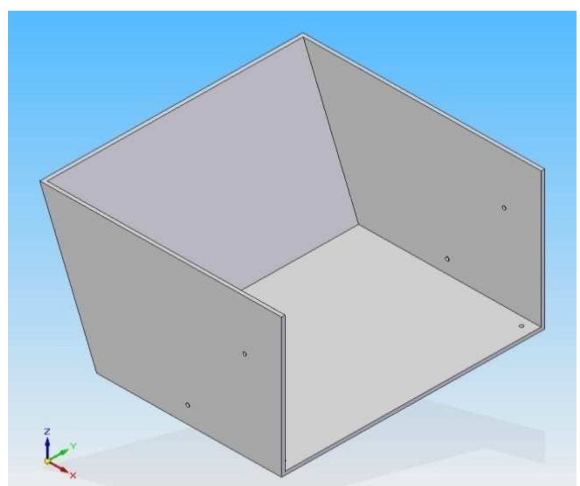


Fig. 7.2 Bucket 3rd Angle

When considering manufacturability, it was determined that this shape can be manufactured by using plates of steel for the three walls and the bottom and welding them together. Steel was chosen due to the lack of aluminum welders. The blade will be a simple steel wedge with a lip that bolts on to the front edge of the bucket. (*Fig. 7.3* and *Fig. 7.4*)



(Fig. 7.3) Blade Solid Edge



(Fig. 7.4) Bucket Solid Edge

When determining the size of the bucket needed, it was assumed that bucket would only fill to two thirds of the total volume. This made our target volume of the bucket V_t=2.04 ft^3. By setting the width (*w*) of the bucket at *w* = 2.0 ft, the area of the trapezoidal side can be found (Eq 7.1).

$$A = \frac{V_t}{w} = 1.02\ ft^2 = \frac{b_1 + b_2}{2} h \qquad (Eq\ 7.1)$$

After setting the angle of the rear wall at 70°, a table of potential bucket dimensions was created (*Table 7.1)* shown on the next page.

Height (ft)	Average Base (ft)	Total Length (ft)	Smaller Base (ft)	Ratio Width/Length
0.5	2.037384619	2.385797908	1.688971329	0.83829397
0.6	1.697820515	1.988164923	1.407476107	1.005952764
0.7	1.455274728	1.704141363	1.206408092	1.173611558
0.8	1.273365387	1.491123693	1.055607081	1.341270352
0.9	1.131880344	1.325443282	0.938317405	1.508929146
1	1.018692309	1.192898954	0.844485664	1.67658794
1.1	0.926083918	1.084453595	0.76771424	1.844246734
1.2	0.848910258	0.994082462	0.703738054	2.011905528
1.3	0.783609469	0.91761458	0.649604357	2.179564322
1.4	0.727637364	0.852070681	0.603204046	2.347223116
1.5	0.679128206	0.795265969	0.562990443	2.51488191

Table 7.1 Dimension Iterations

The red entries in the table were discarded because the ratio of width to length was either below one or too close to one, and a bucket was desired that was wider than it was long. The blue entry was chosen because the width to height ratio was acceptable and the height was still low, allowing the regolith to accumulate to closer to the maximum volume. These dimensions can be seen in inches in *Table 7.2*, as well as slightly modified dimensions to use simpler numbers.

Height (in)	Average Base (in)	Total Length (in)	Smaller Base (in)
9.6	15.28038464	17.89348431	12.66728497
10	15	17	13

Table 7.2 Final Dimensions

7.2 Bucket Subsystem Concept Presentation

There are a couple of different ideas that have been considered for the final design of the bucket subsystem. The previous design of the bucket was just a scoop/shovel with a vibrating bit that scraped the regolith off the surface and provided transportation to the conveyor belt. The conveyor belt then carried the moon dirt to a storage bin for transportation to the regolith hopper. The purpose of the vibrating bit was to help reduce the draft force on the scoop/shovel. After testing the current design, it was observed that the scoop assembly with the vibrating bit was not effective. The vibrating bit would stop oscillating when pushed through the soil. Also, the vibrating bit assembly was mounted directly to the bit and added approximately 25 lbs. to the total weight of the scoop.

The proposed design of the bucket subsystem consists of two parts, the bucket and the blade. The blade is bolted to the inside of the bucket and has a knife-like edge that cuts regolith from the surface and provides a ramp for the moon dirt to slide into the bucket. The bucket acts like a storage bin as the regolith is harvested and transports the harvested material to the hopper for processing. This proposed design takes the place of the scoop, vibrating bit assembly, conveyor belt, and storage bin that is required for the previous design. By eliminating these components, the design is simplified in a couple of ways. One way the design is simplified is that we are reducing the total amount of power needed to run the system by eliminating the voice coils, actuator for the scoop, and the motor and controller for the conveyor belt. The current design eliminates complex subassemblies (i.e. conveyor belt, vibrating bit) that have many different parts that move and have to be controlled. The current design is controlled by simple mechanical linkages and two linear actuators.

7.3 Blade Force Analysis

Currently, the force analysis acting on the blade is done using a model proposed by Mckeys and Ali. This method relates the proportions of the failure mechanisms to the observed shapes. Typical variables that are considered in this model are listed in table 7.1.

Table 7.3 Typical Variables to be considered in Mckeys and Ali Model

Notation	Definition	Units	Value
α	Tool Angle from Forward Horizontal	degrees	10
β	Rupture angle from direction of travel	degrees	(value where N_γ is minimized)
b	Tool Width	cm	Varied
c	Cohesional Factor	N/cm^2	.09
C_a	Adhesional Factor	N/cm^2	.00009
δ	Soil-Tool friction Angle	Degrees	24
ϕ	Soil-Soil friction Angle	Degrees	37
γ	Unit weight of soil	N/cm^3	.01884
q	surcharge	N/cm^2	N/A (zero in this case)
r	Crescent Radius	cm	(varies)
z	Depth of cut	cm	5
H	Draft force	N	(varies)
P	Total force on blade	N	(varies)

In this model the blade causes soil to move in front of and to the sides of the blade. For this model the blade must be flat and create a wedge shaped soil boundary. This wedge is considered to be circular and has a crescent radius (r) that is defined by Equation 7.2.

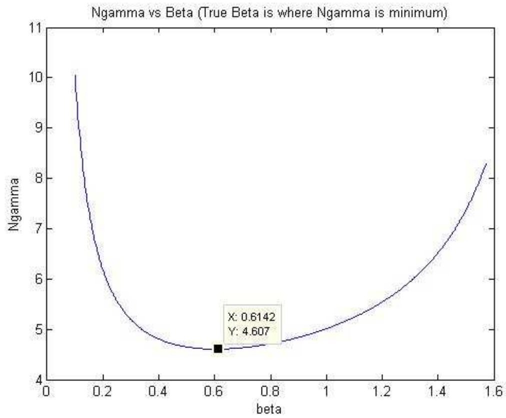


Figure 7.5 Plot of Equation 7.3 with b=60.8cm and z=5cm

$$r = z(cot * (\alpha) + \cot(\beta) \qquad (Eq\ 7.2)$$

The variable β in equation is the soil parameter that is found by minimizing Equation 7.3. This is the angle that the failure wedge creates with the direction of travel and is called the rupture angle. For use of this equation, the dimension "s" must be determined by using Equation 7.4.

$$N\gamma = \frac{\frac{1}{2}(Cot(\alpha)+\cot(\beta))\,\{1+\frac{2s}{3b}\}}{\cos(\alpha+\delta)+\sin(\alpha+\delta)\cot(\beta+\phi)} \quad (Eq\ 7.3)$$

$$s = r[1-\left(\frac{\cot(\alpha)}{\cot(\alpha)+\cot(\beta)}\right)^2]^{\frac{1}{2}} \qquad (Eq\ 7.4)$$

Figure 7.2 shows the plot equation 7.3 with a tool width 'b' of 60.8cm (23.75) and a tool depth 'z' of 5cm (1.97inches). Seen in figure 7.5, β is then equal to 0.6142.

Using these calculated values for r, s, and β; the total force acting on the blade is defined using equation 7.5.

$$P = \frac{\left(\left\{\left[\frac{1}{2}\gamma z^2 \frac{r}{z}\left(1+\frac{2s}{3b}\right)\right] \sin(\alpha+\beta) + cz\frac{\cos(\phi)}{\sin(\beta)}(1+\frac{s}{b})+c_a z(\frac{\cos(\alpha+\beta+\phi)}{\sin(\alpha)}\right\} b\right)}{\sin(\alpha+\beta+\delta+\phi)} (Eq\ 7.5)$$

P was then plotted using values of blade width close to the defined blade width of 60.8cm assuming small change in β (valid for values of b close to assumed value). This plot resulted in figure 7.6.

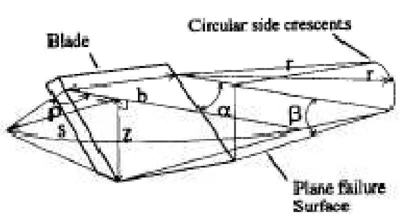


Figure 7.5b Rupture Surface Proposed by Mckeys and Ali

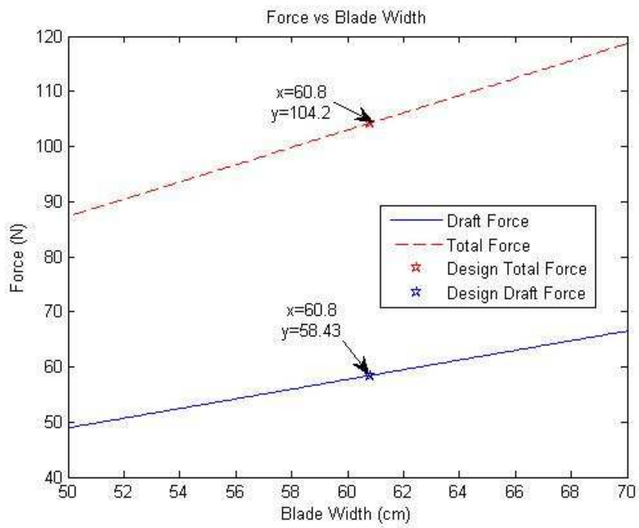


Figure 7.6 Plot of Equation 7.5 for values of blade width assuming small changes in beta.

From Figure 7.3 the total draft force as seen for our design is 58.43N and the total force acting on the blade is 104.2N. Using basic trigonometry the vertical component of this force is defined with equation 7.6.

$$H = Psin(\alpha) \qquad (Eq\ 7.6\ )$$

Solving Equation 7.6 for our design, the vertical component of the force P is 18.09N. This means that the minimum weight of the harvester must overcome this vertical force to keep the blade in the soil.

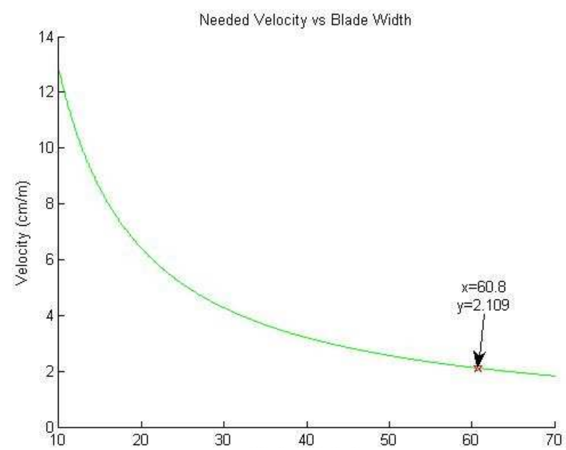


Figure 7.7 Needed Velocity to harvest at least 50kg/h of regolith

Another point to consider when varying the blade width is the required velocity to acquire the minimum of 50kg/h of regolith. The mass flow rate is defined by equation 7.7 and can be solved for the required velocity. Then the required velocity is then plotted with a varied blade width. Seen in figure 7.4, the chosen blade width of 60.8cm (23.75inches) requires a velocity of 2.109cm/min to collect at least 50kg/h of regolith. However, this velocity is too low for our system to travel. The slowest velocity we can travel is about .45m/s (1mph). At this velocity the system will collect 585kg/hour if operating at 100% efficiency. Assuming the system operates about 75% efficiency, the system will harvest about 440kg/hour which is way above the mission objective.

$$V = \frac{\dot{m}_{req}}{\rho A_{cross}} \qquad (Eq\ 7.7)$$

However another force that may be larger than the standard digging force will be exerted on the entire system if we impact an object that doesn't move (i.e. a rock). If the system travels a nominal velocity of 1mph and a maximum mass of 330lbs stopping nearly instantaneously (.01s), using Equation

7.8, the impulse average force is calculated to be 1500 lb-force. If this force were translated through the linkage system, the actuator would experience a force of about 1900lbs acting vertically and about 1458lbs acting along the axial direction.

$$Impulse = F_{avg}\Delta t = m\Delta Vel \qquad \text{Eq 7.8}$$

In addition to the forces acting on the system by the regolith and the impulse forces, the mass of the bucket and collected regolith create a moment about the joint of the bucket during dumping. Initial testing showed us that the force created by the mass of the bucket alone acting on the actuator was about 300lbs during its dumping phase. If the bucket were fully loaded at the time of this, the force acting along the actuator would be about 305lbs.

8.0 LINKAGE SUBSYSTEM

8.1 Linkage Subsystem Requirements

The linkage subsystem is composed of all the linkages necessary to move the bucket into the three necessary positions:

1. Dumping
2. Transport
3. Digging

This will be accomplished by the use of three linkages that are mirrored on either side of the assembly. It was discovered through prototyping in both solid edge and working model that it is possible to overextend the actuator so that the bucket enters into an unrecoverable position. This problem is easily remedied by controlling the motion of the actuator and by designing a system of mechanical stops in the next phase of the design.

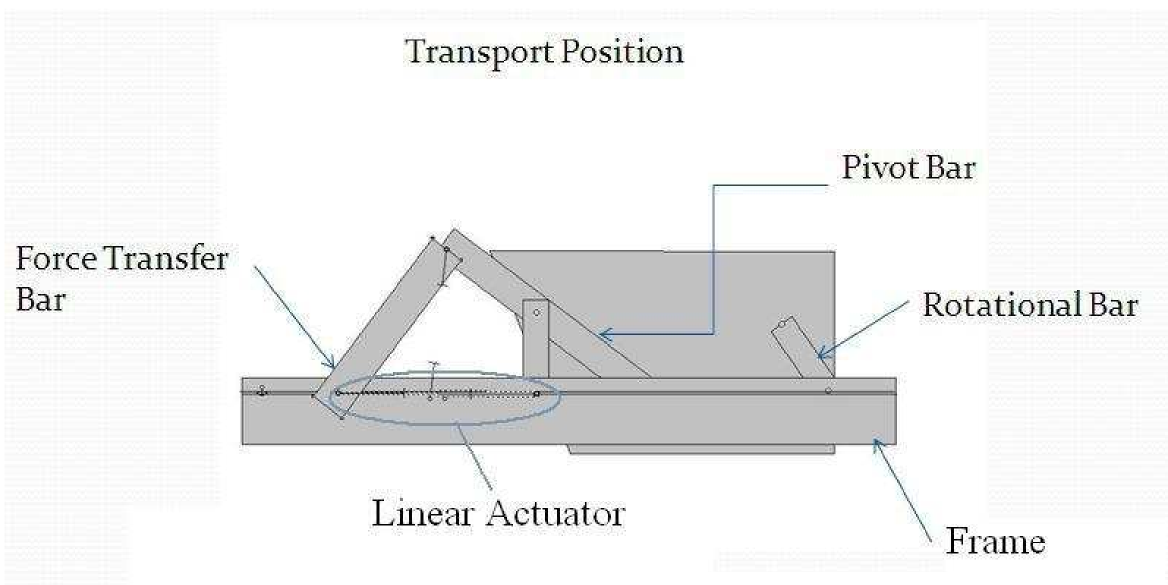


Figure 8.1 Linkage Subsystem Nomenclatures

The subsystem requirements of the linkage subsystem are as follows:

1. Shall be able to move bucket into the three desired mechanical positions
2. Shall be powered by motorized actuator
3. Shall provide mechanical advantage in operating bucket
4. Shall constrain bucket movement to safe bounds

Force Transfer Bar:

The force transfer bar is attached to the actuator and to the pivot bar. Its purpose is to drive the motion by way of a linear actuator. The link will provide significant mechanical advantage, which will make moving the bucket possible with a smaller actuator. This link undergoes both rotation and translation.

