LUNAR HARVESTER

NASA - Corporation 4

“CRITICAL DESIGN REVIEW”

Fall 2008

December 5, 2008

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Project Manager – Phillip Young

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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 prohibitive 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.

After the preliminary design review was completed, work was begun on the steps necessary for a critical design review (CDR). The purpose of the CDR is to ensure the design is complete before moving into the fabrication and testing stage of the design process. The cost of correcting any design flaw will be magnified greatly in the fabrication phase so it is vital to catch all design errors before fabrication begins. To achieve this, the design will undergo FEA modeling of the critical links to ensure the proper function of the design. All constraints not already specified such as bearing and actuator sizes will be designed. Once this is complete, the solid edge drawings will be finalized. This will allow for a complete set of correctly dimensioned engineering drawings to be created. These drawings, which will be part of the CDR, will be used in the next phase of the design process.

The bearing selected was a Dry slide self-lubricating bearing with PTFE coating produced by Daemar Bearings Incorporated. A sliding linear actuator was selected due to its ability to withstand more forces than conventional actuators. This actuator is a mxe32s model produced by Tolo-o-matic. FEA revealed no problems with yielding were present. All solid edge drawings are complete and correctly dimensioned.

3.0 TABLE OF CONTENTS

3.1 LIST OF FIGURES

3.2 LIST OF TABLES

3.3 LIST OF EQUATIONS

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.

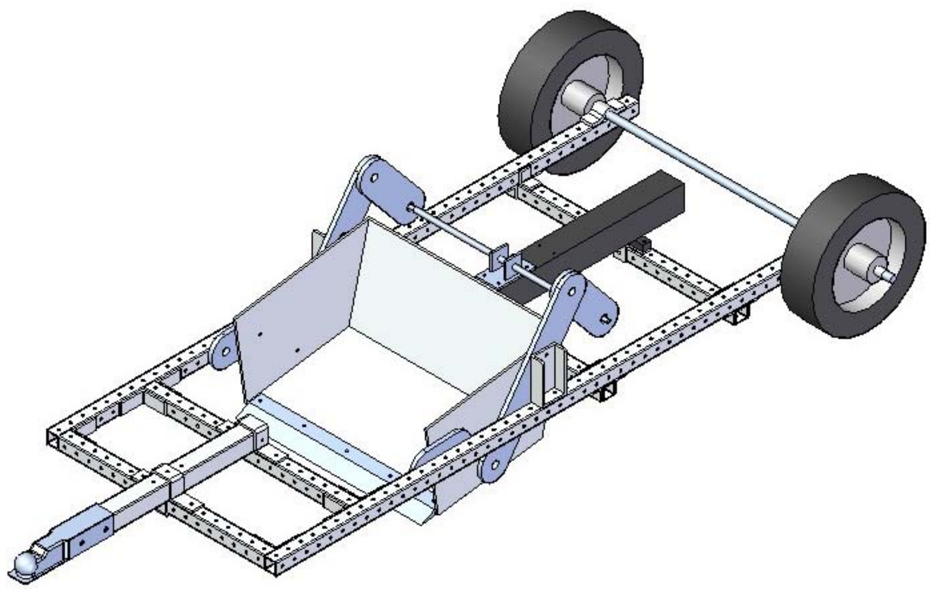


Fig 4.1 Harvester Isometric

5.0 PROJECT MANAGEMENT

The Project Manager of the Lunar Harvester design is responsible for interfacing 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):

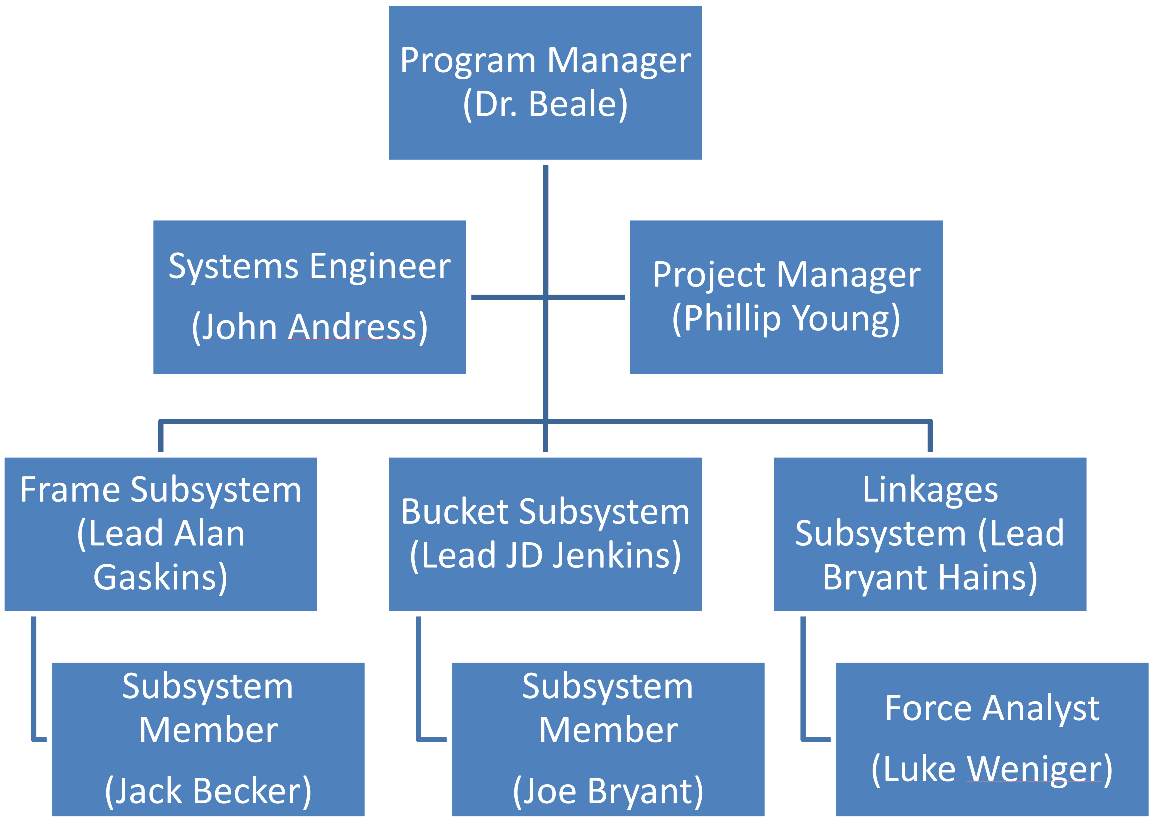


Figure 5.1 Management Breakdown Structure

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):

NASA Lunar Harvester
Corporation 4

Project Lead: Phillip Young

Today's Date: 12/5/2008

Start Date: 8/18/2008 (Mon)

WBS	Tasks	Task Lead	Start	End	Duration (Days)	% Complete	Working Days	Days Complete	Days Remaining
1	Frame Subsystem	Alan	10/02/08	12/15/08	75	100%	53	75	0
1.1	Frame		10/02/08	10/21/08	20	100%	14	20	0
1.2	Wheel Structure		10/15/08	10/21/08	7	100%	5	7	0
1.3	Subsystem Interface		10/20/08	11/08/08	20	100%	15	20	0
1.4	Gator Interface		11/09/08	11/28/08	20	100%	15	20	0
2	Bucket Subsystem	John-David	10/02/08	12/15/08	75	75%	53	56	19
2.1	Bucket		10/02/08	10/18/08	17	100%	12	17	0
2.2	Blade		10/02/08	10/18/08	17	100%	12	17	0
2.3	Front Door		10/02/08	11/09/08	39	0%	27	0	39
2.4	Subsystem Interface		10/02/08	10/29/08	28	100%	20	28	0
3	Linkage Subsystem	Bryant	10/02/08	12/15/08	75	98%	53	73	2
3.1	Mechanical Links		10/02/08	10/18/08	17	100%	12	17	0
3.2	Actuator		10/02/08	10/18/08	17	90%	12	15	2
3.3	Joints		10/02/08	11/07/08	37	100%	27	37	0
3.4	Subsystem Interface		10/02/08	10/30/08	29	100%	21	29	0

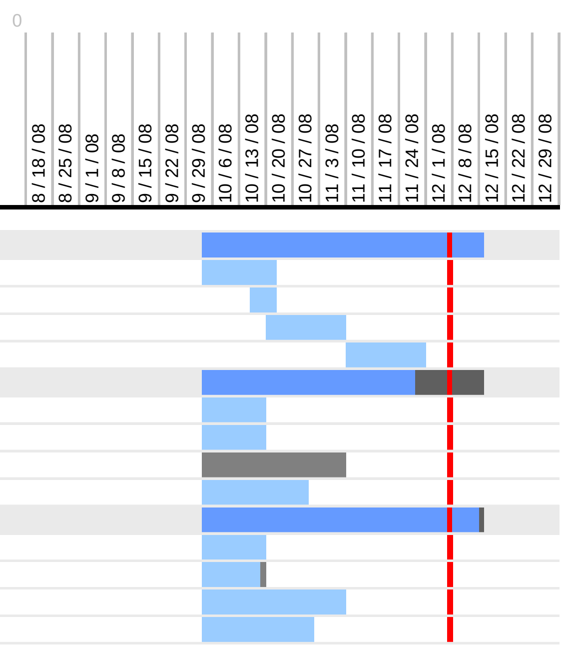


Figure 5.2 Gantt Task Chart

At this point, all tasks have been completed for the Critical Design Review. The Front Door concept for the bucket subsystem was terminated in November. The linear actuator is currently 95% complete, with only final specifications from the manufacture withholding.