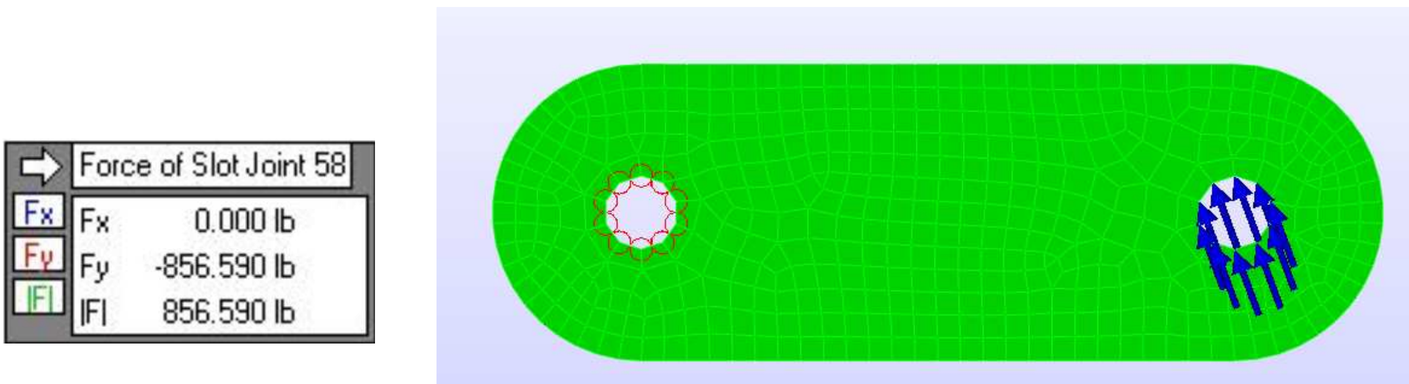


Figure 8.2 Force Transfer Bar Load Condition

The max stress is approximately 75000 psi, and a max displacement of .04 inches. The FEA of this bar revealed in the worst case scenario there may be minor yielding.

Pivot Bar:

The pivot bar is attached to the force transfer bar, frame, and bucket. It purpose is to rotate and lower the bucket while keeping the bucket horizontal in the digging position, and allowing for dumping of regolith into the regolith hopper at the processing plant. Its motion is pure rotation.

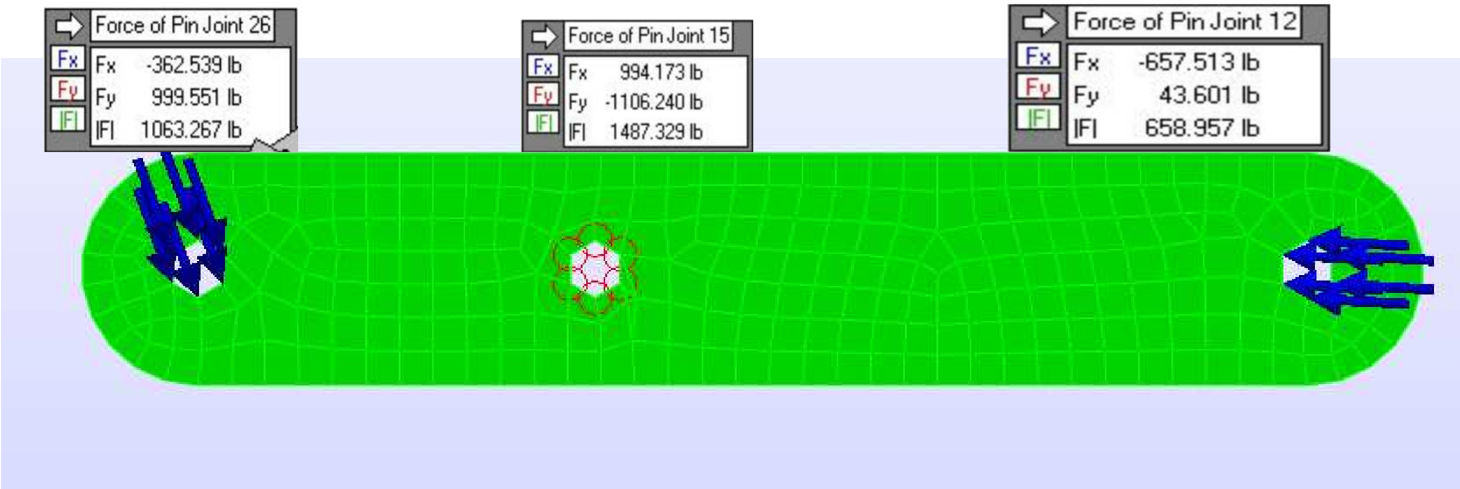


Figure 8.3 Pivot Bar Load Conditions

The max stress is 25000 psi, and a max elastic deformation of .01 inches. The FEA revealed that even in the worst case scenario, there will be no yielding of the bar.

Rotational Bar:

The rotational bar is attached to the frame and bucket. Its purpose is similar to the pivot bar in that it provides the necessary motion to place the bucket in both dumping and collecting mode. Its motion is purely rotational as well.

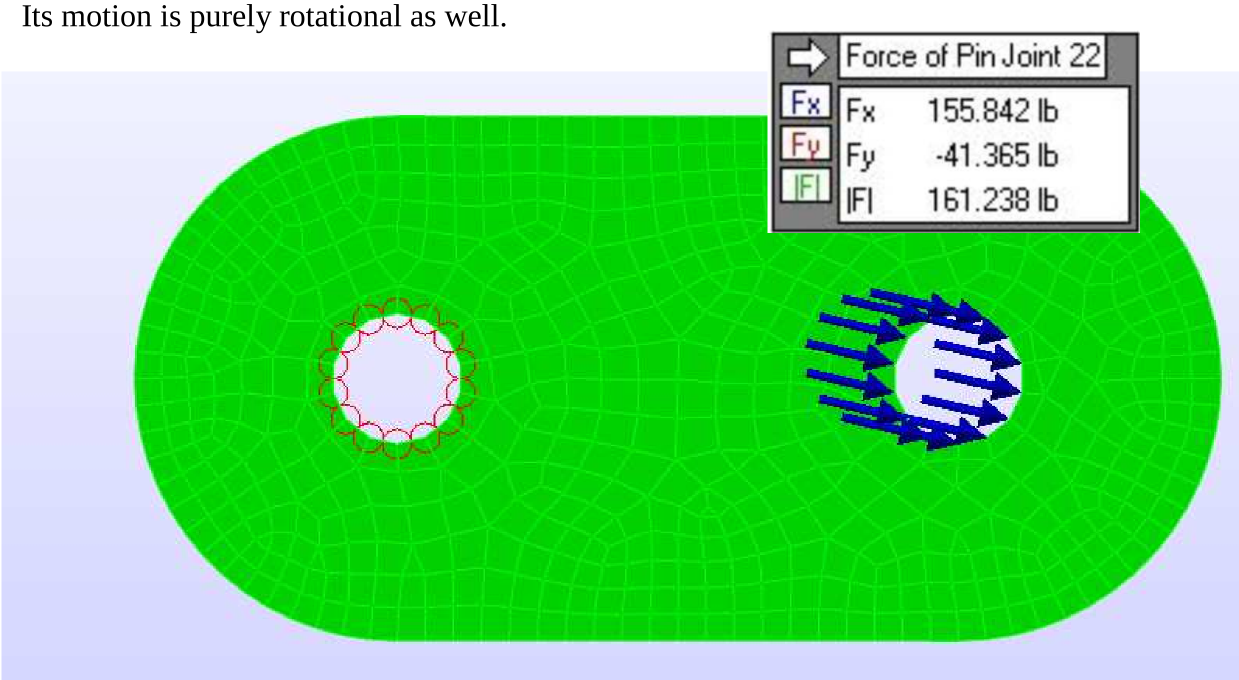


Figure 8.5 Rotational Bar Load Condition

The max stress is 5720 psi, and a max displacement of .0009 inches. Again, the FEA revealed that even in the worst case scenario, there will be no plastic deformation.

Linkage Interface Bar:

The Linkage interface bar is the bar that connects the actuator and the force transfer bar. The intent of this bar is to reduce the number of actuators to one. All the forces that propagate through the system act upon the bar making it necessary for the bar to be quite large to limit the deflection of the bar.

The initial bar was designed to have a diameter of ½”. The ends of the bar were to be machined down to 5/16”.

Initial testing showed an unacceptable level of deflection in the bar. Deflection was as much as 4 inches at the interface between the bar and the linkages. This was remedied by increasing its diameter to one inch and removing the decrease in diameter. This necessitated the purchase of new bearings and the modification of the force transfer bar, but the results of the changes were immediate and effective. Deflection was reduced to 1/10 of an inch. By limiting the deflection, the fatigue loading of the bar is reduced. This will increase the overall lifespan of a vital part of the design.

8.2 Concept Presentation

The goal of the Linkage Subsystem in regards to the overall system is to both raise and lower the collection bucket for dumping, transporting, and harvesting regolith. The more specific derived requirements are:

1) Shall be able to move bucket to and support at three desired mechanical positions
2) Shall be powered by motorized actuator
3) Shall provide mechanical advantage at harvesting position and keep forces reasonable when dumping
4) Shall constrain bucket movement to safe bounds
5) Shall allow variable digging depth that includes the range of 1-5cm

The challenging part in designing this subsystem is the goal of controlling a complex series of movements with one input (an actuator). Referencing an existing system that provides the desired movements seemed like a good starting point. The most available resource was J.D.’s dirt pan, pictures and videos of which were already on hand. The product, SoilMover, is a simple enough machine, powered by two linear actuators (symmetrical) and a straightforward linkage system (Fig 8.7).

Fig 8.7 Industrial Dirt Pan

From this existing model, as well as input from design team members and with a working knowledge of kinematics, a 2-D scale model was made using the Working Model program to function as a preliminary design for the linkage system. The benefits of using a program like Working Model is that the mechanics of the system can be easily viewed as well as measured and the model can be simply tweaked and altered to fit evolving requirements and bounds. The product of that effort is this model, representing one side of the symmetrical system (Fig 8.8). The full, range of motion of the model and the key labeled positions can be seen in (Fig 6.1).

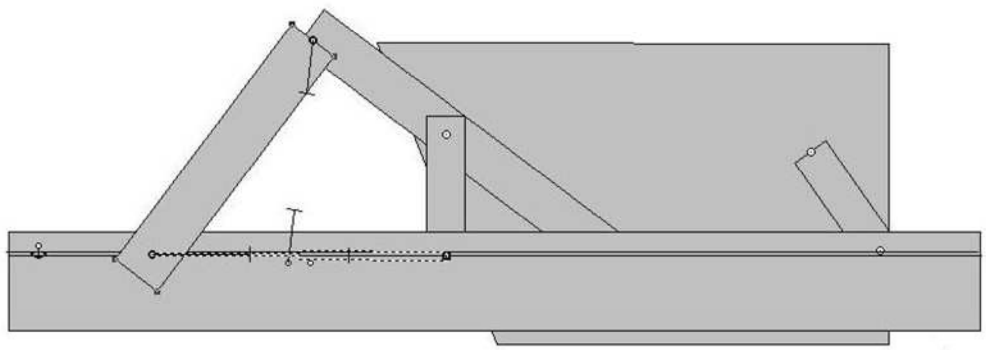


Fig 8.8 Working Model Transport Position

From this, a Solid edge 3-D representation was subsequently developed (Fig 8.9).

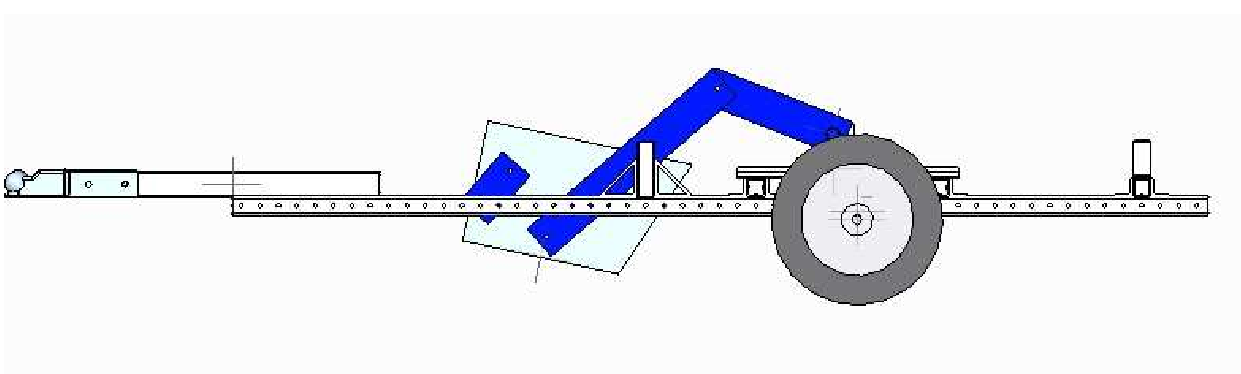


Fig 8.9 Solid Edge Side View

8.3 Working Model and Solid Edge Engineering Analysis

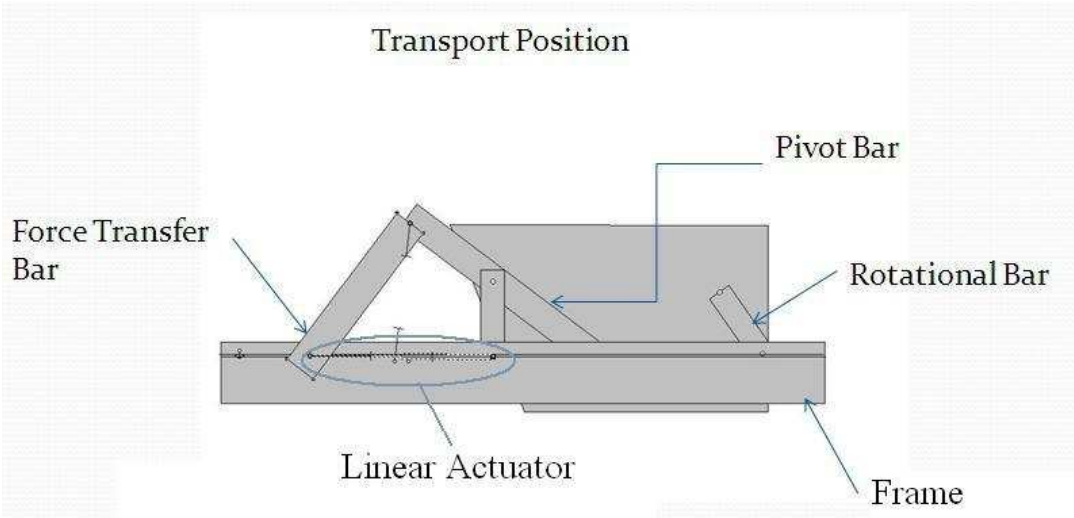


Fig 8.10 Linkage Component Names

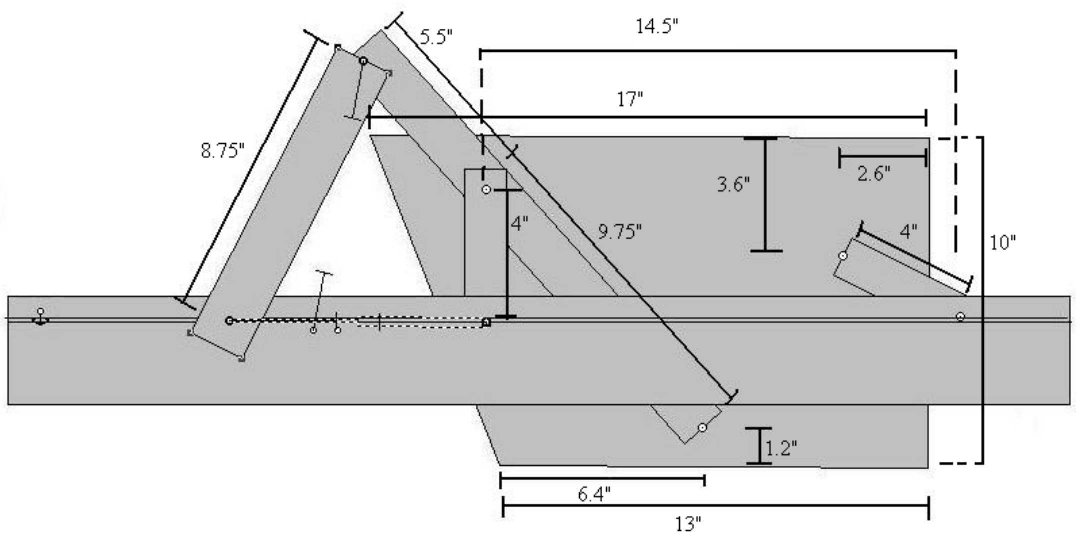


Fig 8.11 Linkage Subsystem Dimensions

Shown above in Fig 8.10 is the "Transporting" position where the bucket would be held when neither dumping the regolith nor harvesting it. The dimensions of the linkage system were all designed around the determined ideal bucket dimensions (13" width on bottom, 10" tall, and 17" width at top) (Fig 8.11).

The bucket is attached to the frame with 2 links: the Rotational Bar (4") at the front and the Pivot Bar (9.75" to the pivot point) near the middle. The Pivot Bar extends past the pivot point another 5.5". The front joint on the bucket is 2.6" from the front and 3.6" from the top, and back joint is 6.4" from the bottom back corner and 1.2" from the bottom. The 2 attachment points from the links to the frame are 14.5" apart with a 4" height difference (the back one is basically an anchored link). The Pivot Bar is attached at the far point to the Force Transfer Bar (8.75") which is attached to the frame via a slider joint and the linear actuator (Fig 8.11).

With this series of connections, the actuator only has to move in a straight line (as opposed to pivoting) and it has a large mechanical advantage when harvesting the regolith (Fig 8.12).

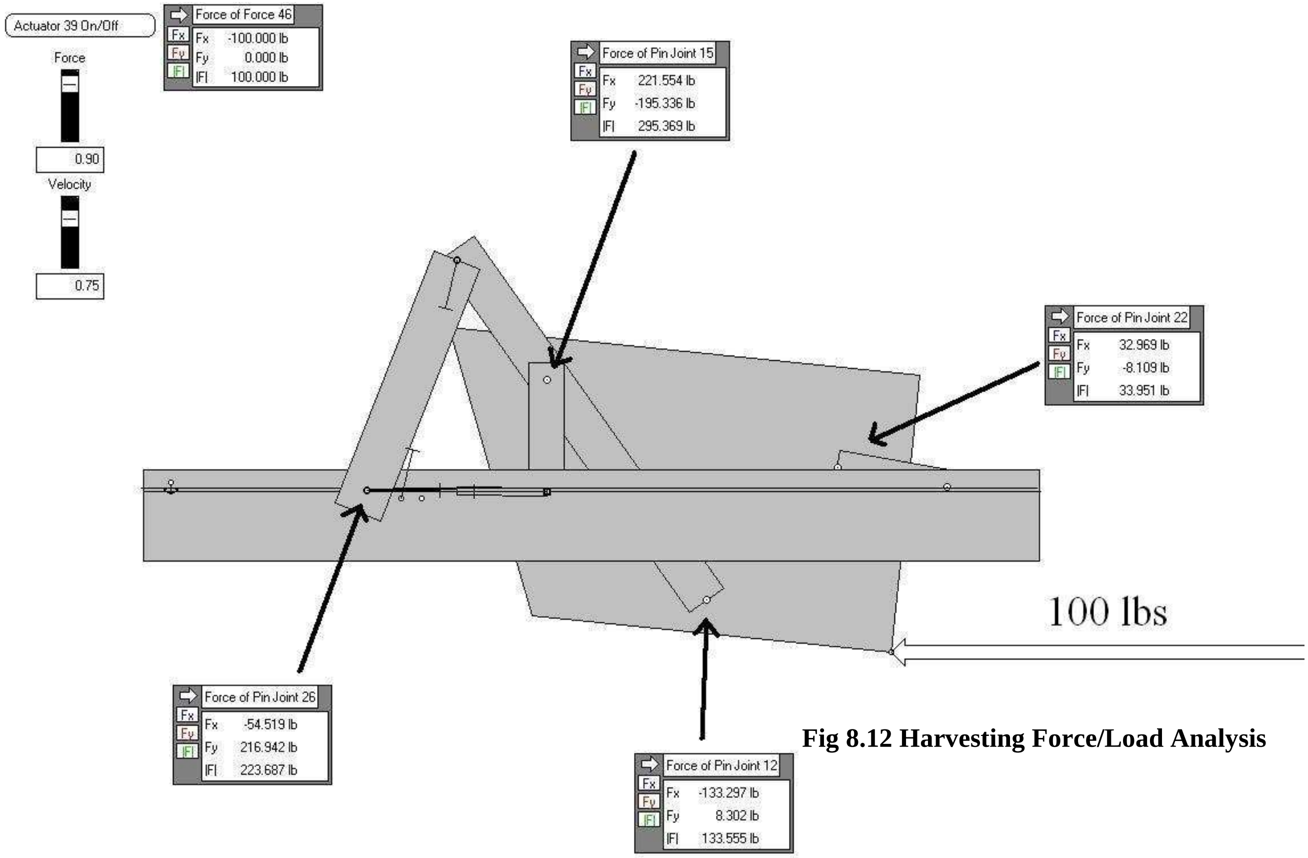


Fig 8.12 Harvesting Force/Load Analysis

Using Working Model, a force of 100 lbs was applied at the harvesting edge of the bucket while in the "Harvesting Position" and the reactions at specific joints were measured. The 100 lbs value was used just for comparative purposes, as the exact force evaluation is varying. Demonstrated, though, is the advantage of the slider/actuator design in that the required force of the actuator (in the x-direction) is 54.5 lbs compared to the 100 lbs input. The majority of the load is dispersed to the pivot of the Pivot Bar (joint 15) and the y-direction of the slider that the actuator travels along.

Another force, representing the load of the regolith in the bucket, was applied to the model which can be seen on the following page:

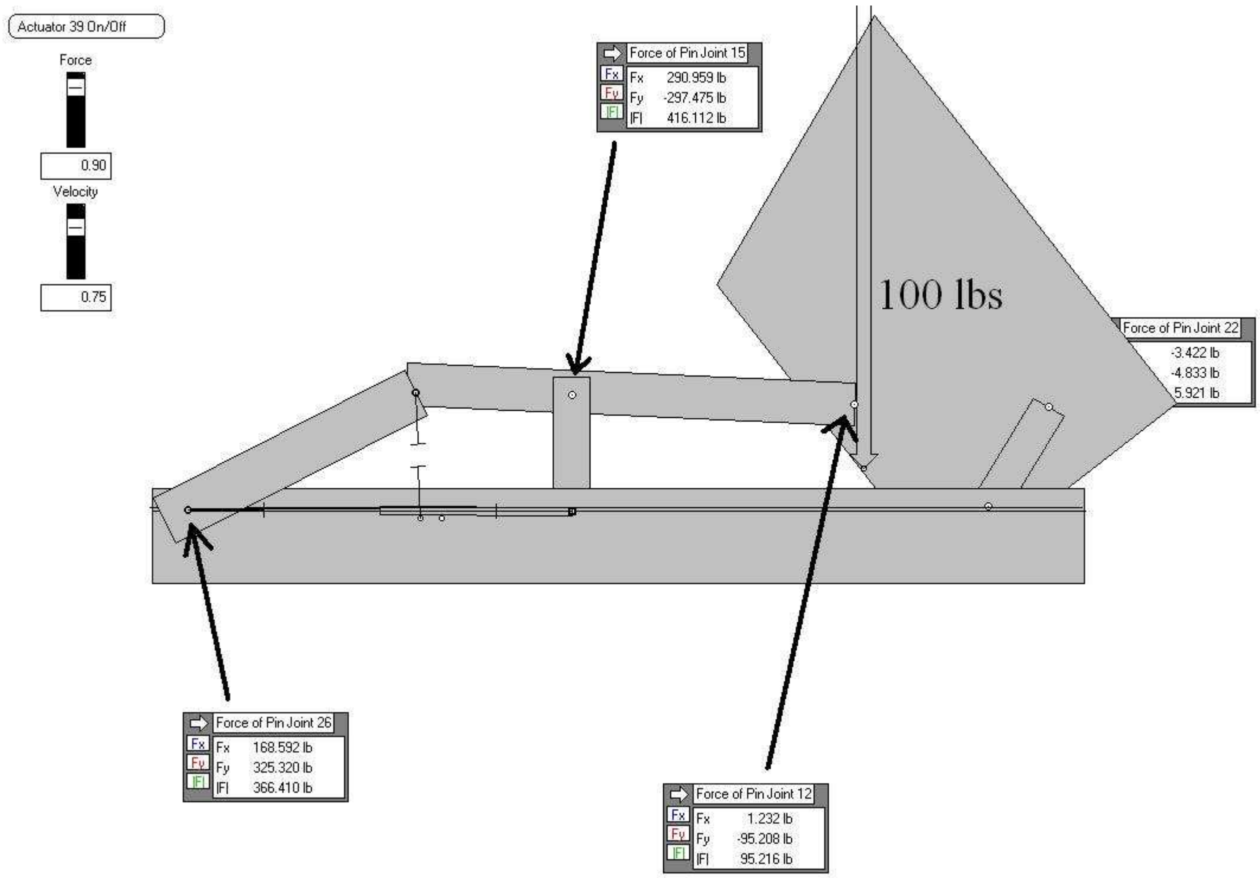


Fig 8.13 Dumping Force/Load Analysis

This load was also set at 100 lbs, but this was determined roughly by the density and volume of regolith to be transported. At the “Dumping Position” displayed in the picture (approximately 60-70 degrees), or where the regolith begins to slide, the force required by the actuator is 168 lbs. This is presumably the maximum force the actuator will have to provide and it is probably even inflated since the front most layers of regolith will already have dumped at this point. Displayed also is the necessity for a strong support at the pivot of the Pivot Bar. The Actuators are of the slider variety, combining the force application and the slider function into one. These are discussed further in the Actuator Analysis section. Joints will be connected with a series of bearings and bolts, discussed further in the Bearing Analysis section.

Adjustments were made to the Working Model to correspond to and design around interferences with the mounting height of the actuator when the final design was being assembled. The basic relations remained the same, but some lengths of links changed. These new lengths are detailed in the linkage drafts and can be seen in Figure 8.14.

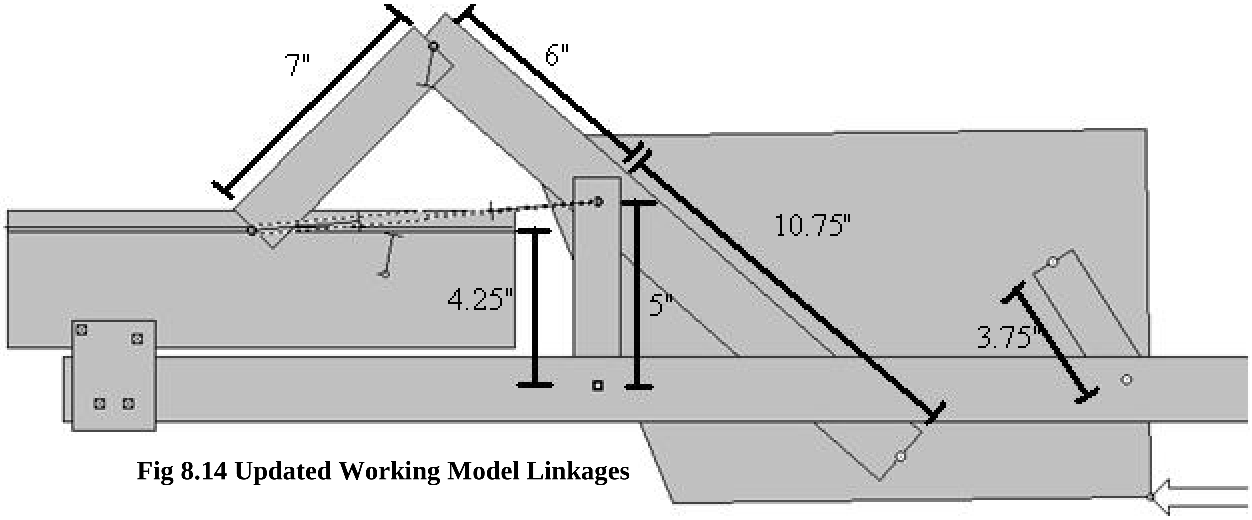


Fig 8.14 Updated Working Model Linkages

This new model was then tested with the same force applied to the original model and the responses were very similar (Figure 8.15)

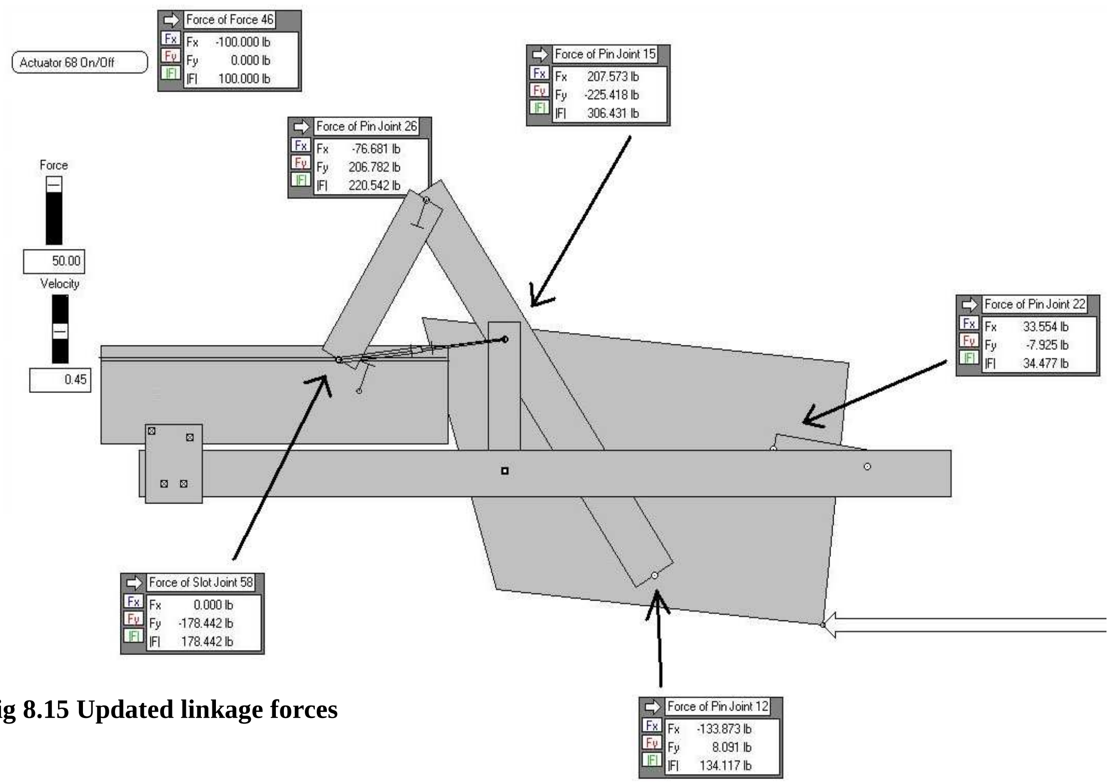


Fig 8.15 Updated linkage forces

The final product in Solid Edge has the full range of motion denoted by the requirements. To satisfy the final requirement of varying digging depth, the wheels were designed around the adjustable digging depth of the system. It allows ample room room for transporting, aided further by the potentially increased height of the Chariot Rover mount (Figure 8.16).