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 first semester Milestone and Phases schedule can be viewed in the following figure:

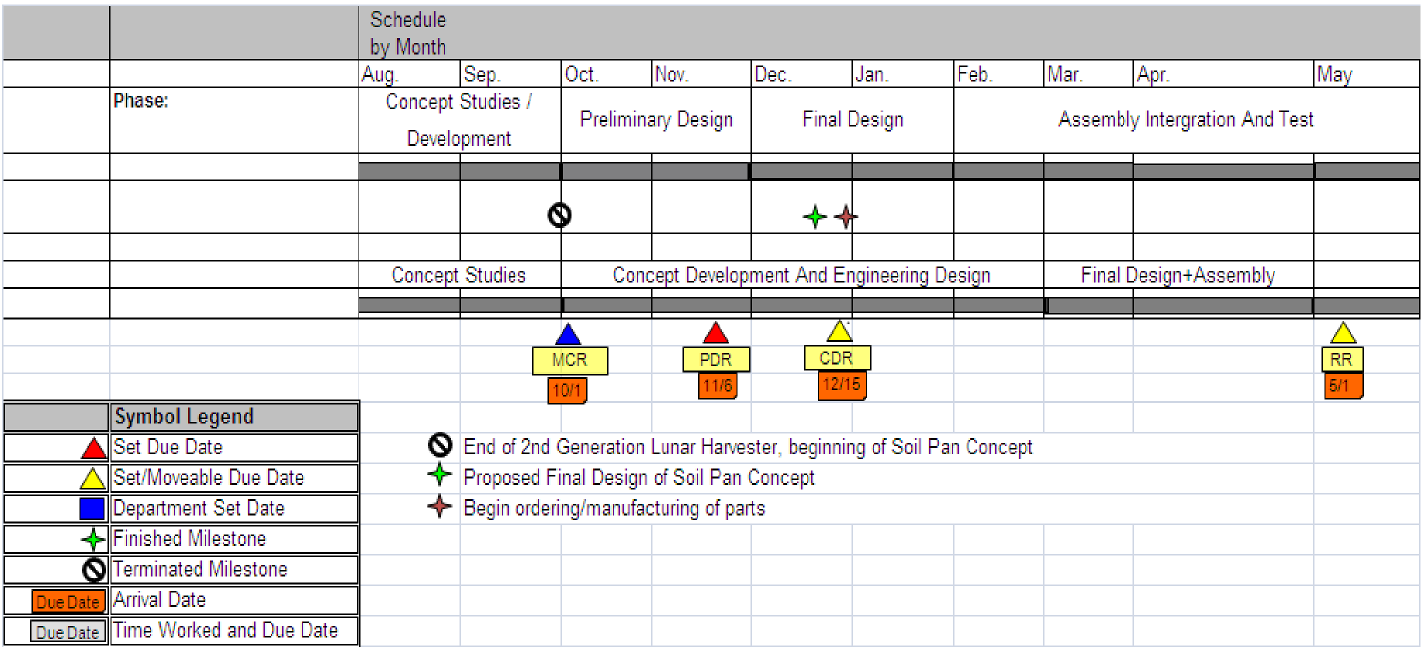


Figure 5.3 Milestone and Phases Schedule

6.0 Systems Engineering

6.1 Introduction into Systems Engineering

Corporation 4 is in the Preliminary Design portion of the Systems Engineering approach, also known as Phase B. In this phase, the goal was to define the project in enough detail at all levels so that there are no unresolved technology issues. The proposed designs have been narrowed down to one selection, the soil pan, and this report will show that the correct design has been selected. More important aspects in Phase B of the Systems Engineering approach are identifying interfaces between the subsystems and having a future plan of verification.

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 can be 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 Lunar Harvester 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 Lunar Harvester 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 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 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 B, the Preliminary Design, the architecture includes all foreseen named components. This structure will be referred to and updated in future phases to include manufacturing methods as well as interfacing with other components. This architecture serves as a starting point to understanding the outline of the design concept

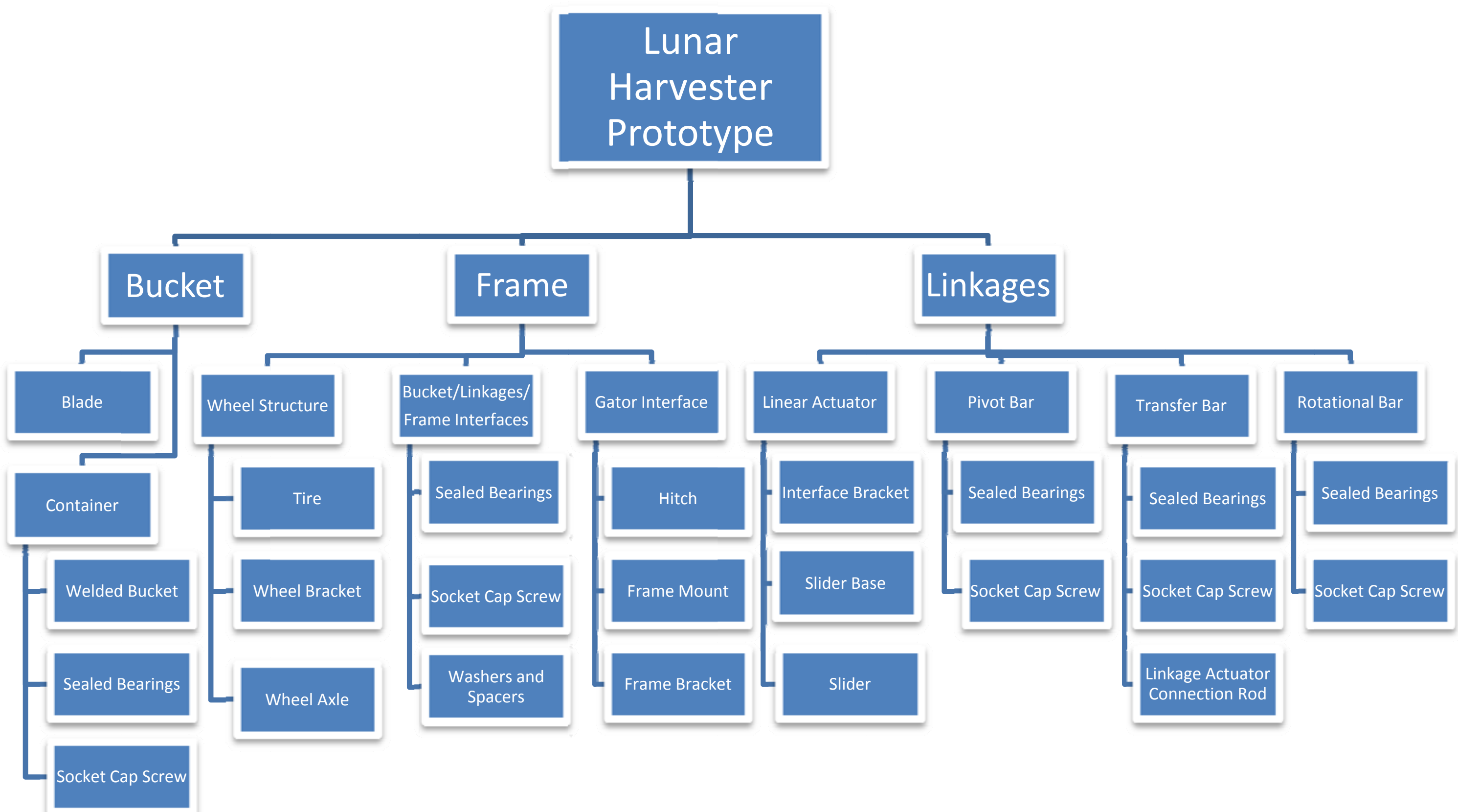


Figure 6.2 System Block Diagram

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 is 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 will be utilized to predict performance. A current and future plan for verification is as follows:

Phase C Verification:

- Determine forces acting on individual links.
- Verify linkage/frame construction by performing FEA using ALGOR.
- Verify linkage design by using synthesis equations to find most efficient lengths.

Plan for Phase D Verification:

- Assemble linkages separately from system to test effectiveness.
- Assemble total system for manual proof of concept testing. This could consist of manual movement of bucket positions, manual pushing of bucket through pseudo-regolith.
- Test for environmental conditioning – compare “loose” tolerances versus “tight” tolerances.
- Conduct proof of concept testing at USDA facility using all components and interfacing to the Gator vehicle.

6.6 Interfaces

Interfaces must be developed in 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. Functional and performance interface requirements for the Lunar Harvester design are:

1. Interface between harvester system and chariot rover interface plate shall have horizontal and vertical rotational movement (spherical joint) to accommodate a turning radius and a raising radius (Functional)
2. Interface between bucket subsystem and frame subsystem shall be constrained to 1 DOF by revolute joint (Performance)
3. Interface between linkage components shall be constrained to 1 DOF by revolute joint (Performance)
4. Interface between actuator and frame shall be defined by 2 points and constrained to vertical motion only (Performance)
5. Interfaces shall be designed to accommodate lunar environmental conditions (Functional)

7.0 Bucket Subsystem

7.1 Bucket Subsystem Specifications and Constraints 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)

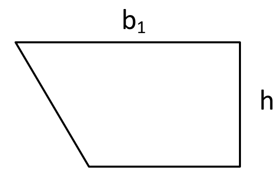


Fig. 7.1 Bucket Side

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)

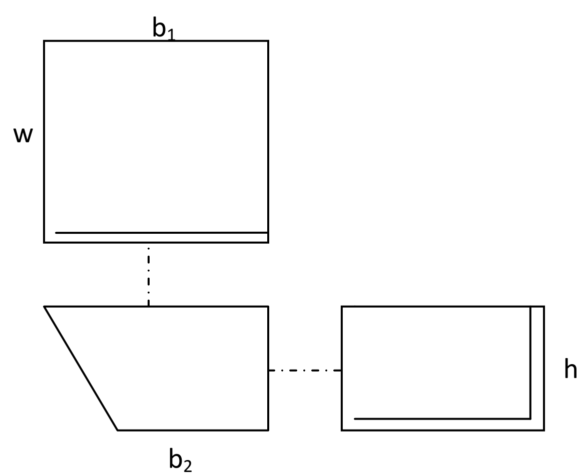
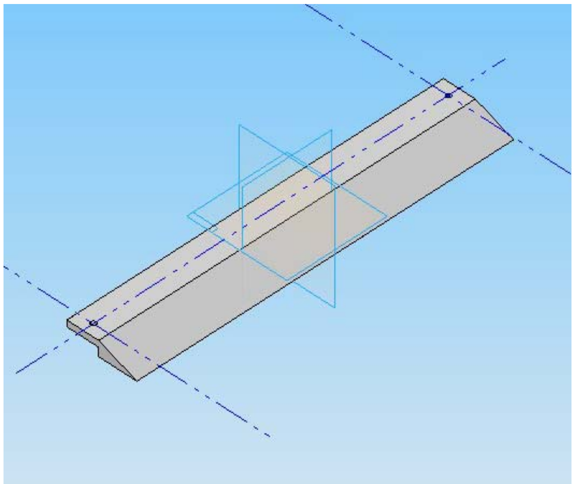
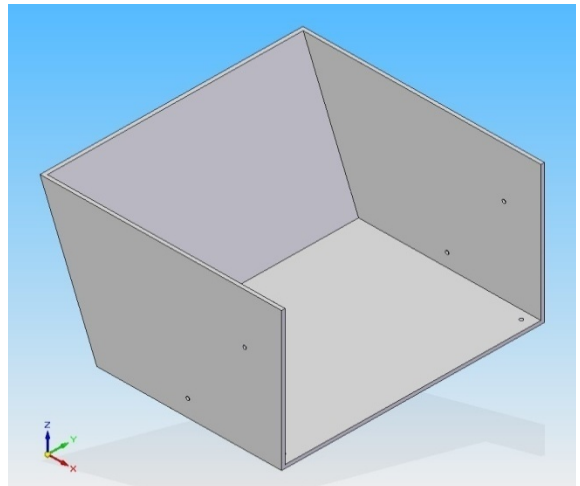


Fig. 7.2 Bucket 3rd Angle

When considering manufacturability, it was determined that this shape can be manufactured by using sheets of aluminum for the three walls and the bottom and welding them together. 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 Engineering Analysis and Linear Actuator Selection

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 Nγ 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)

Currently, the force analysis is 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.