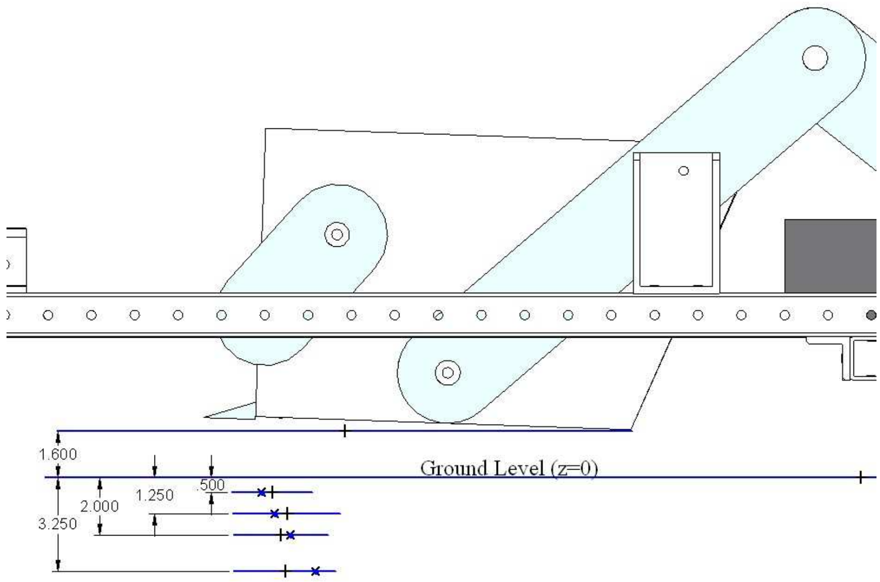
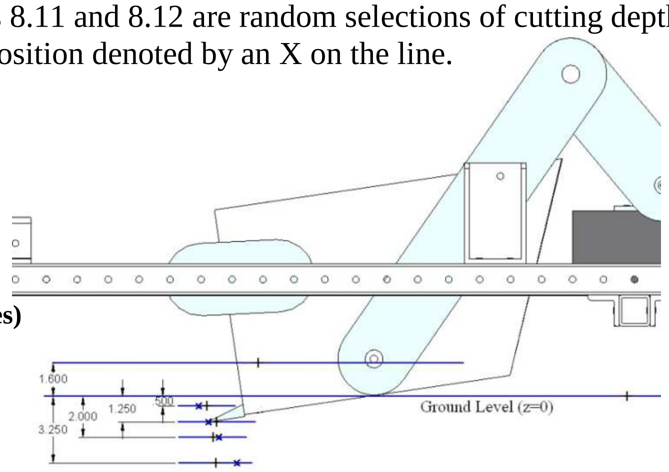


Fig 8.16 Transporting Clearance (All measurements in inches)

The bucket, when lowered, has a wide range of digging depths, from 0" to 3.25", easily encompassing the range given in the requirements and simply adjustable by the user. The horizontal lines marked in Figures 8.11 and 8.12 are random selections of cutting depths it can move through with the blade tip position denoted by an X on the line.

Fig 8.17 At 1.25" Digging Depth (inches)



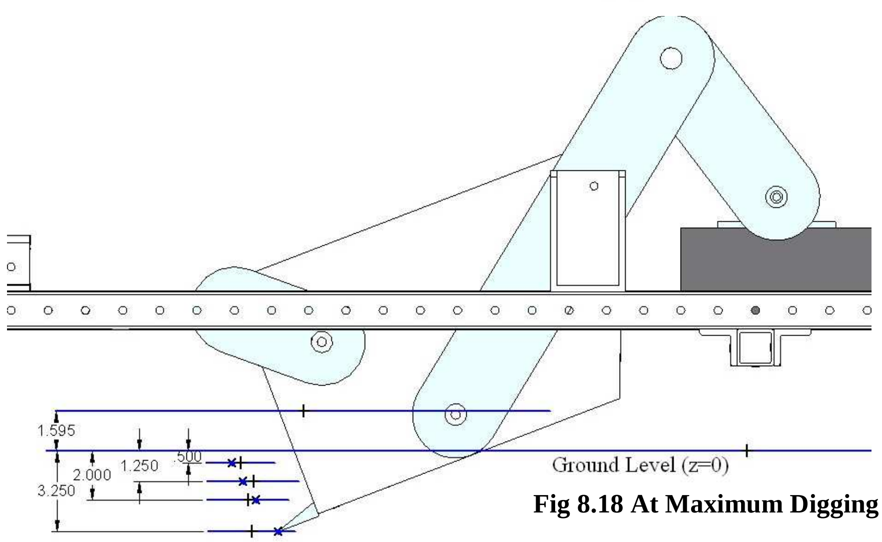


Fig 8.18 At Maximum Digging Depth (In inches)

When dumping, the bucket bottom reaches an angle of 70 degrees, the angle necessary for regolith to slide (Figure 8.19).

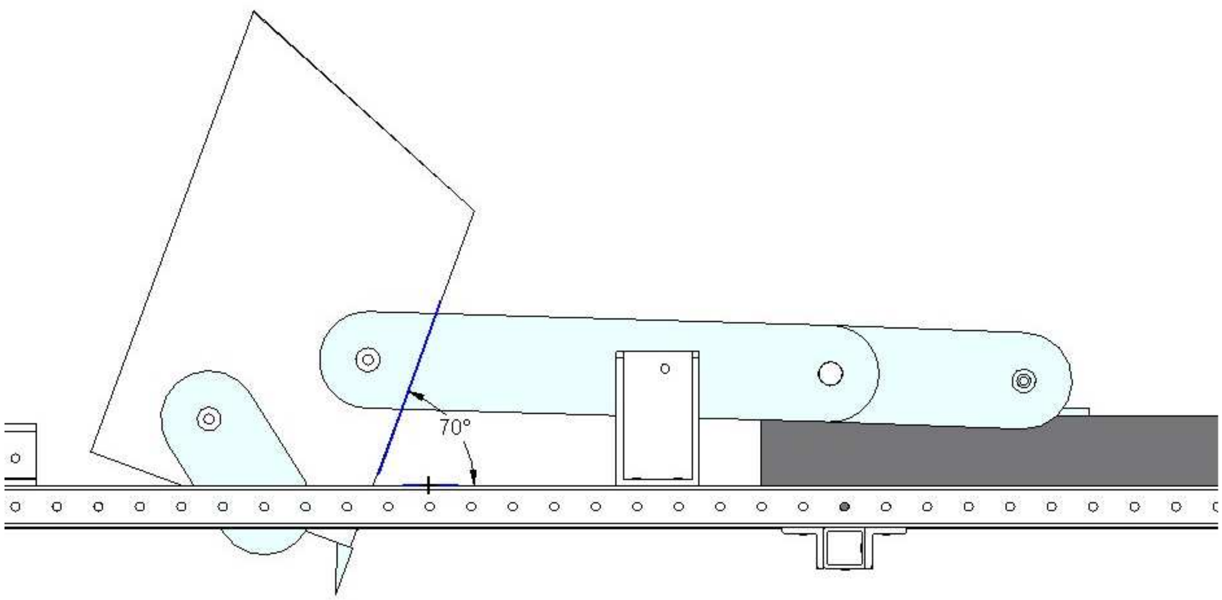


Fig 8.19 Dumping Position Angle

Prototype Modifications and other Considerations:

In manufacturing the final design as detailed above, several modifications had to be made due to either budget constraints or machining difficulty. The first modification was to the interface between the pivot bar and the frame. The designed bracket would be both difficult and expensive to manufacture, so a bracket was made from existing 80/20 parts in our possession (Fig 8.20).

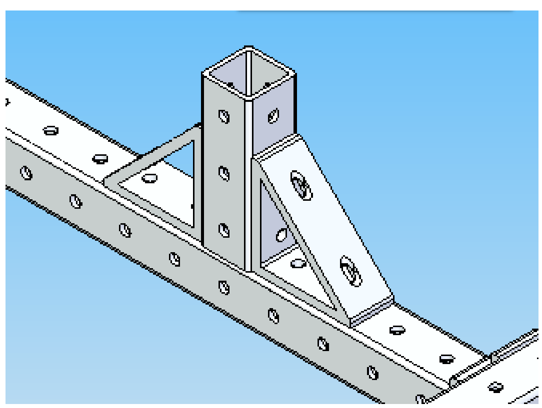


Fig 8.20 Pivot bar mount

Because of the predetermined hole intervals on the 80/20, the distance between the pivot bar mount and the rotational link in both the x and y directions. The new relationships are shown in Fig 8.21.

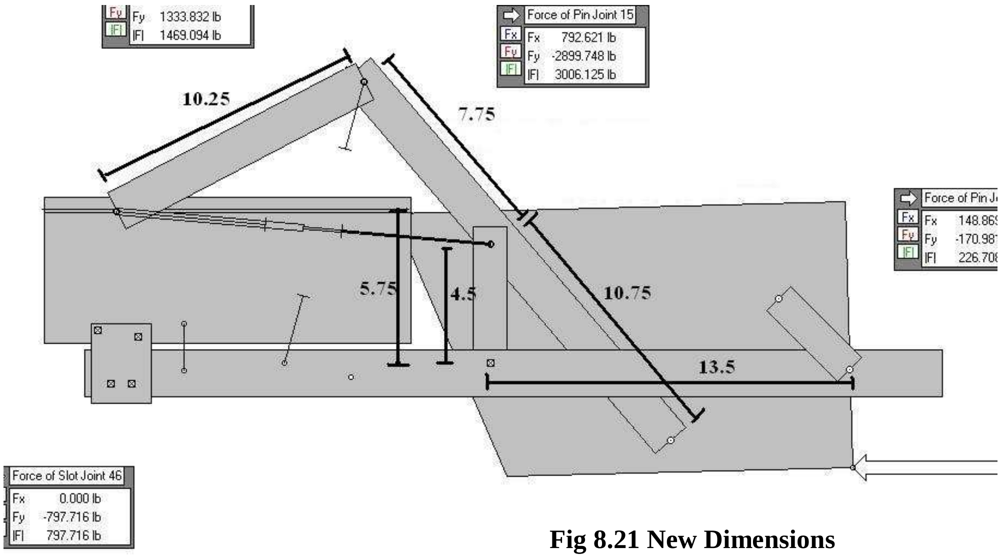


Fig 8.21 New Dimensions

The force relationships stay about the same, but the range of motion is changed slightly. The transport position (fig 8.21) is slightly reclined which actually will aid in regolith retention. The harvesting position is nearly the same (Fig 8.22).

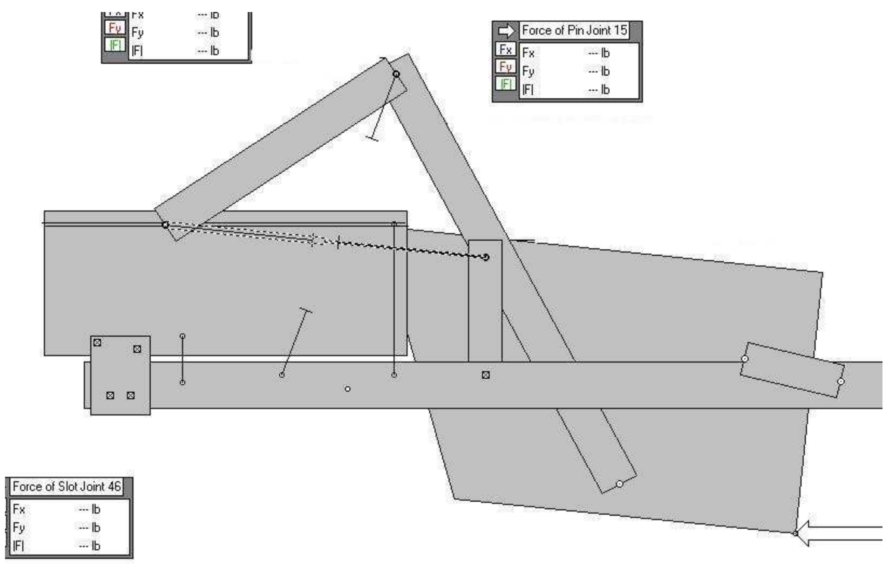


Fig 8.22 New Dimensions harvesting

The major detraction is that the dumping position only reaches to a little over 50 degrees with respect to the horizontal, coming close but not meeting the goal of 70 degrees (Fig 8.23). This is an issue to be addressed in future modifications.

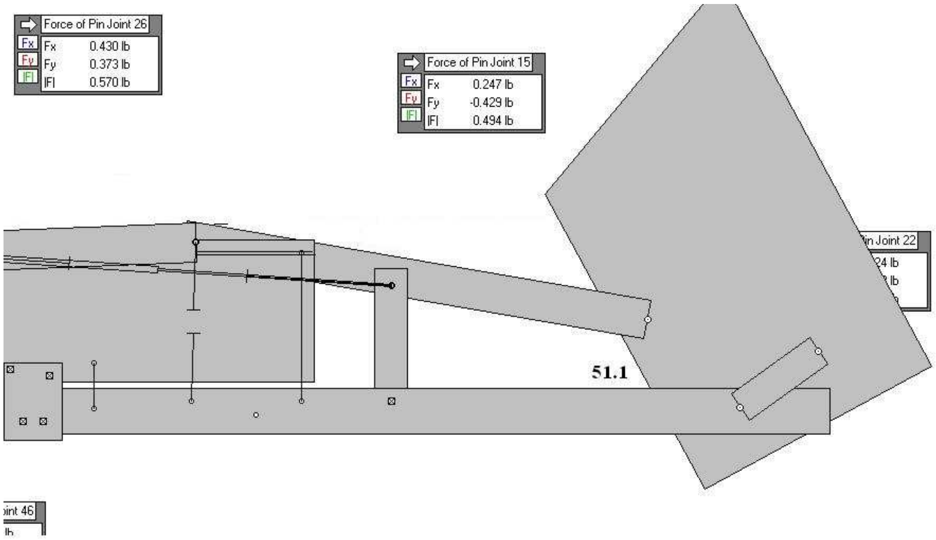


Fig 8.23 New Dimensions Dumping

<u>8.4 Slider Interface Block</u>

The requirements for the design of the slider interface block are:

1) Shall interface with the slider block and linkage interface rod and the piston of the actuator
2) Shall provide correct matching of hole sizes and locations to attach to slider block
3) Shall maintain or enhance structural rigidity/strength of linkage subsystem
4) Shall minimize weight without degrading the structural integrity
5) Shall be able to be manufactured in Design & Manufacturing Lab (DML) in house

The slider block to which the interface block must attach to has four threaded M8 holes in a rectangular pattern 57 mm long and 45 mm wide, measured from the center of each hole. The bolts purchased to us in conjunction with these holes have a 10 mm shoulder. See figure 1.