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.

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

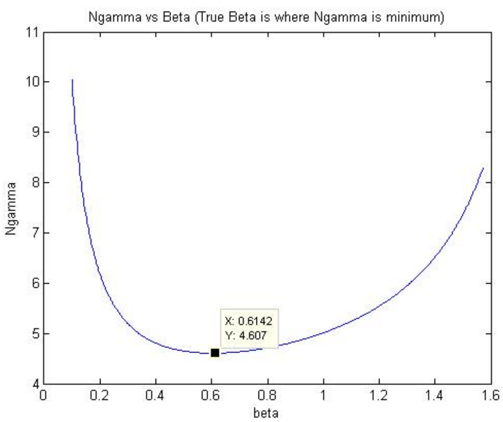


Figure 7.5 Plot of Equation 7.3 with b=60.8cm and z=5cm

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.

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

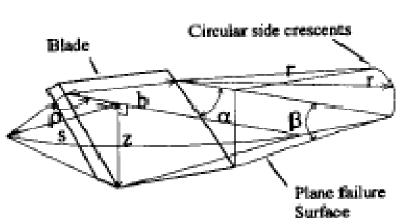
$$s = r[1-\left(\frac{\cot(\alpha)}{\cot(\alpha)+\cot(\beta)}\right)^2]^{\frac{1}{2}} \qquad (Eq\ 7.4)$$

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.



Rupture Surface Proposed by Mckeys and Ali

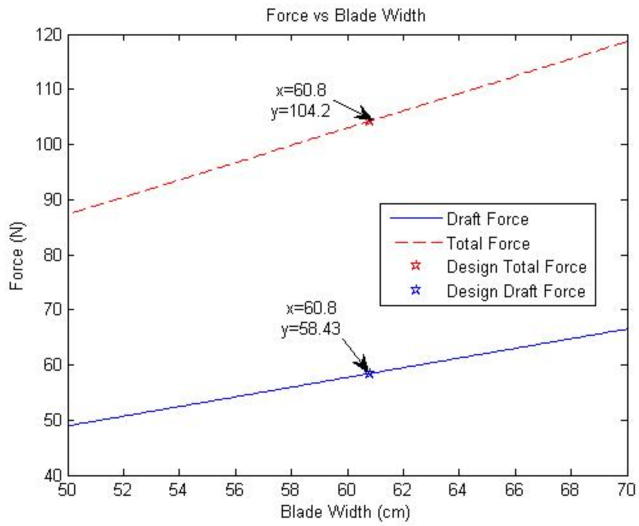


Figure 7.6 Plot of Equation 7.5 for values of blade width assuming small changes in beta.

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.

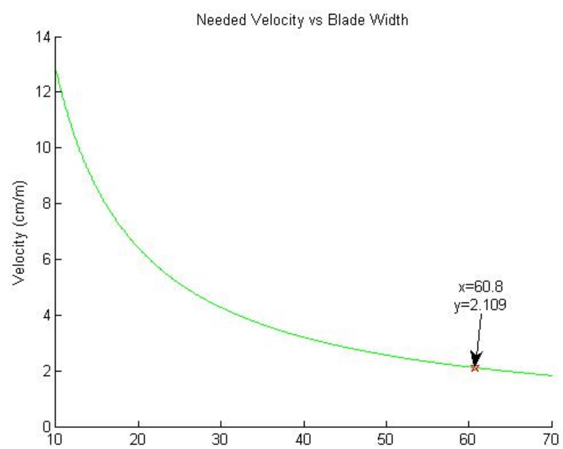


Figure 7.7 Needed Velocity to harvest at least 50kg/h of regolith

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.

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

7.5 Actuator Forces and Actuator Selection

Using the Preliminary concept, a variety of actuators could work in our design. After an estimation of the forces acting on each actuating device, one style stood out as the most practical.

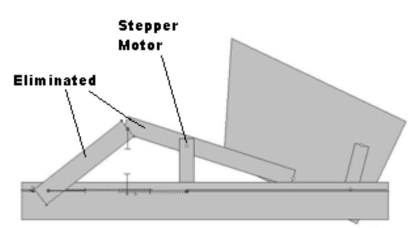


Figure 7.8: Location of stepper motor and links that would be eliminated.

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 100 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.

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. This means that the actuator selected must be very rigid or designed such that it doesn't deflect. The largest axial load seen is when the bucket is near the dumping position and this is a maximum of about 23lbf.

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 traveling just above the required velocity of 2.2 cm/min (3.67×10^{-4} m/s) and a maximum mass of 150kg stopping nearly instantaneously (.001s), using Equation 7.8, the impulse average force is calculated to be 12.36lb-force. If this force were translated through the linkage system, the actuator would experience a force of about 26lbs acting vertically and about 8lbs acting along the axial direction.

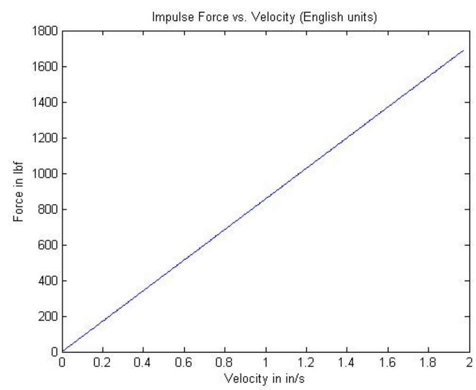


Figure 7.9 Plot of Impulse Force vs Velocity

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

Our group has decided we want an overall factor of safety on our design of about 2 such that each actuator can handle the total load by itself. This means that each actuator must be able to handle an axial load of at least 24lbf and a transverse load of at least 51lbf. Also, using working model we found that the total distance the actuator needed to travel was 15 inches. As a group we decided that we wanted to be able to go through a full range of motion in about 10 seconds. This requirement means that the slider must be able to travel 1.5 in/s under loaded conditions.

The next actuating device considered was a plunger style linear actuator (Figure 7.10). This device would mount to the frame of our final concept. This type of actuator has an anti-backlash nut (Figure 7.11) on the plunging device. An anti-backlash nut (also known as a ball nut) allows the power to be disconnected from the actuator with little reverse movement. This solves the continuous power consumption issue of the stepper motor. A plunger style actuator would have difficulty handling forces in any direction except that of the direction of movement. As a result, a brace that would

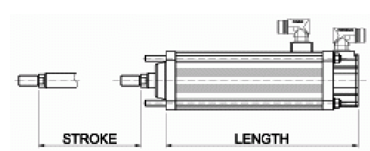


Figure 7.10. Plunger Style Linear Actuator.

Figure 7.11 Types of anti-backlash nuts

attach to the end of the actuator and restrict its motion perpendicular to the direction of travel would be needed.

The final device considered is a guided sliding actuator. This device gives us the ability to use an anti-backlash nut, and still be able to handle large transverse loads acting on the actuator. Currently, the design calls for two of these types of actuators mounted to the linkage system on each side of the harvester. The use of two actuators allows for the load to be split between each actuator allowing for a smaller actuator.

However using two actuators can present a problem of controlling and moving both actuators simultaneously. As a result of slight mechanical and electrical variances, it is possible for the actuators to get out of sync. This could cause undesired loading of the harvester. To solve this with a standard linear actuator, an encoder would be required to observe the position of each actuator to keep them in sync.

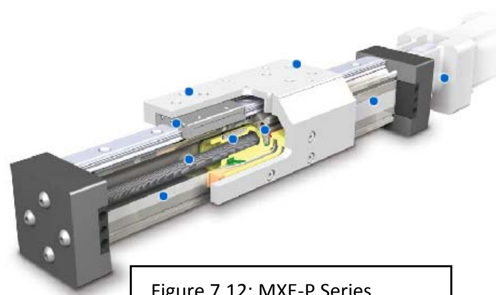


Figure 7.12: MXE-P Series

The company Tolomatic offers an actuator that is a hybrid between a linear actuator and a stepper motor to solve this synchronization problem. Stepper motors offer high position accuracy by controlling the number of steps the motor goes through but require constant current to hold a position. Standard linear actuators hold a position with no constant torque, but may become out of sync. This hybrid linear actuator uses a stepper motor do drive the actuator as opposed to a standard motor. This allows for precise control of how far the slider is moved by controlling the number of steps the stepper motor goes through.

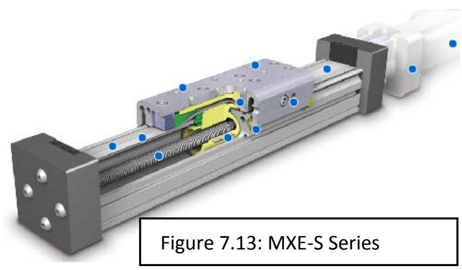


Figure 7.13: MXE-S Series

This is why the Tolomatic MXE-P or MXE-S (Figure 7.12 and figure 7.13 respectively) screw driven hybrid linear actuator is the best actuator for our purpose. Seen in Figure 7.14, The maximum loads that this actuator can handle are about 150lbf acting vertically on the slider. It is designed to accommodate large transverse loads and the positional accuracy for use in two actuators. According to the Tolomatic representative, the MXE-25P will cost $2602.98 and the MXE-32S will cost $1,212.36. Both actuators will handle the forces given, so it is our recommendation to use the MXE-S 32F.

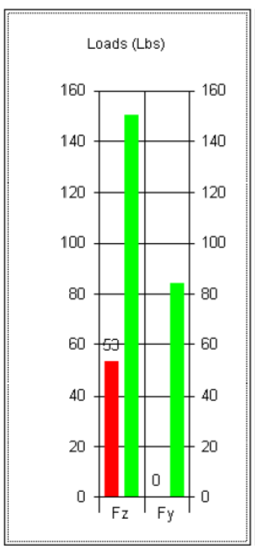


Figure 7.14 Maximum Forces the MXE-S can handle

8.0 Linkage Subsystem

8.1 Specifications and Constraints

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

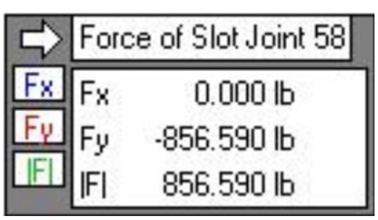
This will be accomplished by the use of three linkages that are mirrored on either side of the assembly. It was discovered through prototyping in both solid edge and working model that it is possible to overextend the actuator so that the bucket enters into an unrecoverable position. This problem is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

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.



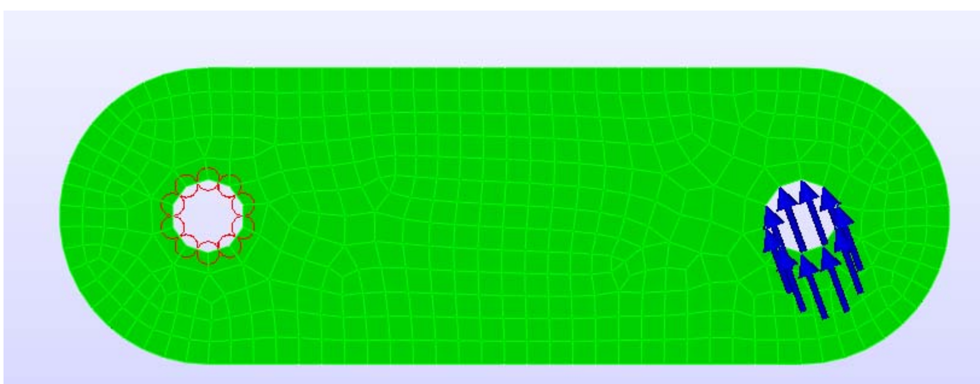


Figure 8.1 Force Transfer Bar Load Condition

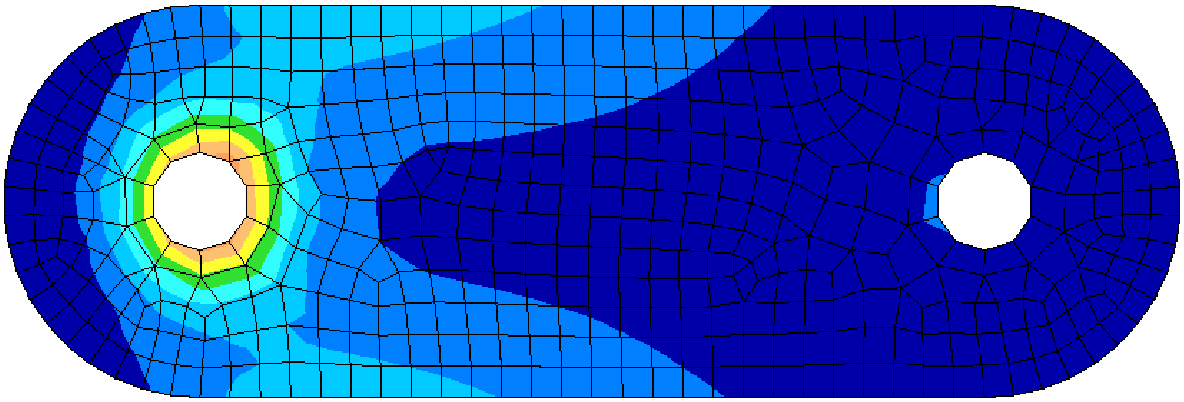
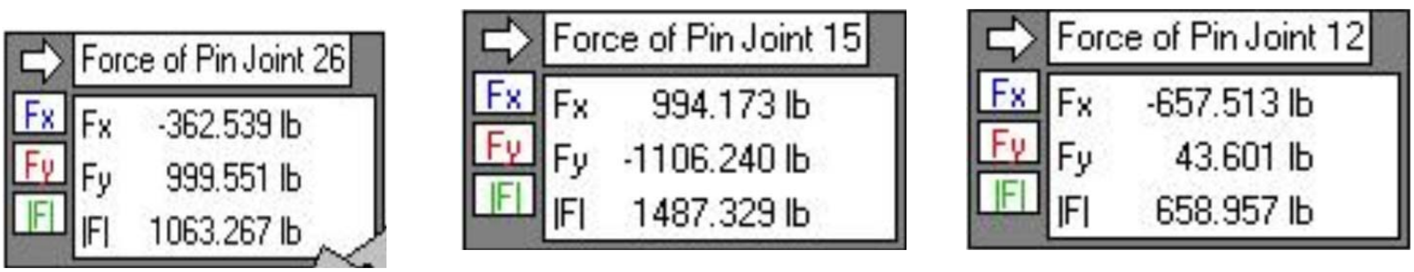


Figure 8.2 Force Transfer Bar Stress

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.



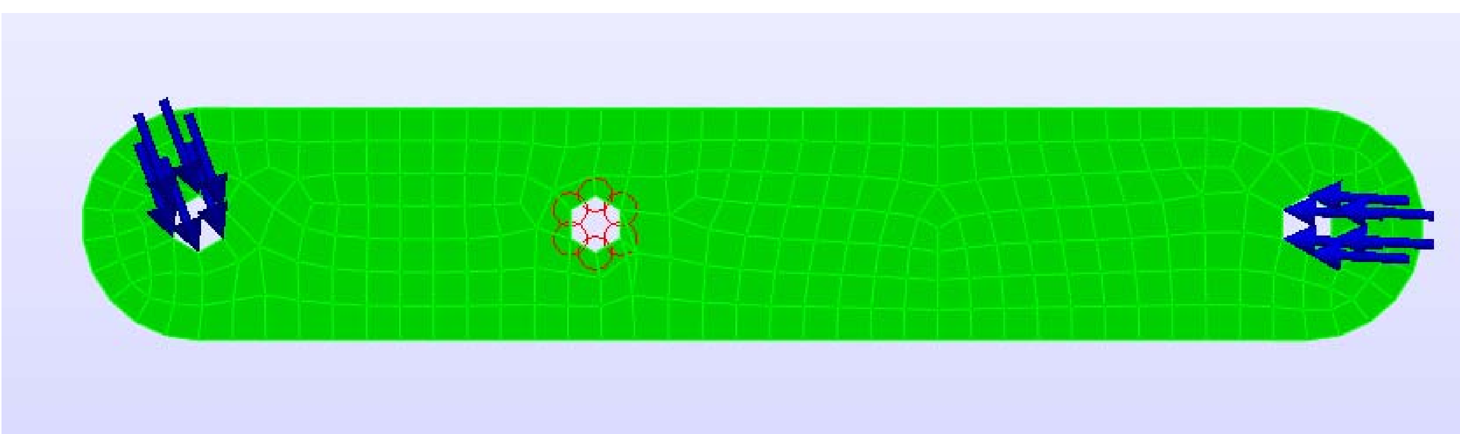


Figure 8.3 Pivot Bar Load Conditions

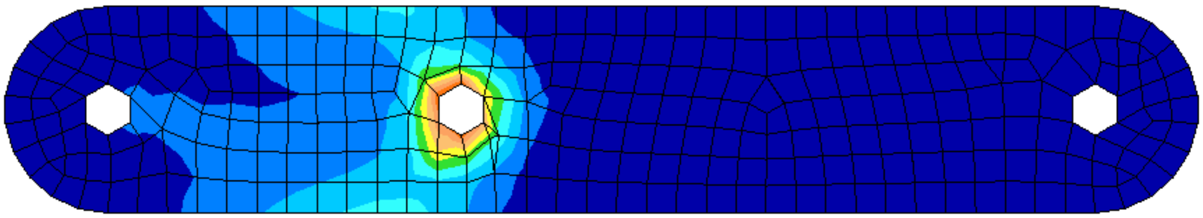


Figure 8.4 Pivot Bar Stress

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.