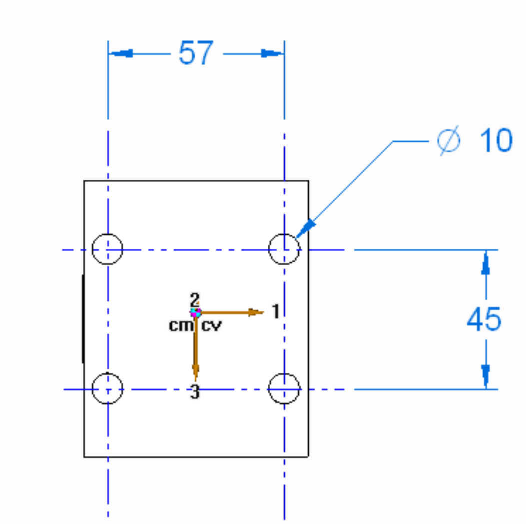


Figure 8.24 Slider Block Draft (top)

In order to properly interface with the actuator piston, measurements indicated that the center of the actuator piston must connect to the block at a height of 7/8 inches above the bottom of the block in order to maintain strictly horizontal motion for the actuator piston. A 10 mm shoulder M8 bolt identical to the four joining the interface block with the slider block will be used to join the interface block with the actuator piston. A slot shall be milled out between the bounds of the bolt holes that extend to a depth of 2.5 inches to allow for the actuator to attach at the previously mentioned bolt hole. See figures 2 and 3 .

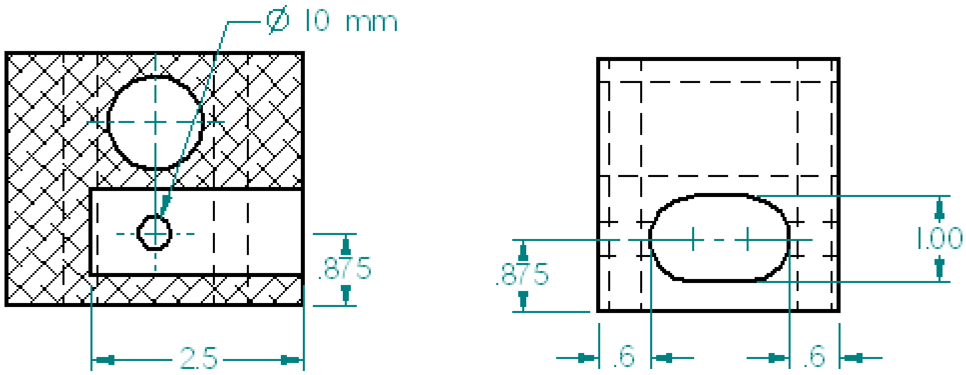


Figure 8.25 Slider Block Draft (Side and Front)

The linkage interface rod shall join to the interface block by fitting through a hole press fit with bearings. This hole shall have a 1.125 inch diameter to accommodate the bearings and the 1 inch diameter rod. The center of this hole will be positioned .8 inches from the top surface of the block to avoid interfering with the slot for the actuator piston. See figure 4.

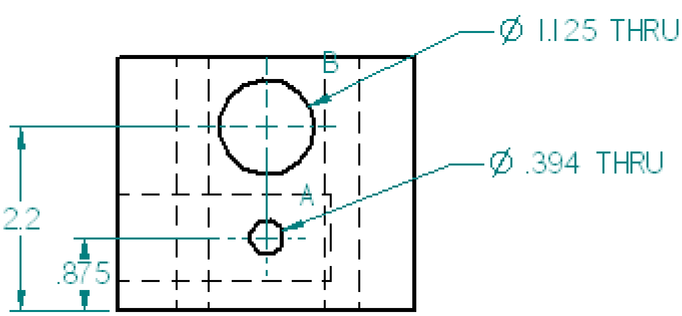


Figure 8.26 Slider Block Draft (side)

After considering the weight that could be shed by milling out unnecessary material, the benefit is so slight that it is not recommended for this prototype. For a production model more material would be milled out to lighten the payload when transporting to the lunar station.

8.5 Actuator Selection

After a primary force analysis was done, the multiple actuator devices were possible. Further analysis showed that two actuators were possible however one was more feasible.

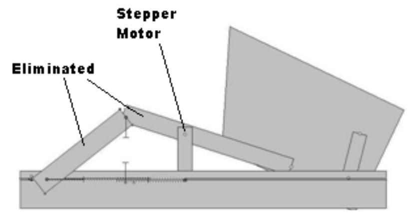


Figure 8.27: Location of stepper motor and links that would be eliminated.

The first device considered was a stepper motor located at the vertical linkage. Putting a stepper motor here would allow the design to be simplified by eliminating part of the linkage system. Looking at the basic forces this motor would endure through routine operation this idea no longer seemed possible. If this actuator was used, the force required to move the bucket into the dumping position is equal to the moment created by the bucket through the linkage at this point (see figure 7.5). This moment equates to a little more than 200 foot-pounds of torque. In order for the stepper motor to reach this torque, a gear box would be required. Assuming this load could be obtained through proper gearing, a flaw with a stepper motor arises. In order to hold position under load with a stepper motor, constant current must be applied. This would mean an increase in the total power usage.

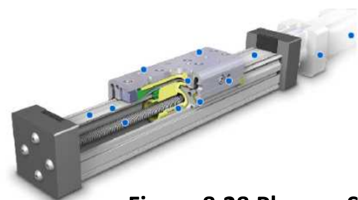
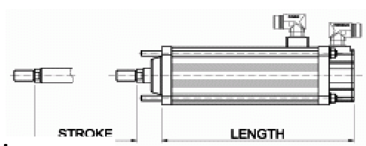



Figure 8.28 Plunger Style Actuator vs Slider Style Actuator

This power consumption by the stepper motor showed that the linkage system with a linear actuator was the best design. Using working model and the forces calculated from the force calculations, it was seen that the transverse loads were much larger than the axial loads acting on the actuator. During standard digging mode, the actuator sees a force of about 14 lb-force acting on the axial direction. However, in the transverse direction (vertical) the actuator experiences a downward force of about 51lb-force. But since the largest forces seen are from the impulse loading with an axial load of 1458lbf and a vertical load of 1900lbf, the actuator selected must be very rigid or designed such that it doesn't deflect.

Also, using working model we found that the total distance the actuator needed to travel was 15 inches. At this point two actuators seemed possible, a plunger style and a slider style, both seen in Figure 8.28 and 8.29. Since it is important to conserve space when sending this into space, the slide style actuator seemed to appeal more. It was able to deliver the same stroke length in half the room. But after a price quote on that actuator to meet our specifications, the slide style was shown to be too expensive. So we chose a plunger style actuator. With the plunger style actuator, the design needed to be able to restrict non-axial loads acting on the plunger. By incorporating a guide block from Mcmaster-Carr, we were able to get the rigidity needed to handle the large transverse loads experienced by the system. Once the transverse loads were accounted for, we decided that we needed an actuator that could handle the impulse load of 1458lbs. We found an actuator from Northern Tools, seen in Figure 7.7, which could handle 1350lbs and was only $140. Considering the dampening created by the linkage system, and the dampening at the force transfer bar, this actuator selection was acceptable for the earth prototype.


Figure 8.29 Northern Tools Electric Actuator

8.6 Bearing Selection

Introduction: The Lunar Harvester Prototype has several parts that require bearings to reduce friction when the parts are moved from one position to another. The bearings will need to be able to withstand harsh conditions of dirt, dust, and other types contaminates.

The bearing that has been chosen for the Lunar Harvester Prototype is a Daemar (DMR) Dry Slide bearing. The bearing has a steel outer shell that is lined with self-lubricating bronze. The internal bronze surface has a PTFE Teflon coating to help reduce friction. This bearing is designed for high radial loads and can perform in a harsh environment such as dirt and debris. This bearing is an off the shelf part, is inexpensive, and is readily available. The links the bearings are going to be press fit into is ½" thick. The desired length of the bearing is ¾". Simulating the Lunar Harvester in working model with a100 lb. load acting on the blade of the bucket, the maximum force seen at any of the pin joints was 306 lbs. Also, there was a simulation done with a force of 500 lbs. on the bucket (possible example of impulse loading) and with that input the maximum force seen at any of the pin connections was 1487 lbs. Below in Table 2 there is a conversion of radial pound force to psi using a bearing width of 3/8" and 1/2". This table shows that the bearing used that the bearing is more than capable of handling the loads that have been simulated (see Max. Load in Table 1).

Figure 8.30 Daemar Dry Slide Bearing

Manufacture: Daemar Bearings Inc.
Type: Dry slide Self Lubricating Bearing with PTFE Coating
Part #: 05TH06

Bearing Specs					
	(in)	Max. Load		(N/mm^2)	(psi)
Outer dia.	0.375		Static Load	250	36250
Inner dia.	0.3125		Very Slow Speed	140	20300
Width	0.375		Rotating/Oscillating	60	8700

Table 8.1 – Bearing Specs

Supplier/Distributor: Alabama Bearing Inc.

Location: Dothan, Al

Qty needed: 30 (2 per hole)

Phone: 334.793.1421

Price: $2.25 each

Delivery Time: Approximately 5 days from order date

Radial Load (lbs. to psi)					
Shaft Diameter (in)	Width (in) (1 Bearing)	Load on Bearing (lbs)	P (psi)	Width (in) (2 Bearings)	P (psi)
0.3125	0.375	100	853.3333	0.75	426.6667
		200	1706.667		853.3333
		300	2560		1280
		400	3413.333		1706.667
		500	4266.667		2133.333
		600	5120		2560
		700	5973.333		2986.667
		800	6826.667		3413.333
		900	7680		3840
		1000	8533.333		4266.667

Table 8.2 – Radial Load Conversion

Part #: 08TH16

Bearing Specs					
	(in)	Max. Load		(N/mm^2)	(psi)
Outer dia.	1.125		Static Load	250	36250
Inner dia.	1.00		Very Slow Speed	140	20300
Width	1.125		Rotating/Oscillating	60	8700

Table 8.3 – Bearing Specs

Supplier/Distributor: Daemer Bearings.
Location: Atlanta, GA
Qty needed: 25

Phone: 877.432.3627
Price: $1.00 each
Delivery Time: Approximately 2 days from order date

Other Bearings Considered: In the search for bearings, two other bearings were considered. The first one considered was a double-sealed greased ball bearing. This bearing is rated to handle high radial loads and high rpm. This bearing is not well suited for impact loading or vibration. Impact loading of this bearing could cause premature failure because it could damage the seal, and thus allow dirt and other undesirable particles into the bearing. Also, if this bearing was exposed to vibrations this could damage to the seals creating an opening for grease (lubricant) to escape and dirt and debris to enter.

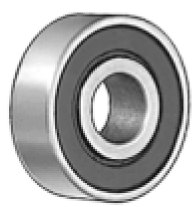


Figure 8.31 Double Sealed Greased Ball Bearing

The other bearing that was considered was a spherical bearing that had a Teflon liner attached to the inner race. This bearing was designed for extremely high radial loads and harsh conditions such as dirt and debris. This bearing was rejected because it added an undesirable degree of freedom.

Figure 8.32 Spherical Bearing

Summary: The bearing that has been selected is a readily available, off the shelf part. The bearing is inexpensive and is a good choice for this particular application.

9.0 FRAME SUBSYSTEM

9.1 Frame Subsystem Requirements and Concept Presentation

The frame subsystem was designed with the 4 derived requirements in mind:

1) Shall be able to provide rigidity/load bearing capabilities on which the bucket and mechanical linkage can fasten
2) Shall be designed to provide easy interfacing to the bucket and mechanical linkages, and accommodate the use of spacers/bearings at these interfacing locations
3) Shall be designed using 80/20 modular aluminum struts for easy interfacing and manufacturability

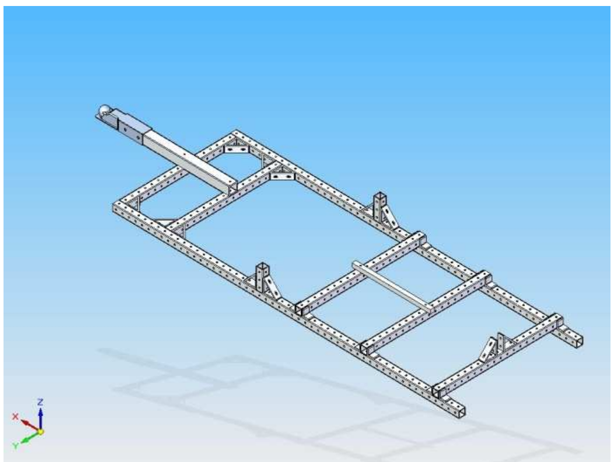


Figure 9.1. Isometric view of A-frame

The frame is constructed using 80/20 Model 9701 modular aluminum struts. It will consist of six struts, two 60" long two 25.5" long, and three 28.5" long. The 25.5" struts will be connected together with 5/16"-18 bolts, nuts, and 80/20 Model 4303 joining plates. The 28.5" struts will be connected with 5/16"-18 bolts, nuts, and 80/20 Model 4302 joining plates. These are connected on top of the frame to allow for plenty of clearance while the machine is working. This system of struts allows us to easily connect and interchange strut pieces for our frame.

Also, by purchasing the struts and not having to make them in-house, we are able to manufacture and update this design much easier more efficiently. We are unable to make this frame collapsible at this time as it will destroy the structural rigidity of the frame. The 3-hole link connecter, attached 34.5" from the front of the frame, is crucial in making the bucket perform the desired function. It is a 4.5" piece of 80/20 Model 9701 connected by two 80/20 Model 4303 joining plates. This allows us to increase stability while minimizing deformation caused by loads. The actuator will be attached in the center on the rear crossbar of the frame by 80/20 Model 4303 joining plates to allow it to extend to the back of the slider. The slider is then connected to the linkage system via a 1" steel rod, allowing the linkages to apply mechanical advantage to the system. By attaching the slider directly to the frame, it securely attaches the slider, and therefore the actuator, and holds it horizontal. In between the two rear crossbars, U-bolts are attached to connect the wheels via a ¾" aluminum axle.

An interfacing subsystem was designed to connect the frame to the Chariot rover/Gator with the 3 derived requirements in mind:

1) Shall interface with a Gator utility vehicle
2) Shall maintain horizontal orientation of harvester frame for optimal harvesting and dumping positions
3) Shall maintain or enhance structural rigidity/strength of frame subsystem
4) Shall accommodate yaw motion required for "trailing" Gator

The frame interfaces with the transportation device through the use of a 2" coupler and ball mount. It is connected to the frame by a 2" square tube that connects to the middle of the A-frame. The tube is 25" long, giving plenty of room between the transportation device and the regolith pan for turning. To attach this tube, we will use 5/16"-18 bolts that connect in the center of the two front crossbars.