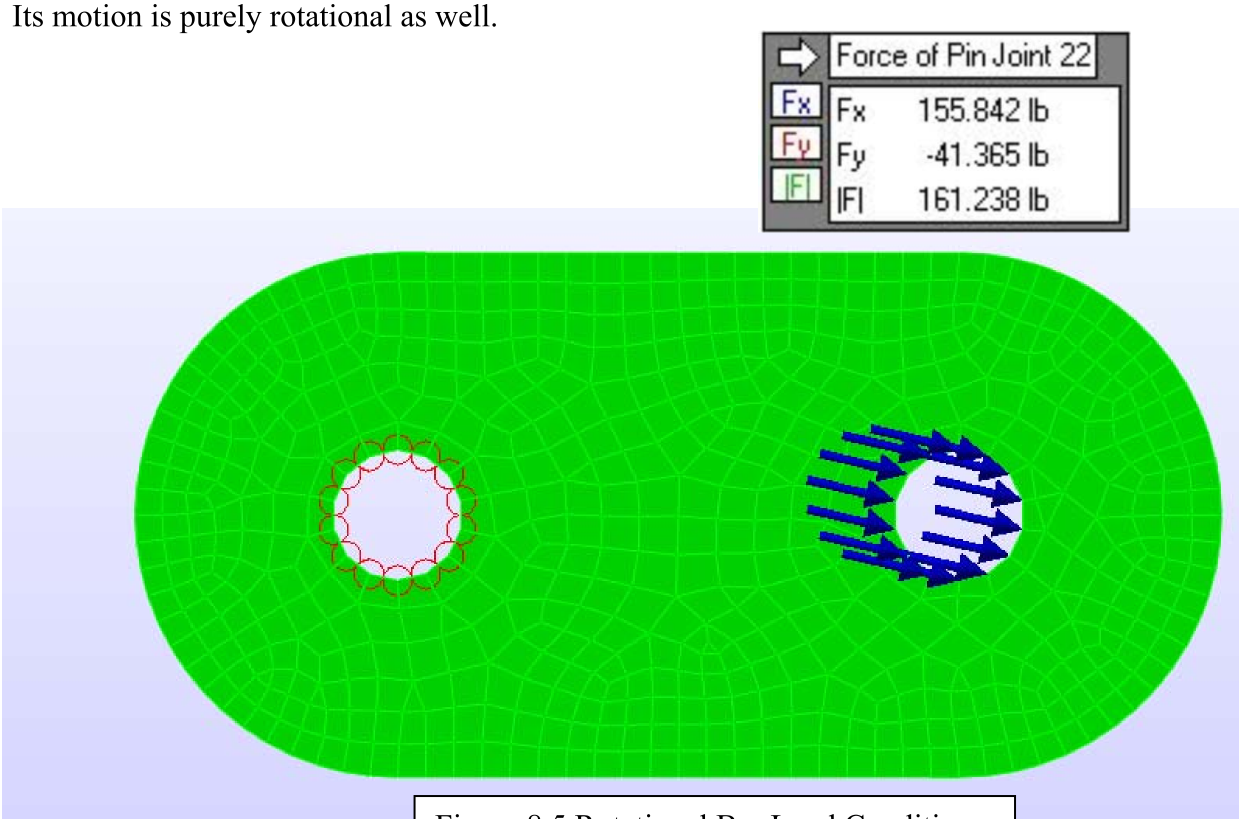


Figure 8.5 Rotational Bar Load Condition

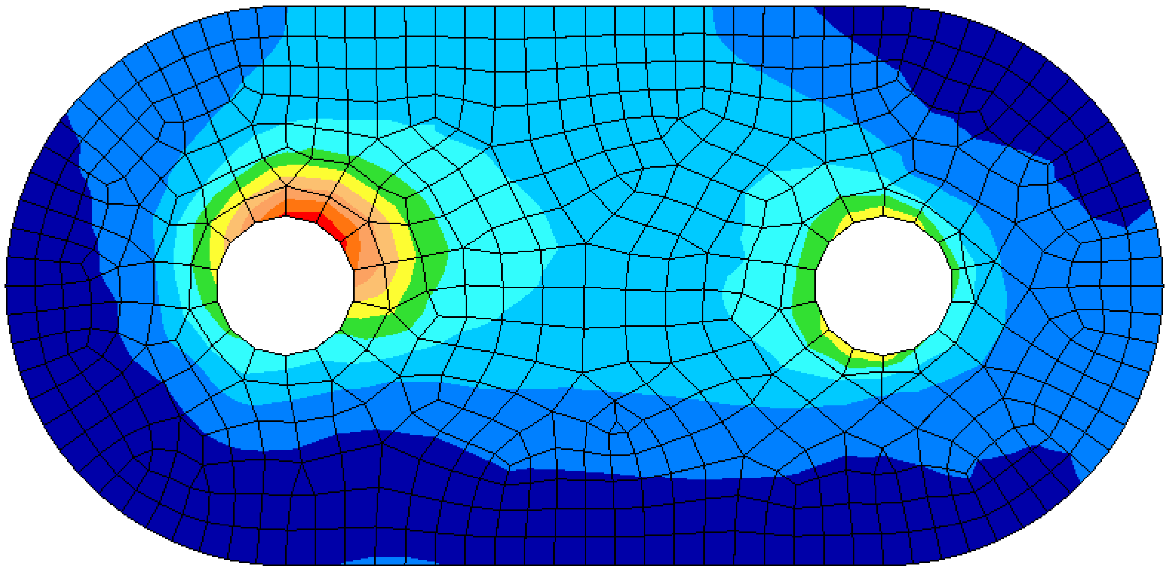


Figure 8.6 Rotational Bar Stress

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

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).

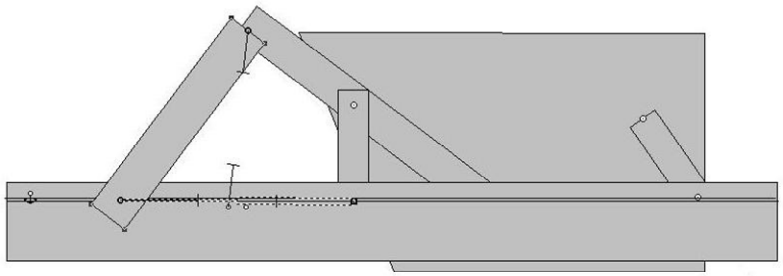


Fig 8.8 Working Model Transport Position

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

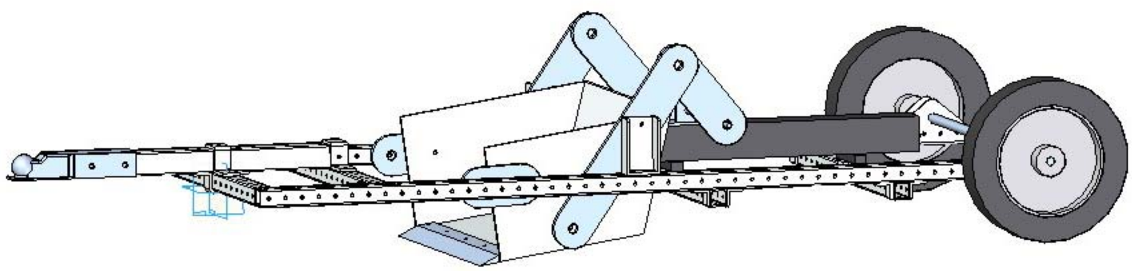


Fig 8.9 Solid Edge Isometric View

8.3 Working Model and Solid Edge Engineering Analysis

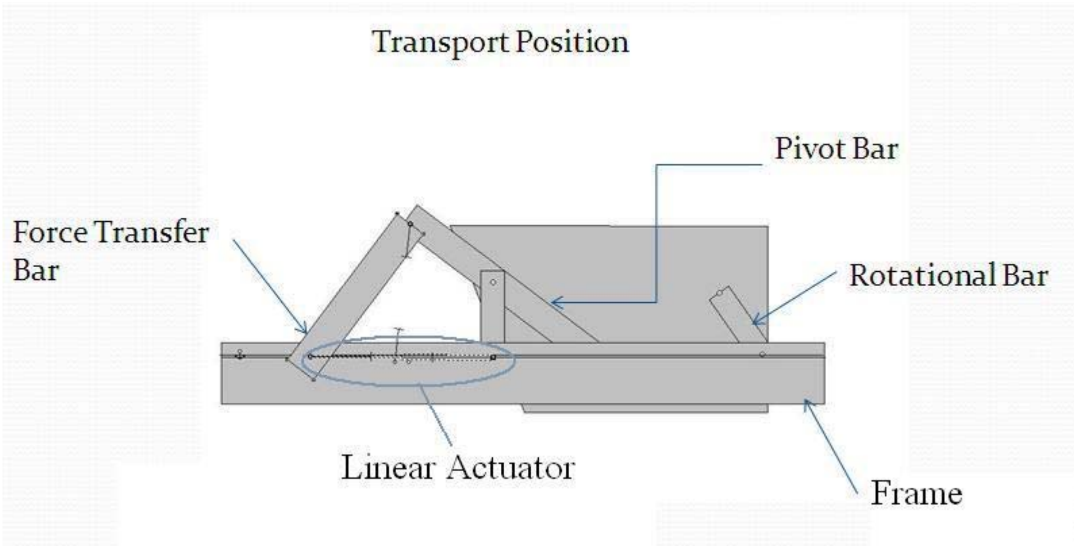


Fig 8.10 Linkage Component Names

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).

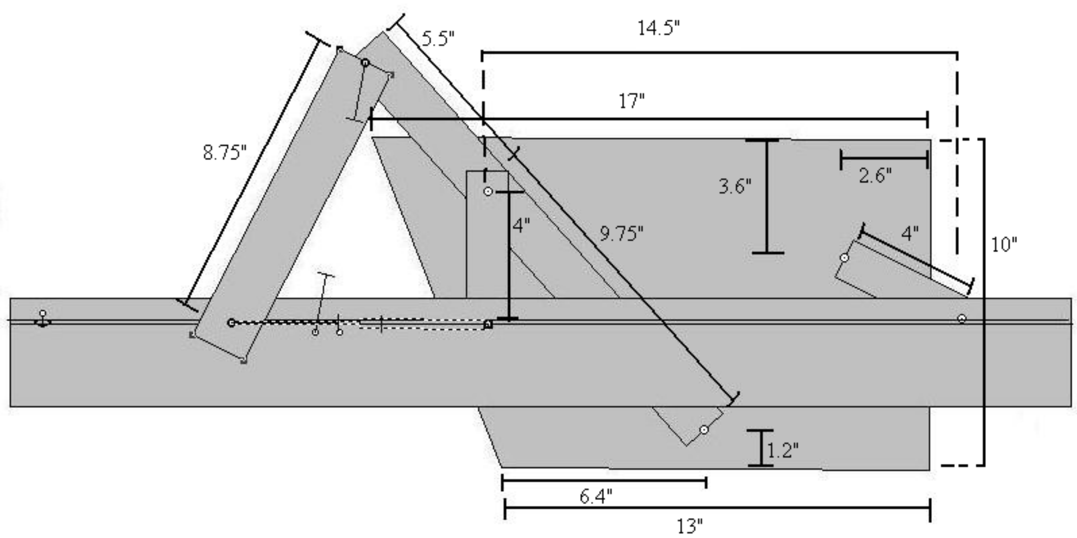


Fig 8.11 Linkage Subsystem Dimensions

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).

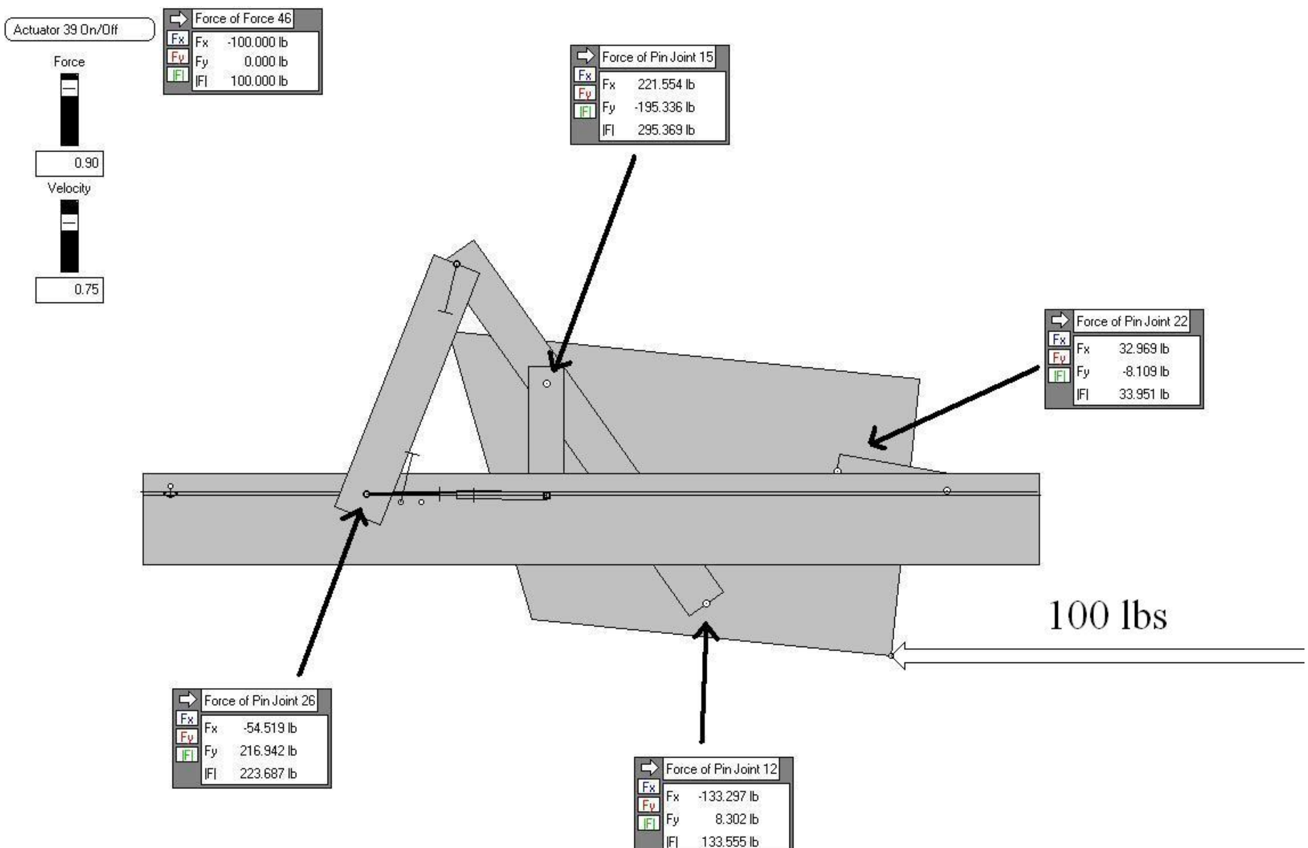


Fig 8.12 Harvesting Force/Load Analysis

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:

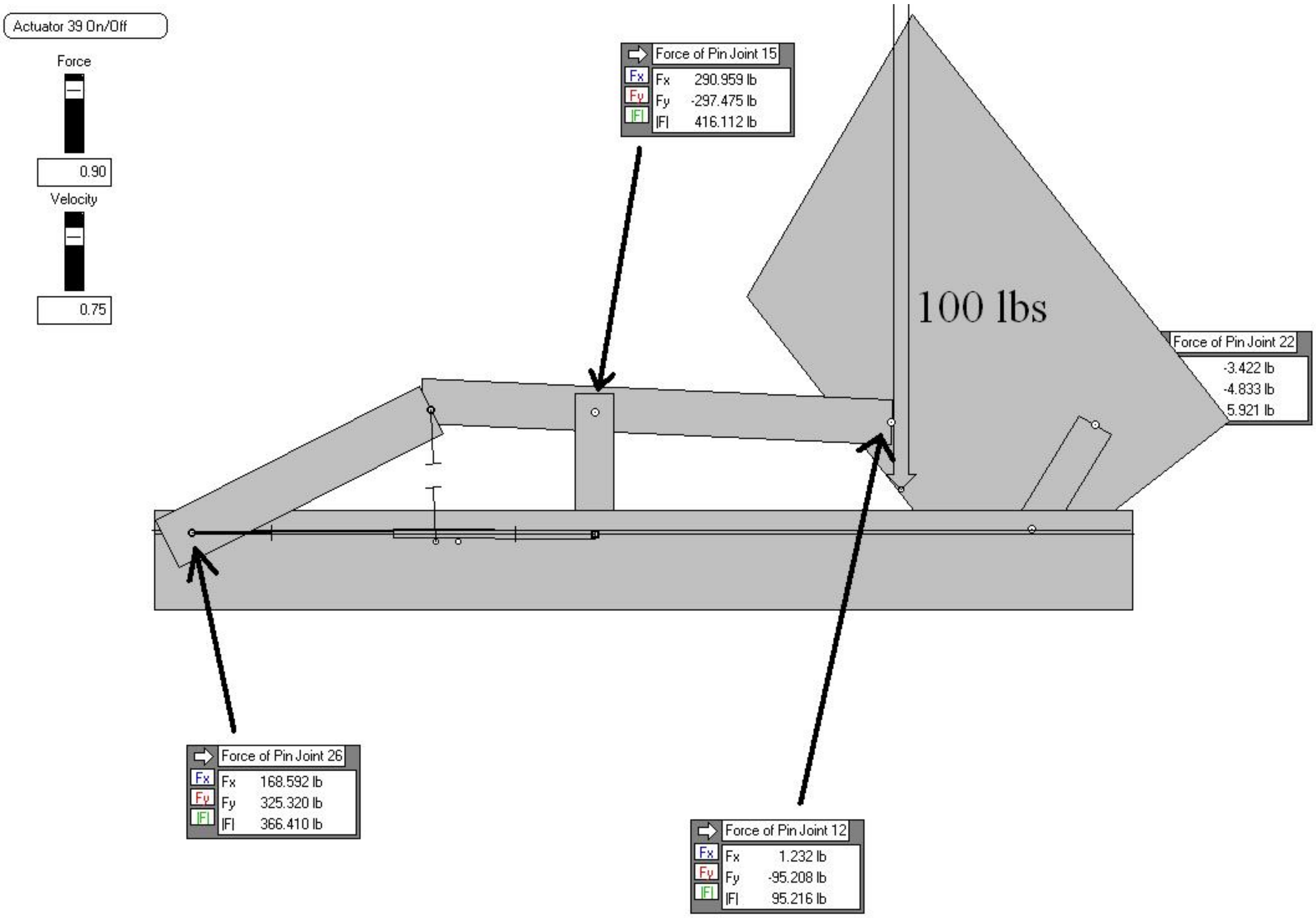


Fig 8.13 Dumping Force/Load Analysis

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.

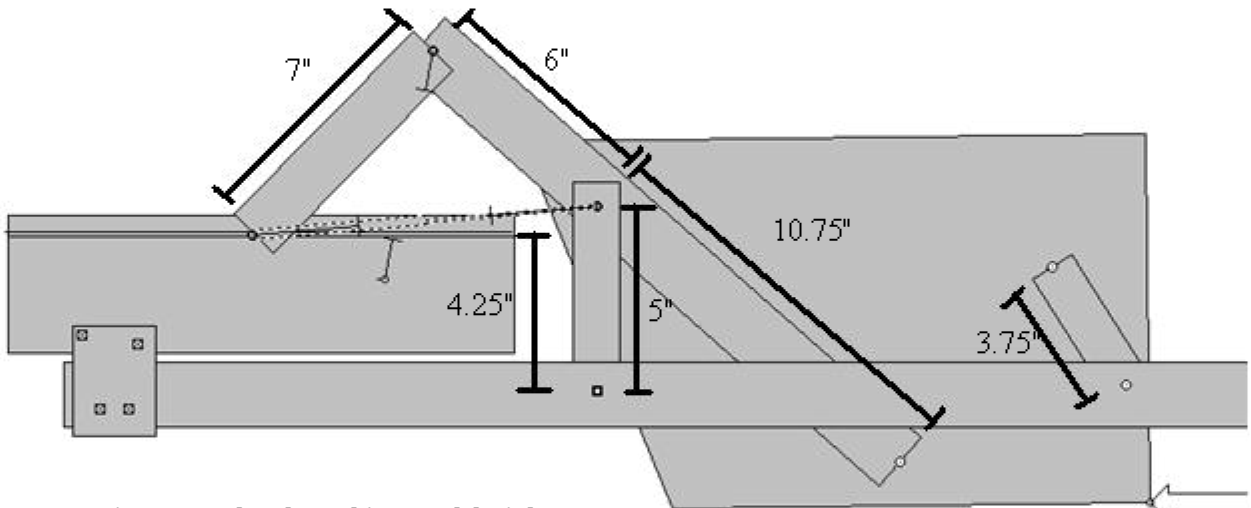


Fig 8.14 Updated Working Model Linkages

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

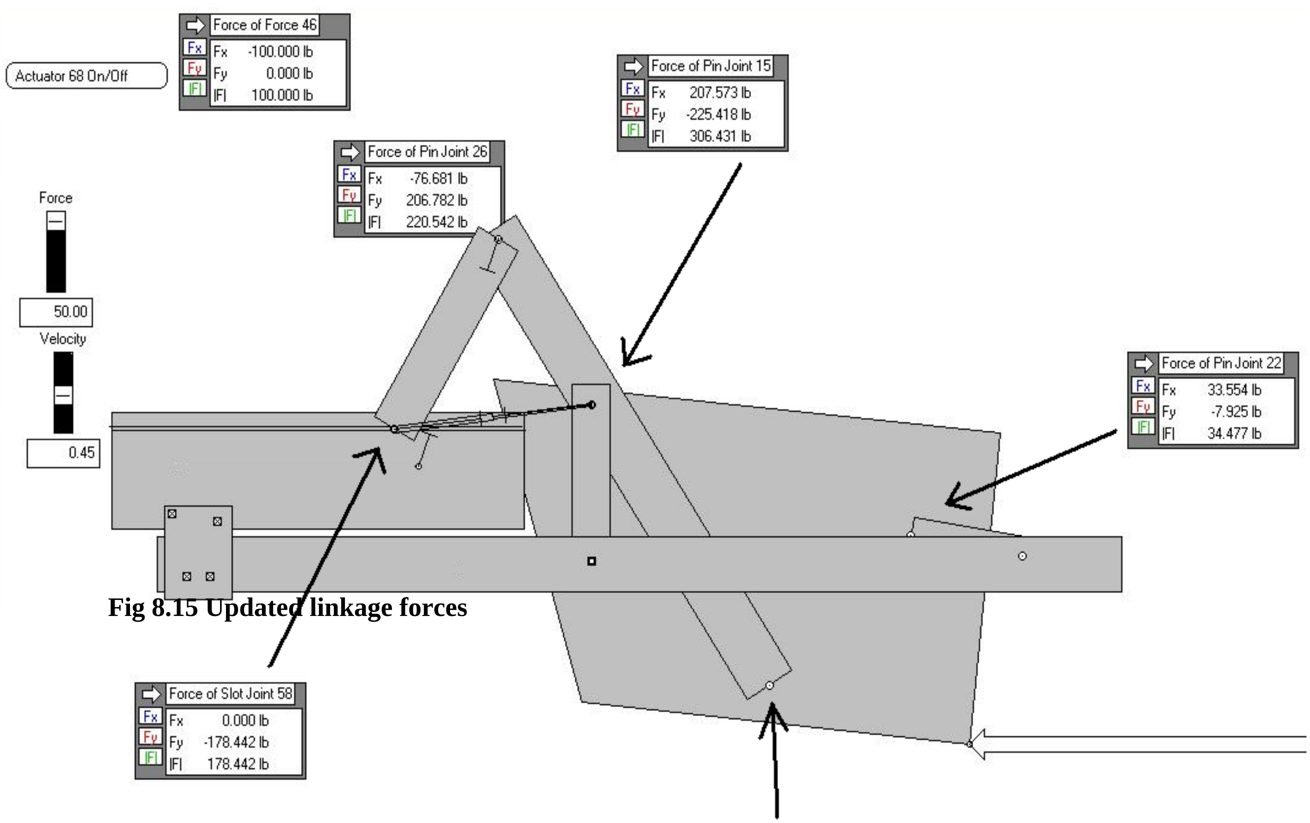


Fig 8.15 Updated linkage forces

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).