From the requirements above, the concept we are presenting consists of a frame with 5 crossbars for rigidity and stability. The frame is built from 80/20 modular aluminum struts to minimize weight while keeping rigidity. Also, these struts have holes at an equal distance to allow for easy attachment of components such as actuators and connectors for linkages. The simple 80/20 connector frame attached by bolts allow for easy interfacing from the linkages to the bucket. The connector closest to the rear of the frame is used to attach the linkage that is powered by the actuator. This height and position gives us the proper rotation we desire to move the bucket to the 3 desired positions: digging, transporting, and dumping. The actuator is attached to the frame by using mounting hole on the actuator. The actuator will be bolted securely on the center of the rear crossbar to allow for the horizontal movement needed to power the system. The linkage system will be connected with the actuator with the use of a slider, helping maintain horizontal movement and preventing vertical loads from reaching the actuator. The front linkage is attached to the frame of the bucket, as the frame has holes in it for easy attachment of parts. This connection is crucial in forcing the bucket into the 3 functional positions. By attaching most of the components directly on the frame, we are able to create a more reliable system. On the rear of the frame, U-bolts are attached that will attach to the wheels. These wheels provide support to the bucket in the three positions while the axle keeps the wheels at a distance from the frame, minimizing regolith hitting the actuators, bearings, and other moving parts. At the front of the frame, a ball socket joint (similar to a trailer hitch) allows the frame to easily interface with the Chariot rover/Gator. This allows us to connect with our primary mode of transportation and be able to collect regolith, while maintaining rotational movement to allow the regolith pan to function like a trailer to the transportation device.

This frame in its design is similar to other earth pans, which allows us to observe that this frame with provide support for our bucket and linkage subsystems. Through the use of Working Model 2D, we were able to design a frame with the proper connections to allow the frame design to function properly. Using the Solid Edge drawings of the interfacing plate from previous groups, we were able to design an interfacing plate that properly interfaces with the rover.

9.2 Gator Interface

An interfacing system was designed and modeled in Solid Edge that integrates into the existing frame subsystem and connects to the earth testing rover, the John Deere Gator. It uses efficient and structurally sound means of connection and satisfies the predetermined requirements:

1) Shall interface with a Gator utility vehicle
2) Shall maintain horizontal orientation of harvester frame for optimal harvesting and dumping positions
3) Shall maintain or enhance structural rigidity/strength of frame subsystem
4) Shall accommodate yaw motion required for “trailing” Gator

When connecting a trailer to a towing vehicle, it is common to use a system known as a ball and coupler interface in which the trailer has a female spherical connector that sits on top of the ball mount on the towing vehicle (Figure 9.1).

Fig 9.2 Trailor Hitch

This type of connection, in addition to being extremely common and easy to implement, also provides the freedom of movement described by the requirements that is inherent in a ball and socket joint. This joint is also convenient in that the Gator has a receiver on the back that accommodates a 1 ¼” ball mount.

A coupler and ball mount size were decided upon that would interface well with both each other and with their respective ends. The ball is a standard 2” ball (Figure 2b) while the coupler is a standard 2” receiver (Figure 2a) that mounts to a 2” wide square tube with ½ inch bolts, all readily available for purchase.

Fig 9.2a

Fig 9.2b

On the ball side, the mount can be purchased to be one that is adjustable to match the adjustable height characteristics of the Chariot Rover interface (Figure 9.3).

Figure 9.3a Adjustable Mount

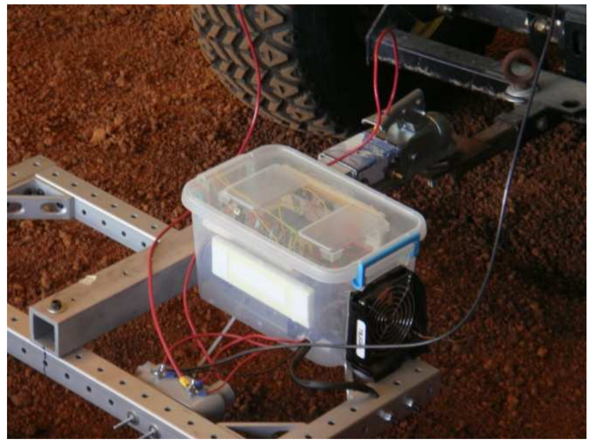


Figure 9.3b Actual Gator Interface Mount

On the Lunar Harvester side, the coupler bolts to a 2" square tube which in turn will connect to the existing frame subsystem. This is where a series of decisions had to be made as to how exactly it would attach. Welding was out of the question, so some form of bracket and bolt connection was in order. Multiple sketches were made involving cut, angled tubing and more simple straight T connections (Figure 9.4).

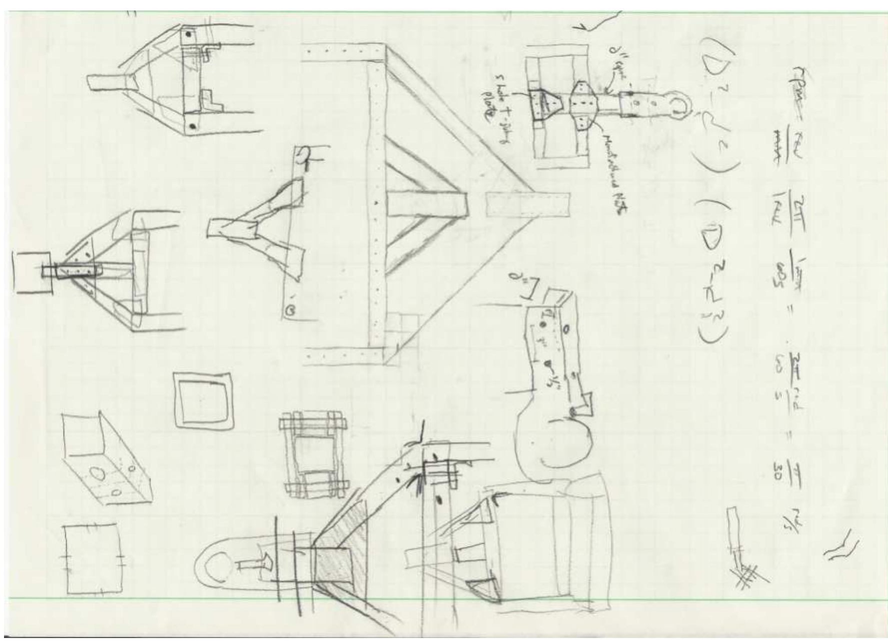


Fig 9.4 Interface Hand Sketches

Weighing the given options in terms of ease of implementation as well as structural rigidity and strength, the simple T connection was decided upon, but the tube was extended to overlap with the frame multiple times to provide more support. The tube is long enough (24") to allow clearance when the rover is turning, but not too long to be excessive. 24” is also a standard length for square tubing. The connections to the frame are at two points where the tube crosses over lateral supports of the frame (Figure 9.5).

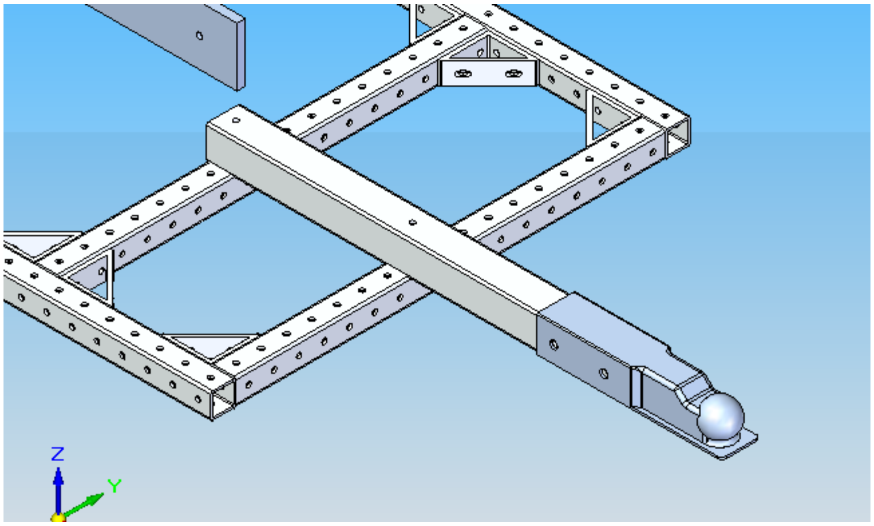


Fig 9.5 Solid Edge Interfac

This provides more area over which the towing force will be dispersed as well as better support for lateral loads when turning. The connections to the frame are made with a set of two 4.5” long 5/16 that run through the 2” tube all the way through the frame.

9.3 Chariot Rover Interface

Introduction:

Solid Edge is a graphical tool that allows an engineering design team to produce a visual prototype to determine dimensions, clearances, and other important considerations when designing a complex mechanical system. It allows for the design team to create parts to specifications, and then interface those parts with the other parts of the project, making Solid Edge a vital component of a systems engineering approach. Using this program, we are able to create a graphical representation to scale of what we will construct, which is an interface system between the chariot rover and the lunar excavator.

Results:

This interface plate was designed with 4 requirements in mind. These are as follows:

1) It shall interface with a Chariot utility vehicle’s interface plate
2) It shall achieve horizontal orientation of harvester frame at startup on a horizontal plane for optimal harvesting and dumping positions
3) It shall maintain or enhance structural rigidity/strength of frame subsystem
4) It shall accommodate yaw motion required for “trailing” Chariot Rover

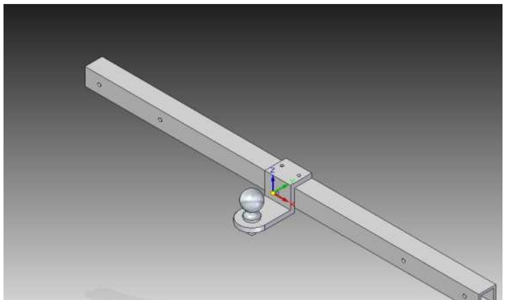


Figure 9.6 Chariot Rover Interface

From these requirements, this interface plate was constructed. The first piece is a 2” aluminum tube, 46” in length. This allows us to be structurally rigid and maintain horizontal orientation. It interfaces with the rover by attaching itself with 4 bolt holes (2 on each side) matching the bolt pattern on the rover. It connects to the ball connector by way of three 5/16-18 holes, 2 on top of the tube and one on the side. The ball connector is designed to also assist in maintaining horizontal orientation. It interfaces with the lunar excavator by having a 2” trailer ball and connecting via the trailer hitch on the front of the lunar excavator’s frame. Due to the Chariot Rover’s interfacing plate having an adjustable height, we are able to use that to adjust the interface plate to whatever height needed to maintain horizontal orientation of the frame for its intended functions.   With the trailer ball/hitch design, we are able to achieve yaw motion, allowing the rover to turn easily without stressing the frame of the lunar excavator.

Conclusions:

From this design, we have created an interface plate that meets the requirements that were created. This allows our lunar excavator to be connected to the Chariot Rover while maintaining all the functions we designed it to perform. From this drawing and draft and the purchase of a 2” trailer ball, we are able to manufacture a way to connect the excavator to the rover that completes our requirements for an interface plate.

<u>9.4 Wheel Structure</u>

A wheel structure was designed to meet the following requirements:

1) Shall provide necessary clearance from ground for harvesting and dumping positions

2) Shall maintain horizontal orientation of harvester frame for optimal harvesting and dumping positions

3) Shall maintain or enhance structural rigidity/strength of frame subsystem

4) The wheels utilized shall be able to withstand harsh environmental conditions

To provide the necessary clearance, consideration must be given primarily to the harvesting and transporting positions. There must be the right amount of clearance to hold the harvesting bucket off of the ground in transport, yet still be able to reach our desired scraping depth. Given that our maximum change in height for the harvesting edge of the bucket is around 4.5” and that our desired scraping depth is 5 cm (1.97”), an ideal clearance height was set at 2”. Mounting for the wheel axle was first considered under the frame, but after some initial calculations it became apparent that the wheels would have to be rather small to obtain our 2” clearance, which would be undesirable for bumpy terrain. So, the decision was made to mount the axle above the frame.

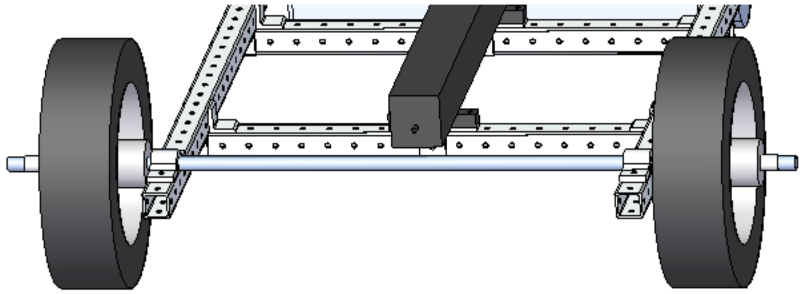


Figure 9.7 Rear View Wheel Assembly

By mounting the axle just above the frame, a desired wheel diameter of 14” could be calculated using the following equation.

$$h_{rest} + \Delta h - (r - h_{axle}) = 2$$

In this equation, h_{rest} is the distance of 3” the bucket extends below the frame in the transport position, Δh is the change in height of 4” from transport to harvesting position, r is the wheel radius, and h_{axle} is the height of the axle from the bottom of the frame, all resulting in a desired harvesting depth of 2”. By placing the axle close to the frame (h_{axle} = 2) and substituting the known values into the equation, we obtain

$$3 + 4 - r + 2 = 2$$

$$r = 7$$

Once the wheel height is set, the horizontal orientation of the harvester frame will be accomplished by the mounting to the pulling vehicle.

The structural rigidity of the frame is not compromised by this design. The axle will mount to the frame using u-bolts that go around the axle and fit into the existing holes in the frame. After assembling the excavator, the team decided to move the axle forward due to bending observed in the frame between contact points with the ground. Due to clearance issues with the slide assembly, the axle was mounted below the frame. However, the digging depth was greater than anticipated, so digging depth will still be within goals.

The wheels selected are Item# 121024 from www.northerntool.com. These wheels are designed for use on a wheelbarrow, and should be able to handle some rough terrain. These wheels will not be suitable for the lunar environment as they involve an air-filled tire, but were chosen for the prototype in the interest of cost.

Tire Type	Pneumatic
Rim Size (in.)	6
Tire Size	13.5 x 400 x 6
Diameter (in.)	3/4
Bearings Included	Yes
Hub Width (in.)	6
Load Capacity (lbs.)	300
Rim Included	Yes
Tubeless Tire	Yes
Tread Type	Ribbed
Shipping Weight (lbs)	7

Table 9.1 Wheel Specs

Another wheel considered that would not involve an air-filled tire is Model # W-1430-R-1 at www.hamiltontoncaster.com. Though extensive searching was done on the recommended TWEEL, no specifications on sizing or pricing could be found.

10.0 MANUFACTURING AND ASSEMBLY PLAN

The facility in which all manufacturing will take place is the central machine shop located on Auburn University's campus. This shop contains all machines necessary for the manufacture of the lunar excavator. This includes mills and sheet metal bending equipment, as well as hand tools and measuring equipment. Most manufacturing that takes place will be milling. In addition to milling, it will be necessary to bend sheet metal for brackets and other attachment points. Hands tools will be necessary for tapping and reaming bolt holes, and all manufacturing will have to take place under exact tolerances, necessitating the use of measuring tools such as calipers. The bucket is made out of sheet metal and will be welded. This necessitates the need for a certified welder. To insure all parts are built to correct tolerances, both the person responsible for manufacture and the person responsible for the assembly of a part will be expected to check tolerances. This will insure that all parts meet specified tolerance and quality, and lack defects.

Manufacturing Steps:

- Linkages will be purchased as close to final design as possible
- Linkage machining will be carried out using mills
- Brackets will be made out of sheet metal
- Bracket machining will also be carried out using mills
- All machining will take place in the Design and Manufacturing Lab
- All parts will be measured to insure correct tolerances before being approved
- Bucket will be manufactured out of sheet metal and welded

Assembly Steps:

- Frame will be assembled using pre-existing bolt pattern and brackets purchased from 80/20
- All brackets and attachment points will be connected to the frame
- Linkage sub-assembly will be attached to their respective mounting points
- Bucket will be attached to rotational and pivot bars
- Actuator will be mounted to frame
- Actuator and force transfer bar will be connected
- Tire axle will be attached to frame
- Tire will be attached to tire brackets
- Trailer hitch will be attached

Figure 10.1 Assembled Harvester Prototype

11.0 HARVESTER PROTOTYPE TESTING

11.1 Initial Testing

Introduction:

Testing what has been built is very important to the design process. It gives the ability to get immediate feedback as to if what has been built functions and the degree of performance to which it does function. It also easily allows for making conclusions as to what can be upgraded or improved.