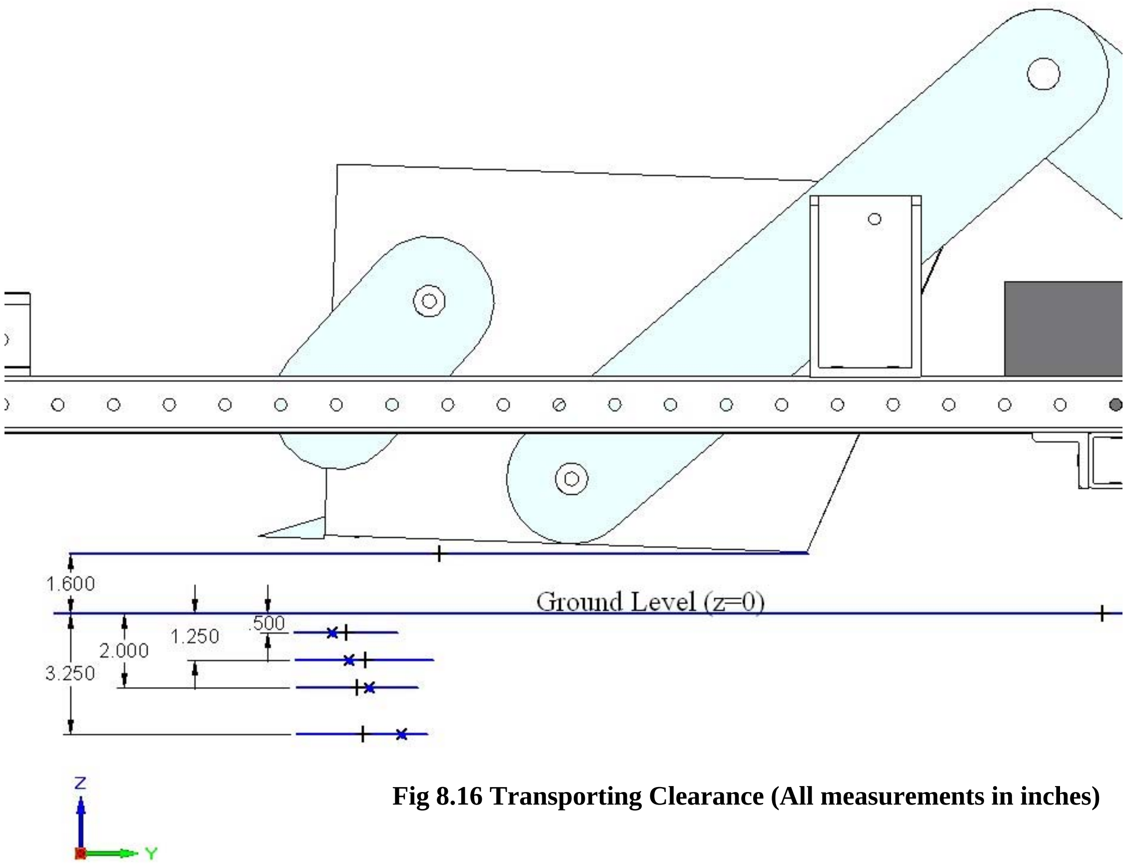


Fig 8.16 Transporting Clearance (All measurements in inches)

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.

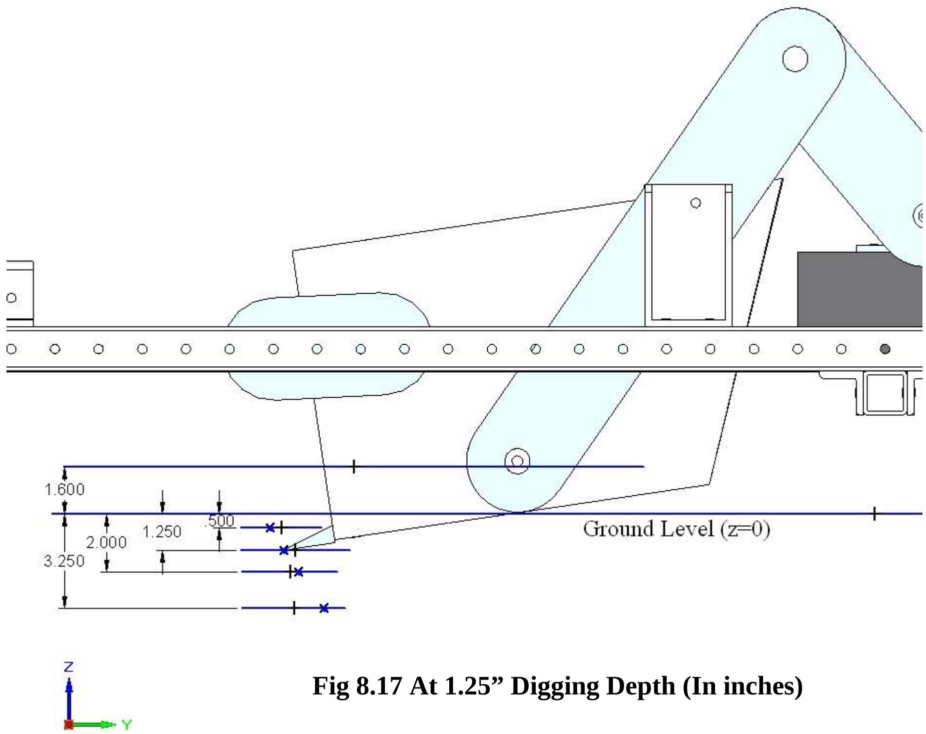


Fig 8.17 At 1.25" Digging Depth (In inches)

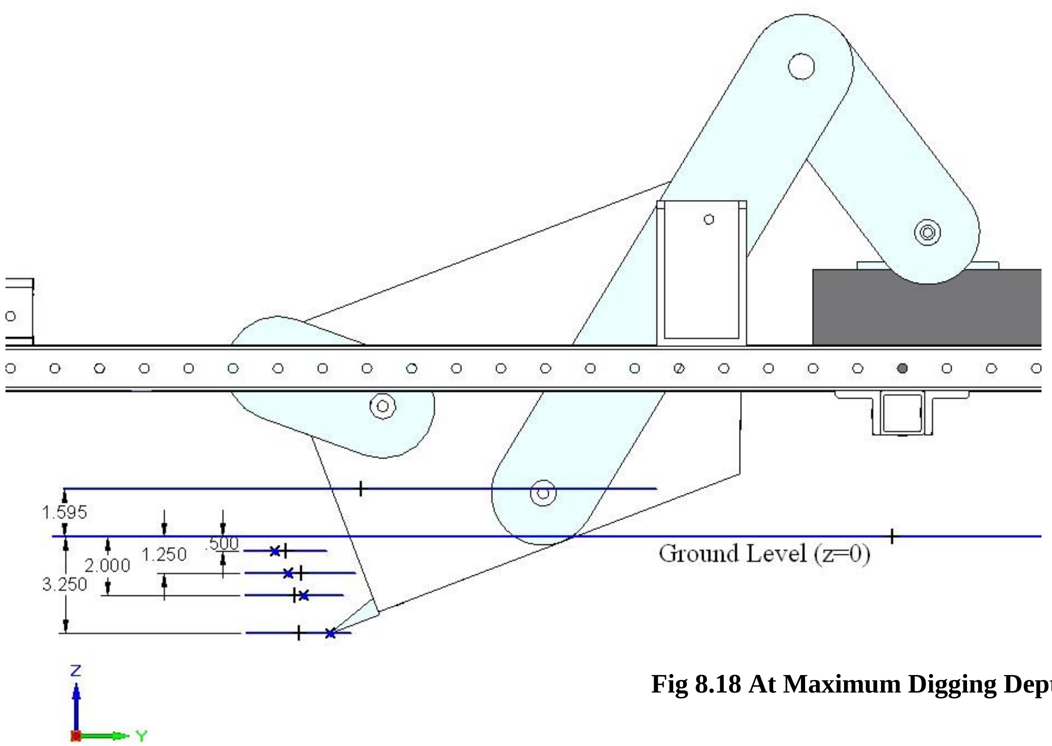


Fig 8.18 At Maximum Digging Depth (In inches)

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

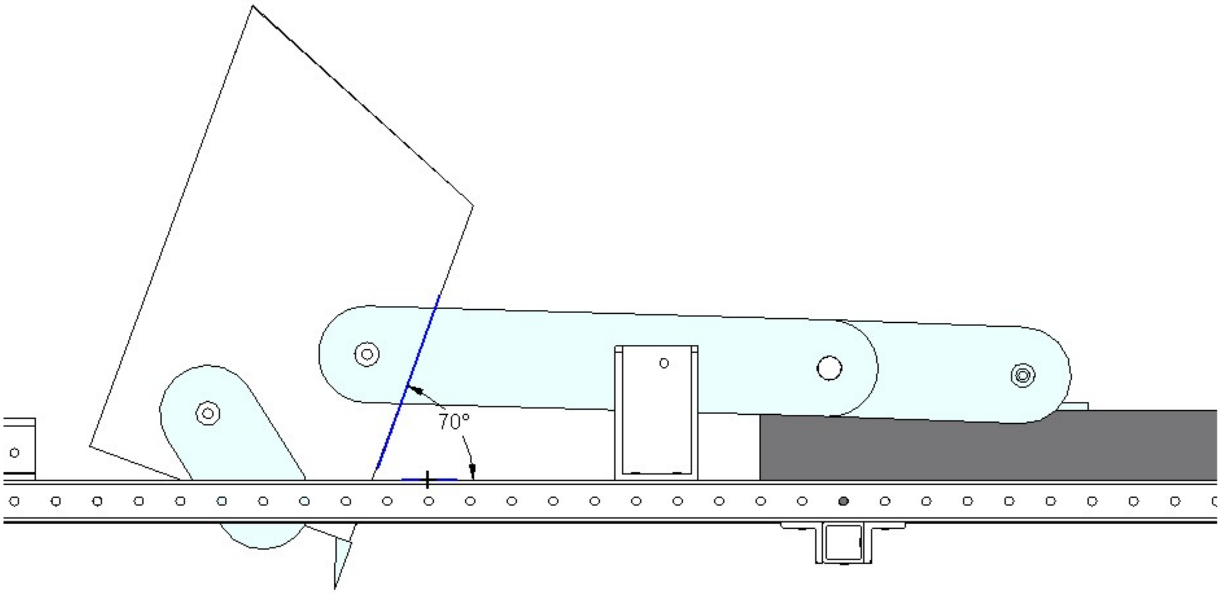


Fig 8.19 Dumping Position Angle

8.4 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”. The reason for the extra length of the bearing is to act as a spacer between other links and also mounting locations. The desired length is not available off the shelf or special order, therefore to get the desired length two 3/8” bearings will be press fit per hole. 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 ¾”. This table shows that whether one or two bearings were used that the bearing is more than capable of handling the loads that have been simulated (see Max. Load in Table 1).

Figure 8.20 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

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 would cause premature failure. 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.