Results:

Initial testing of the system showed large deflection on the force transfer bar. This deflection was measured to be about 2.5 inches; a deflection large enough to call for immediate redesign. The modification selected was to make the force transfer bar larger to decrease deflection. Although this redesign reduces the damping in the system from a shock load, it is necessary to ensure the force transfer bar does not yield.

The maximum diameter the force transfer bar can be is limited by the interface block on the slider. It can only be a maximum of 1 inch. Using this diameter, we found the deflection a 1inch bar would experience using the equations below.

$$P = \frac{\delta_1 L^3}{3EI} = \frac{\delta_2 L^3}{3EI} \qquad I = \frac{\pi r^4}{4}$$ Equation 11.1

Where δ is the deflection, L is the length of the force transfer bar, E is the Young's modulus, and I is the moment of inertia of the beam. Because the L, E and the coefficient of 3 don't change, they may be canceled out of the formula reducing it to:

$$\frac{\delta_1}{I_1} = \frac{\delta_2}{I_2}$$ Equation 11.2

Using this formula, it was found that a force transfer bar of 1inch diameter would deflect about 0.1inches. This is a reasonable deflection and is the final design that we chose.

After this initial test was done, a full test of all components was done. Our first test was to ensure that each component functioned properly. This was measured by watching the machine in motion and making sure each component moved in the way it was meant to. The linkages, actuator, and slider all moved as designed. The actuator has a tendency to slide to the right on the bolt connecting it to the slider; this is being fixed by adding washers in between the slider block and actuator to prevent the movement. Next, we found all the interfaces for possible interference. We only found one: the left side of the bucket (viewing form the front of the harvester) and the bolt head connecting the linkage system to the frame slightly rubbed against each other, preventing a smooth movement. This was fixed by sanding down the welds on the bucket, allowing for a smooth transition from transporting to dumping.

We then began to quantify the performance we have, starting with the bucket. The dumping angle was first measured. In our tests, we found the angle to be 51.1 degrees (all angles are measured relative to the horizontal frame). This allows for complete removal of the regolith. Next, the digging depth was tested. We found the digging depth to be around 1.7 – 2 inched, meeting our target of at least .5 inches. Also, the angle when digging is 7.7 degrees. Next, the transport mode angle was measured. It holds regolith at an angle of -13.1 degrees, allowing the regolith to calmly sit in the bucket without falling out the front.

After the bucket was tested, we moved to the actuator. Our main tests on the actuator were concerning the times to which it took to move to our positions. We tested the movement from the transporting stage to the digging stage and the transporting stage to the dumping stage. We felt it was only necessary to measure these as these will be the main transitions that the harvester will see. The transporting stage to the digging stage took approximately 16 seconds and the transporting stage to the dumping stage took approximately 11/5 seconds. We ran these tests 3 times to get a more precise judgment of how quickly it moves. The electrical engineers also calculated how much power the actuator was drawing, which is approximately 78 W.

The linkage system was the analyzed. First, the deflection of the new 1-inch bar was measured; it only deflected 1/8 inch. Attached are pictures of each correct position of the links. The maximum travel of the slider is 8.25 inches. There are two possible toggle positions, but the linkages will never enter them due to design choices. For the toggle position in the digging position, the slider is at the end and cannot move any further. For the toggle position in the dumping position, the 4-hole gussets chosen prevent the bucket from moving any further and keep the linkages from becoming parallel.

Test	Run		
	1	2	3
dumping angle (degrees)	51.1	N/A	N/A
digging angle (degrees)	7.7	N/A	N/A
transporting angle (degrees)	-13.1	N/A	N/A
digging depth (inch)	1.7	N/A	N/A
transport to digging time (seconds)	15.5	16.1	15.8
transport to dumping time (seconds)	11.4	11.6	11.1
power drawn (watts)	78	N/A	N/A
deflection of bar (inch)	0.125	N/A	N/A
angle between links in digging (degrees)	86.5	N/A	N/A
angle between links in transport (degrees)	110.9	N/A	N/A
angle between links in dumping (degrees)	168.7	N/A	N/A
maximum travel (inch)	8.25	N/A	N/A

Table 11.1 Initial Testing Results

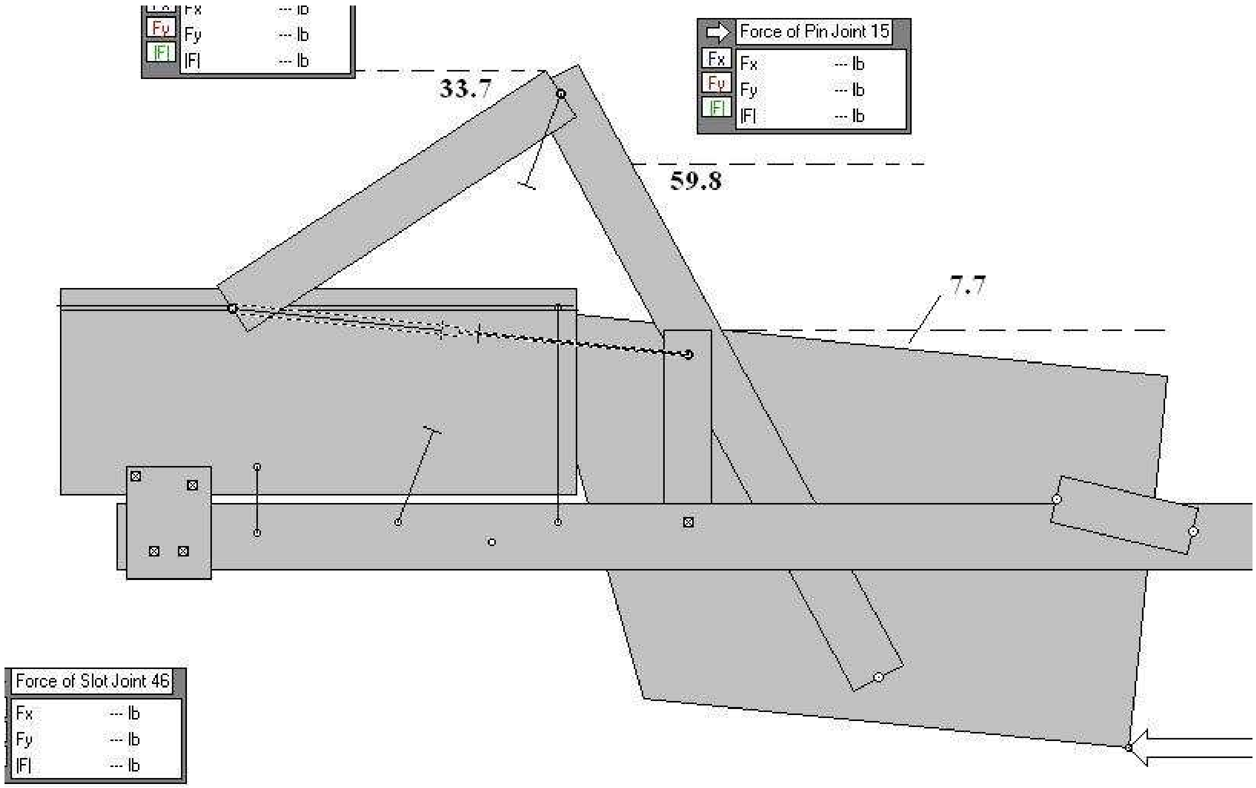


Figure 11.1 Maximum Digging Force

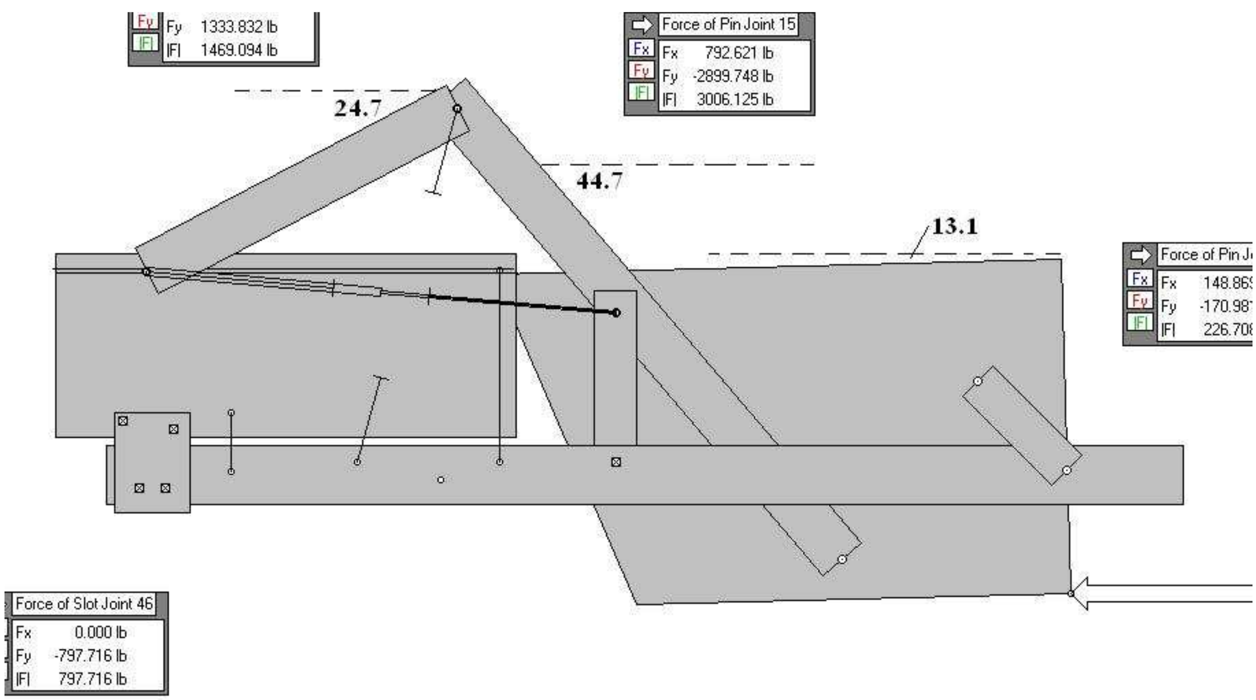


Figure 11.2 Shallow Digging Force

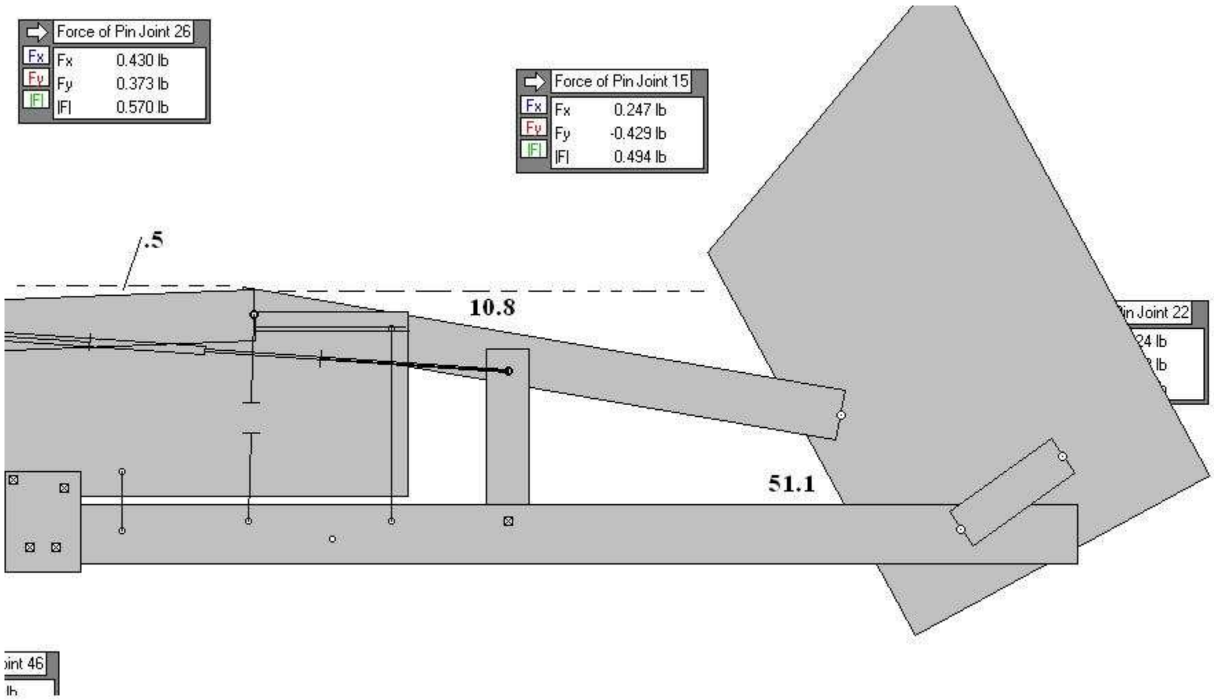


Figure 11.3 Dumping Forces

Conclusions:

From this testing, we have discovered that our prototype functions with an empty bucket. While this is not as helpful as testing with a regolith substitute, it is necessary to ensure that the prototype works without a substance in the bucket before we attempt to dig and transport with it. Since we know the prototype functions, we are no able to move to Phase 2, which is test with a regolith substitute.

11.2 Final Testing

Testing of the design as outlined in the MPCOD is to verify the effectiveness of the design and obtain objective values on its performance. Validating the correct mechanical operation of the system was paramount. The testing is also intended to determine whether or not a blade is necessary for the collection of lunar soil. These tests were conducted at the USDA soil laboratory in rocky, loose topsoil.

Testing Methodology:

The lunar excavator was attached to a gator utility vehicle with a standard trailer hitch. The actuator was remotely controlled from the ground station which was in visual distance of the excavator. The camera was not ready for integration with the mechanical portion of the design and was therefore left off in favor of visual inspection. The excavator was lowered into the dumping position by the ground station. The gator was then driven no more than 50 feet due to space considerations in the testing area. At the end of the run, the bucket was raised into the transport position and driven in reverse to the starting location. In the first round of testing, the soil was dumped to test the dumping ability of the bucket. When that was confirmed, the soil was instead emptied into a container and weighed to determine the production rate of the harvester. All runs were timed. Finally, the density of the soil was measured.

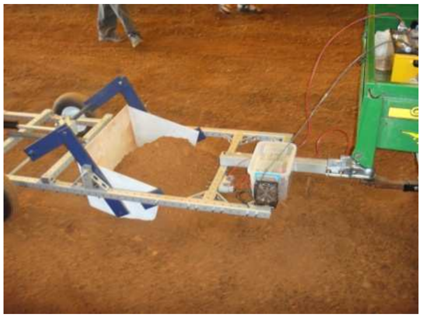


Figure 11.4 Digging Testing

Figure 11.5 Dumping Testing

Figure 11.6 Density Testing

The granular size of the dirt was fairly large in comparison to normal dirt and proposed size of moon regolith. However, the granular size of the dirt gave it the "loose" quality that we would expect regolith to be, and thus was a valid substitute for regolith. The bulk density of the soil was determined to be 1.9 g/cm^3. This was found using the core-cutter method. The mass of the soil gathered with a coring probe in a known-volume cylinder was weighed, thus allowing the density to be calculated. This density is close to the density of regolith (10.5% error), thus rendering the test valid.

Figure 11.7 Dirt Granular Size

Results:

RUN 1

No mechanical problems were in evidence. The dirt extracted exactly as planned, spilling over itself continuously into the back of the bucket. This ensured that the maximum amount of soil was collected. The bucket filled to approximately 1/3rd of the height. Slight flexing of the frame was observed, no causation of performance issues resulted.

Time: 1 minute 34 seconds
Weight of soil extracted: 42 lbs
Production Rate: 1241 kg/hr of Regolith

RUN 2

No mechanical problems were observed. Angle of attack (AoA) for bucket best determined to be between 5° and 7°. Traction was maintained by Gator until the AoA exceeded 7°.

Time: 1:22
Weight: 59.08 lbs of soil
Production rate: 1490 kg/hr

RUN 3

Time: 1:35
Weight: 77 lbs
Production rate: 1942 kg/hr

RUN 4

Run 4 was used to determine the effectiveness of dumping of the bucket. The prototype was operated several times through the complete concept of operations and then the collected soil was dumped out of the bucket. The soil restricted the complete evacuation of the bucket, however if a hopper was in place beneath the bucket, the soil would not completely dumped out of the bucket.

Conclusions:

The design met or exceeded all of its mechanical design requirements. The amount of soil collected far exceeds the system requirements as outlined in the MPCOD. It was possible to control the excavator from a remote ground station and raise and lower the bucket which was used as a digging implement.

Figure 11.8 Testing Group at USDA Facility

12.0 FUTURE CONSIDERATIONS FOR LUNAR CONDITIONS & EARTH PROTOTYPES

This particular design is for use with earth testing only. The lunar environment is quite harsh. A number of considerations must be considered to make the design capable of surviving in such conditions. Radiation, temperature swings and micrometeorites are some of the considerations.

In addition to the harsh lunar environment, the design must be optimized for the flight to the moon. Weight and size will be of primary concern here.

Radiation:

Due to the lack of an atmosphere, a large amount of radiation will reach the lunar surface. Some of the frequencies in this radiation are capable of degrading polymers such as plastic. Therefore it will be necessary to either select plastics that will not degrade due to the radiation, or not use any polymers in the design of the excavator.

The solar wind, in addition to providing the materials in the soil that this excavator will harvest, is also a constant low energy stream of particles that can cause charge to build on the excavator causing an electrical discharge. To prevent this, the vehicle will have to be grounded. This can be achieved by making sure the excavator is not insulated from the chariot rover.

Solar cosmic rays are lethal to both people and electronic equipment. An early warning system to detect these rays would have to be installed. Upon receiving a message warning of a solar event, the rover and excavator will have to be moved to a radiation protected area. It is important that the excavator reach this shelter in time, because the solar flare will interrupt radio communications.

Of primary concern is radiation damage to the electronic components of the excavator. Next to biological matter, electronics suffer the most adverse effects of radiation. To prevent this, all electrical components must be shielded and rated to survive the amount of radiation expected.

Temperature:

The surface temperatures of the moon are quite extreme. At the equator, temperature swings of 280 K are not uncommon. At the poles, where a lunar base will be located, the highs are not as high, but the lows are lower. This leads to a problem when part of the vehicle is in shadow and the other is in direct sunlight. A high thermal stress will develop due to the temperature difference, possibly leading to deformation of the material. When selecting a material, the designer must be considerate of thermal expansion qualities. Brittle fracture due to micrometeorite impact is also a concern.

Regolith:

Harvesting regolith is the sole reason for this excavator's existence, but it also presents an engineering challenge. Regolith is capable of infiltrating the joints of any of the components. This is especially a concern on moving parts such as the linkage bearings. To prevent these from jamming, all bearings must be sealed against infiltrates. This will take care of most problems associated with regolith. Also, a sealing bellows will need to be added to the actuator to keep dust out of it.

Weight and Size:

Weight will be a primary concern due to the cost of putting objects into earth and lunar orbit. This can be ameliorated by selecting materials like high strength aluminum or titanium that have high tensile strengths and low densities. This is imperative to maintain the structural integrity of the vehicle, and keeping weight within reasonable limits. The size of the vehicle is entirely determined by the necessary amount of regolith to be collected. If the number is changed from 50 kg/hr, the design can easily be scaled up or down. The size of the harvester can also be reduced by changing the current actuator-slider combination to a slider actuator. This combines both processes into one system and would reduce the length of the harvester by 1.5 feet.

Future Considerations:

In addition to these things needing to be changed for lunar applications, there are a number of systems that should be added to increase the user-friendliness of the system. A load sensor should be added to the tongue of the frame as a means of impact reduction. When the load sensor reads a spike larger than 1350 lb, a microcontroller immediately shuts the Chariot rover from forward motion to minimize damage done to the actuator. Also, the frame needs to either be constructed from a less flexible material or needs to be reinforced with plates to discourage bending. This can be done by adding thin aluminum plates to the outside of the 78 inch frame sections. Also, the bucket can be modified to prevent regolith from spilling out of the corners. Side guards can be added slow the amount of regolith expelled from the sides.

13.0 CONCLUSIONS

The regolith pan is a complete redesign of the previous senior design group's lunar harvester. The goal is to collect 50 kilograms of regolith per hour for hydrogen reduction, and from our analysis, we ultimately decided the product could be done more efficiently with a new design as opposed to the old design with or without voice coils.

This regolith pan is designed to overcome the problems of the older models while keeping similar design requirements. The regolith pan is designed to complete all of the requirements of the previous designs while doing it faster and more efficiently. These design specifications are as follows:

1) Shall be designed to conduct studies on earth but be able to operate in a Lunar environment
2) Shall interface with Gator utility vehicle
3) Shall be operated remotely
4) Shall collect and hold at least 50 kg soil per hour

With the new design, we will be able to more accurately conduct regolith harvesting studies on earth and, ultimately, the moon. As with all previous designs, it will interface with the KSC interfacing plate as this is how the regolith pan will connect to the chariot rover. The pan will now be controlled remotely from a ground station, allowing for a person on earth to operate the machine without being in a lunar environment. When in use, the bucket will collect at least the required 50 kilograms per hour for the hydrogen reduction process. All parts are selected to work effectively and reliably in a lunar environment.

Ultimately, we chose the regolith pan redesign over the previous design for several design considerations. First, it is simpler than the previous process. By combining the digging and storing concepts into one solution, we are able to minimize weight and power as compared to the previous design. From our analysis, we discovered the effectiveness of the vibratory bit was inconclusive at best and thus decided to eliminate it to also minimize weight and power.

From all of this analysis, we have developed a manufacturing plan for our design to be able to assemble the regolith pan. From the parts we have chosen, we are able to edit the design as needed to maximize efficiency and correct problems we may encounter when constructing the regolith pan.

APPENDIX A - REFERENCES

Willman, Brian M., and Walter W. Boles. "Soil-Tool Interaction Theories." Journal of Aerospace Engineering 8 (1993).

"McMaster-Carr Raw Materials, Fastening and Sealing, and Machining and Clamping." 2008 McMaster-Carr Supply Company 27 Oct 2008 <http://www.mcmastercarr.com>.

"MetalsDepot.com" 2008 Ledford Steel Company 23 Nov 2008 <http://www.metalsdepot.com>.

"Tolomatic – Excellence in Motion" 2008 Tolomatic, Inc 23 Nov 2008 <http://www.tolomatic.com/>

"Northern Tool + Equipment" 1996 Northern Tool + Equipment, Inc 23 Nov 2008 <http://northerntool.com>

APPENDIX B – BILL OF MATERIALS

Item #	Name	Part	Quantity	Material	Part #	Cost	Supplier
	-	**Fasteners**				$282.98	
	-	Nuts				$14.72	
1	-	5/16"-18	1 Box (100 count)		6676T132	$3.78	MCC
2	-	1/4"-20	2 Boxes (25 count)		90630A110	$9.78	MCC
3	-	8mm	1			$0.58	AH
4	-	6mm	1			$0.58	AH
	-	Bolts				$70.75	
5	-	1/4"-20 Thread - 5/16"Shoulder 3.5"long	16		91264A553	$24.80	MCC
6	-	1/4"-20 1" Long	1 Box (100 count)		91251A542	$14.72	MCC
7		1/4"-20 2" Long	6			$3.50	AH
8		1/4"-20 3" Long	2			$0.58	AH
9	-	M10-6mm shoulder 60mm long	5			$8.00	AH
10	-	M6x80mm	2			$2.50	AH
11	-	1/4"-1.5"center-center U-bolt	2			$2.75	AH
12	-	5/16"-18 2"	1 Box (100 count)			$8.90	MCC
13	-	1/4"-20 1/2"dia 3"long	2			$5.00	MCC
	-	Washers				$19.43	
14	-	5/16"Screw Size, 11/32"Inner Dia, 11/16 Outer Dia	1 Box (50 count)	Zinc	98023A030	$8.40	MCC
15	-	1/4" Washer	1Bag (100 count)			$6.43	AH
16	-	10mm Washer	15			$2.15	AH
17	-	6mm washer	2			$1.25	AH
18	-	1/2" Inner Dia	2			$1.20	AH
19	-	Codder Pins	1 Bag (25 count)			$5.25	AH
	-	Bearings				$78.08	
20	-	1" Self Lubricating Bushing	1	Steel/Bronze/Teflon		$10.58	ABI
21	-	5/16" Self Lubricating Bushing	30	Steel/Bronze/Teflon	05TH06	$67.50	ABI
	-	Brackets				$100.00	
22	-	80/20 2 Hole Bracket	16		4302	$47.20	BA
23	-	80/20 4 Hole Bracket	12		4303	$52.80	BA
	-	**Materials**				$820.19	
24	-	Frame Rail - 72" 80/20	3		9701	$53.65	BA
25	Gator Interface Beam	2"x24" Sq Tubing 1/4"Thick	1	Al 6063 T52		$28.00	OM
26	Bucket Stock	24"x24"x1/2" 1018 Steel Sheet	2	1018 Steel	1388K181	$462.24	MCC
27	Actuator/slider Interface	3"x3"x12" Aluminum Stock	1	Al 6061	SQ33	$81.00	MD
28	Wheel Axle	3/4" Dia x 6' Long 4140/4142 Alloy Steel	1	4140/4142 Alloy Steel	8935K39	$29.97	MCC
29	Linkage Stock	4"x3'x1/2" 4140/4242 Alloy Steel	2	4140/4242 Alloy Steel	6554K323	$147.12	MCC
30	Force Transfer Bar	1"dia x 3' 1L40 Alloy Steel Rod	1	41L40 Alloy Steel	6776T132	$18.21	
	-	**Tools**				$440.33	
31	-	Solid-Carbide Fractional Chucking Reamer 3/8" Dia (.3750"), 3-1/2" L O'all, .3745" Shank Dia	2		3026A37	$122.58	MCC
32	-	Jobbers Twist Drill Bit Black-Oxide, 3/8" Size, 5" L Overall, 3-5/8" L Flute	8		2931A34	$27.52	MCC
33	-	Solid-Carbide 2 in 1 Countersink Trade Size 2, 5/64" Drill Size, 3/16" Body Diameter	1		2925A54	$19.55	MCC
34	-	Bright Finish High-Speed Stl Spiral Point Tap 1/4"-20, H3 Pitch Diameter, 2 Flute	1		2523A411	$5.05	MCC
35	-	Cold Saw Blade (180)	1		4190A61	$92.97	MCC
36	-	Vert. Band Saw Blade	1		4179A316	$43.26	MCC
37	-	Horz. Band Saw Blade	1		4179A979	$55.05	MCC
38	-	5/8" 2-Flute End Mill	4		2782A77	$22.05	MCC
39	-	1/2" 4-Flute End Mill	1		8912A45	$26.12	MCC
40	-	1/4" 2-Flute End Mill	1		8909A41	$26.18	MCC
	-	**Parts**				$230.83	
41	-	2" Ball Hitch	1	Zinc	63831	$7.45	
42	-	2" Quick Release Ball Hitch Mount	1		21128	$15.90	AAC
43	-	Blue 16oz Layout Fluid	1			$7.61	
44	-	Wheelbarrow Assembly, 3/4in. Bore — 13.5 x 400 x 6	2	121024		$49.88	
45	-	11 13/16 Stroke 1350lb 12volt Actuator	1		125012	$149.99	NTE
					Total Cost	**$1,774.33**	
		MCC - Mc Master Carr					
		BA - Bertelkamp Automation					
		AAC - Automotive Accessories Connection					
		OM - OnlineMetals.com					
		TI - Tolomatic, Inc.					
		TPS - Trailor Parts Superstore					
		NTE - Northern Tool + Equipment					
		MD - MetalsDepot.com					
		ABI-Alabama Bearing Inc.					
		AH - Auburn Hardware					

APPENDIX C – MPCOD

Manager's Project Contract of Deliverables

NASA - Corporation 4
Jack Becker, Joe Bryant, Alan Gaskins, Bryant Hains, JD Jenkins,
Luke Weniger and Phillip Young
January 29, 2009

- The prototype shall be designed to conduct studies on earth – Observations about adapting to space environment shall be included during testing phase.
- The prototype frame shall be constructed using 80/20 modular aluminum struts and brackets for easy interfacing and manufacturability wherever applicable – prototype frame shall be able to be adjusted to provide different horizontal orientations.
- The prototype shall interface with Gator utility vehicle – the interface between prototype and the Gator shall have horizontal and vertical rotational movement (spherical joint) to accommodate a turning radius and a raising radius (yaw motion required for "trailing" Gator). A solution for interfacing with the chariot rover interface plate shall be designed and ready to manufacture.
- The prototype shall collect and hold at least 50 kg soil per hour – This performance goal will be measured visually while attached to Gator utility vehicle. The camera attached to prototype shall also be used to check performance away from immediate vicinity. The prototype shall be able to accommodate a cutting blade mounted on the front edge of the bucket, but shall go without blade for prototype testing. If wear is excessive, or bladeless prototype is ineffective, a blade will be attached and testing resumed.
- The prototype linkages shall be able to move bucket to and support at three desired mechanical positions – Shall provide rigidity to digging, shall remove all pseudo-regolith from bucket during dumping, and shall keep pseudo-regolith from spilling during transport.
- The prototype linkages shall allow variable digging depth that includes the range of 1-5 cm.
- The prototype shall integrate Electrical Engineering subsystems into the mechanical design. Final prototype shall be able to accommodate all electrical systems after final assembly and physical testing is completed. The prototype shall utilize linear actuator from first generation Lunar Excavator.
- During testing, improvements shall be made to design to increase and maximize production rate. The most efficient prototype design will be determined.

Proposed Schedule:

Stage 1: Initial Manufacture, Assembly, and Testing

Manufacture:

1. Bucket side/back/bottom
2. Linkages
3. Frame Cuts

Assembly:

1. Frame
2. Linkages (Press fits, Linkage Subsystem)

Testing:

1. Rigidity of Frame
2. Rotation and Clearance of Linkages

Other:

1. Send Bucket off to be welded
2. Purchase all fasteners and miscellaneous supplies (including sod blade)

Stage 2: Secondary Manufacture, Assembly, and Testing

Manufacture:

1. Linkage Interfacing Rod
2. Axle Rod

Assembly:

1. Frame and linkages, Frame and actuator
2. Wheel Assembly

Testing:

Functionality of all assemblies

Stage 3: Final Manufacture, Assembly, and Testing

Manufacture:

Gator Interfacing Tube

Assembly:

1. Frame and Bucket Assembly
2. Frame and Gator Interfacing Tube
3. Frame and Electrical Systems (camera, WiPort, etc)

Testing:

1. Manual actuation of mechanical components
2. Motorized actuation of mechanical components

Final Testing:

Full testing at USDA with Gator Utility Vehicle, including Electrical Systems