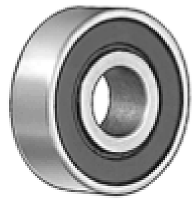


Figure 8.21 Double Sealed Greased Ball Bearing

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 8.22 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 Specifications and Constraints 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

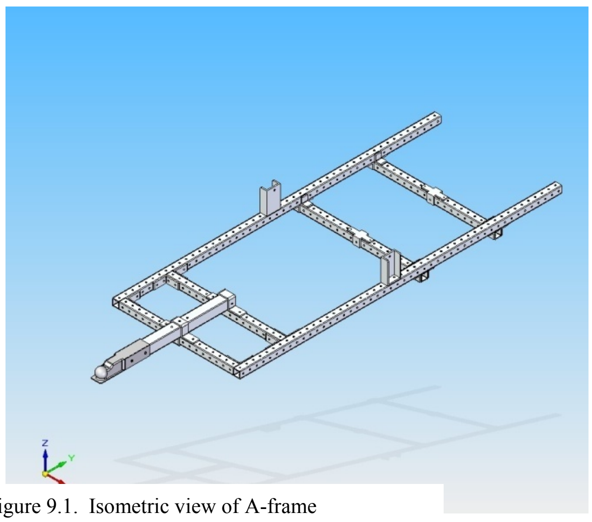


Figure 9.1. Isometric view of A-frame

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 two 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 under the frame to place the actuator as low as possible to give us mechanical advantage. 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 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 has three holes in it; two for mounting with the frame and one for attached the 3-hole linkage. It has the tabs on the sides to increase stability and reduce deformation caused by loads. The actuator will be attached on the rear two crossbars of the frame to allow it to extend to the rear of the frame, allowing the linkages to apply mechanical advantage to the system. In the drawing, two actuator holders are shown at the center of the crossbars. We are using a slide actuator which will bolt directly onto the frame via the two actuator brackets. These securely attach the actuator and hold it horizontal. On the rear of the frame, brackets are attached to connect the wheels. These brackets hold the axles that hold the wheels.

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 three ¼" strap brackets attached with the standard 5/16"-18 bolts. Two of these brackets connect the tube on top of the frame while the third connects the tube to the front crossbar.

From the requirements above, the concept we are presenting consists of an A-frame with 4 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 connector tab 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 holes on the actuator. The actuator will be bolted securely on the center of the rear crossbars to allow for the horizontal movement needed to power the system. 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, brackets 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 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.

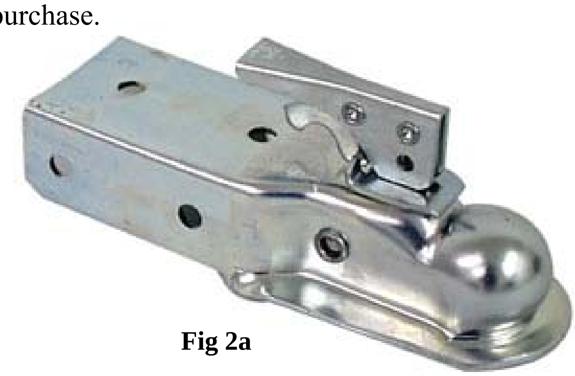


Fig 2a

Fig 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 3).

Fig 9.3 Adjustable 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 4).

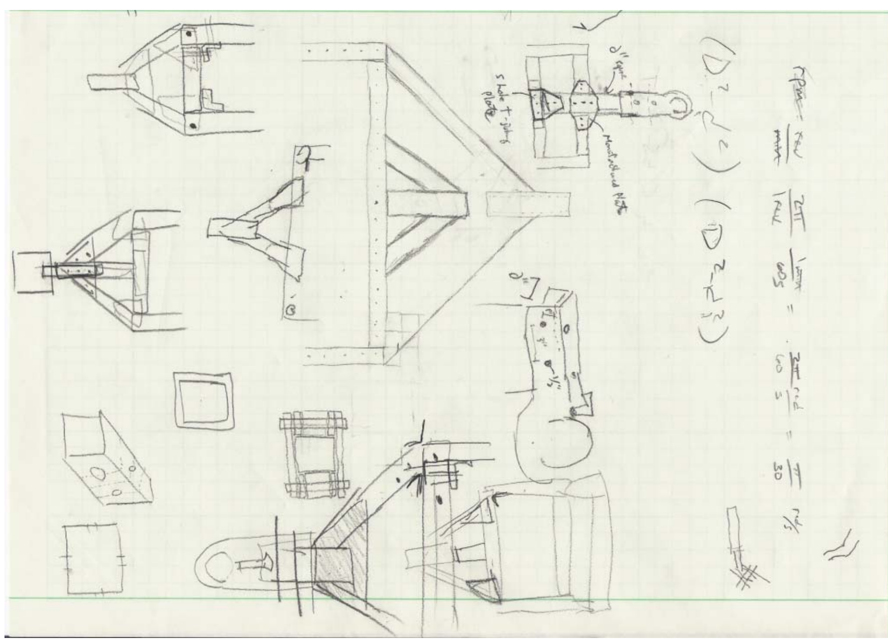
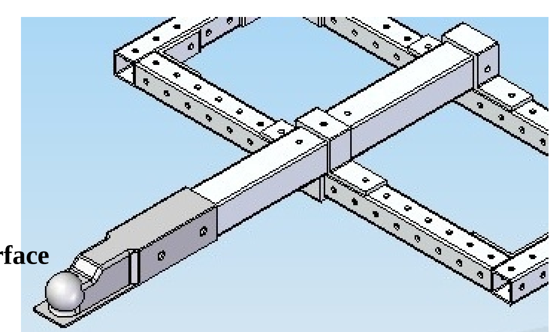


Fig 9.4 Interface Hand Sketches

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 5).

Fig 9.5 Solid Edge Interface



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 through a series of ¼” sheet metal strap brackets, using bolts that are standard with the rest of the frame and linkage assembly. Two strap brackets straddle the 2” square tube on top at both crossover points (Figure 6a,b) and one straddles the 1 ½” frame and bolts to the 2” tube from below (Figure 6c).

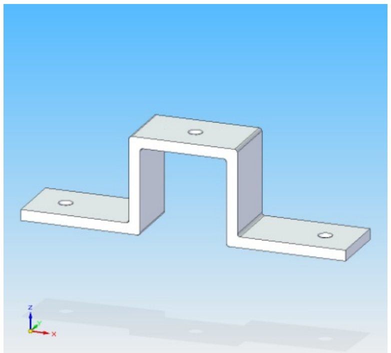


Fig 9. 6a

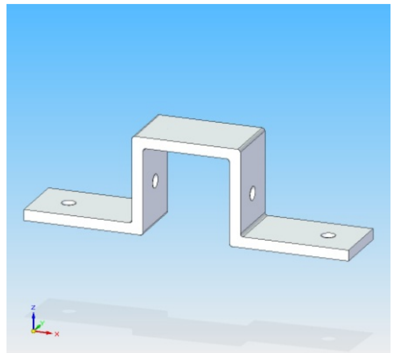


Fig 9.6b

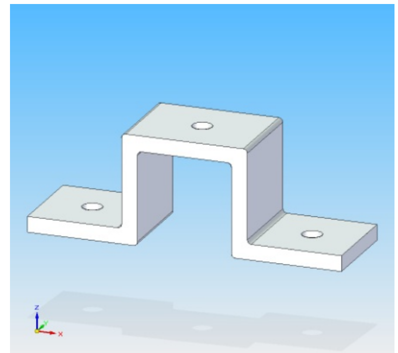


Fig 9.6c

All brackets are of the same width (1 ½”) but the holes are in different locations and the bottom bracket has a smaller inner square profile to match the tube that it straddles. The top brackets’ bolt pattern can be easily matched to be universal if necessary and the overall design can be changed easily through Solid Edge to meet different requirements. All part drawings have been drafted in Solid Edge.

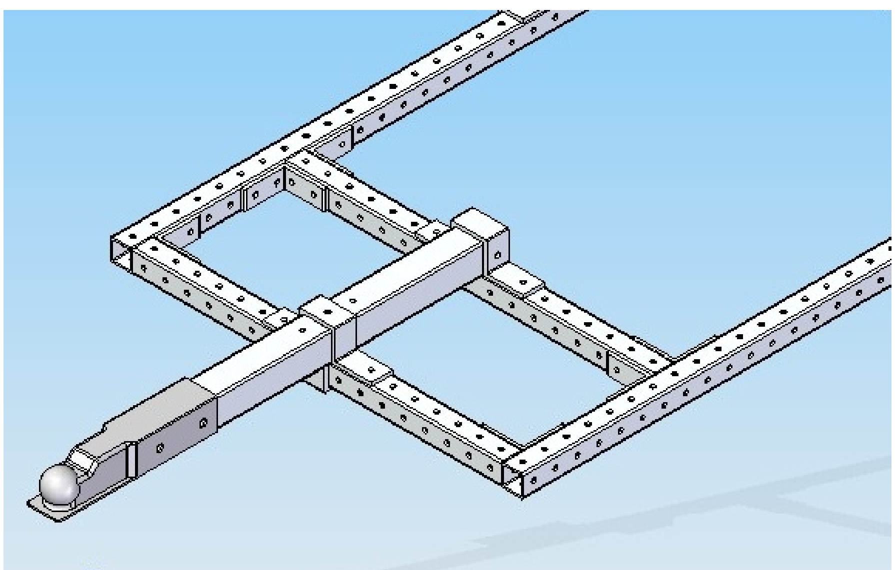


Fig 9.7 Isometric Gator Interface

9.3 Wheel Structure

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.

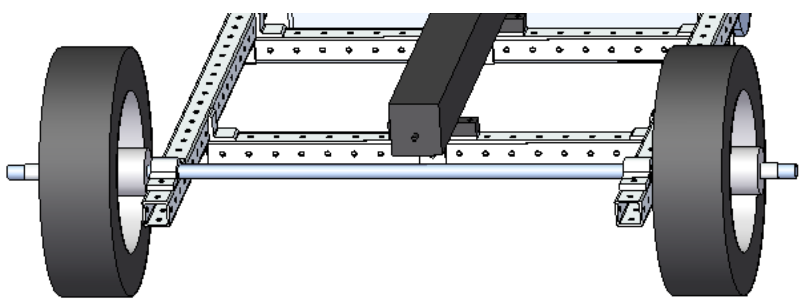


Figure 9.8 Rear View Wheel Assembly

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 mounting brackets bolt onto two of the existing bolt holes and hold the axle immobile. These brackets will be made out of 3/8" thick aluminum and built to accommodate a 3/4" diameter axle and 5/16" bolts.

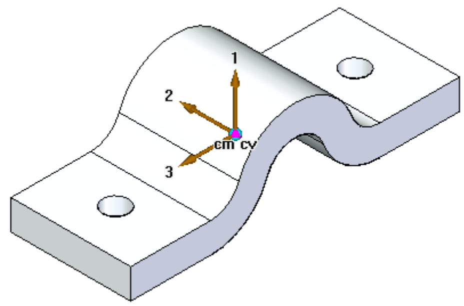


Figure 9.9 Rounded Wheel Bracket

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

11.0 Considerations for Lunar Condtions

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.

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. Also, the trailer hitch that is currently providing for attachment to the test vehicle will be replaced with the interface plate. This is lighter and will make it easier to reduce the total weight of the vehicle. 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.

12.0 Conclusions

The regolith pan is a complete redesign of the 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. The Lunar Prototype will interface with the Gator utility vehicle. 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

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

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