The Monarch Wheel III Final Report

Senior Design Project

MAE 435 – 25178

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1.0 ABSTRACT

The Monarch Wheel III project is focused on producing an innovative new design for a rover wheel capable of traversing either the lunar or Martian surface. The wheel design meets or exceeds design constraints that had been established for a previous NASA hosted design competition as well as additional constraints levied by test vehicle selection. The lunar wheel design improves upon previous rover wheel designs through both an inventive tread design, as well as innovative incorporation of springs in the structure of the wheel.

The team has proposed a tread design that incorporates independently spring mounted feet. This design was chosen in an effort to improve both lateral traction as well as improved handling over various terrains including the very loose, low tractability of regolith and the point loads imposed by the various geometries of rocks encountered on the lunar or Martian surfaces. In an effort to address regolith clogging issues the group has researched various chemical and biological coatings which would reduce adhesion of the minuscule regolith to the various wheel components.

The group produced a one fifth full scale prototype. Lab testing and environmental testing over various terrains were conducted utilizing the final prototype. The results of the tests as well as the proving of the wheel will be analyzed and are documented in this final report and the final class presentation.
2.0 INTRODUCTION

National Aeronautics and Space Administration (NASA) has sent 6 men and 19 rovers to the moon. These journeys required each vehicle be outfitted with wheels capable of functioning in the special lunar environment. A previous lunar wheel design which was used on the Apollo 15 mission was the Wire Mesh design developed by General Motors [1]. This design was light weight, provided a comfortable ride, and traversed obstacles well. However, the Wire Mesh design accumulated lunar dust in the wire mesh, which created a smooth surface around the wheel thereby reducing the traction. This caused the wheel to act similar to a smooth inflatable rubber inner tube which relied solely on the surface area and the deflection of the wheel to provide traction[2].

In the near future, NASA is planning a mission to revisit the Moon and to explore it with innovative lunar technologies. The prototype for future lunar rovers will carry a heavier load and will travel a farther distance than the rovers that were developed in the past [3]. The Lunar Roving Vehicle (LRV, Figure 1) has the ability to cover a large amount of land, carry more samples, and get more scientific exploration done in less time than previous missions [4]. Although not currently in operation, the LRV was a success and showed that Goodyear’s wire mesh wheel (Figure 2) or elastic wheel would function in the lunar regolith [5]. Even though the LRV was an efficient vehicle, there were improvements that needed to be made [6,7]. The LRV had issues with the amount of regolith clogging the tires, extensive mesh deformation, and general terramechanics, which caused the main issue [6, 7].

During a manned mission, a functioning rover will reduce the amount of time astronauts will be exposed to harsh radiation, reduce the amount of oxygen used by the astronauts, and allow the astronauts to move farther away from the landing site. During an unmanned mission, high performance wheels will prevent a rover from becoming stuck and ending the mission, costing the tax payers millions of dollars. Some preliminary research that needed to be done included: springs, durability of different materials, geometrical structure, and clogging issues. A recent design proposal used a honeycomb structure to support the wheel, but there have been problems getting the honeycomb to support the weight of the rover as well as issues with the honeycomb clogging with lunar dust similar to the General Motors wheel, degrading its performance [8].

The lunar surface is the most important factor when considering design criteria. Lunar exploration presents several conditions that must be considered when developing testing for a wheel design [4, 6, 7, 9-11]. Our lunar wheel design will be subjected to an environment which combines near vacuum, intense radiation, and extreme temperatures [10]. Furthermore, the lunar surface consists of large rocks as well as tiny particles of lunar soil. The lunar rocks typically

Figure 1 - Lunar Roving Vehicle

Wheel
range from heights of 6cm to 25cm however about 5% of the rock traversed can range from 25cm to 50cm [12]. The particles of regolith are very tiny which makes the wheels susceptible to slippage and sinking due to rutting; this may result in time consuming delays or even vehicle immobility [13]. Therefore the wheel must be equipped with tread designed to be able to provide lateral and longitudinal traction. It is impractical to evaluate the production model prototype of the lunar wheel design on the lunar surface. Therefore testing must be carried out terrestrially on approximated terrains [7, 9, 14]. Although not under true conditions “real world” testing provides performance data when the design is subjected to combined loadings and dynamic conditions. One method of improving traction that has been used is having each wheel independently controlled by a computer to allow the rover to compensate for any slipping that might occur [14]. This system works similarly to an anti-locking braking system in a car, in that the person driving does not need to engage the system.

Previous lunar wheel designs were capable of absorbing small deflections in the terrain and smaller rocks; however, they would fail under point loads. There have been many studies on different elastic wheel and tread designs and how they affect various mechanical and motion parameters [6, 7]. The loading conditions of the wheels were tested on flat surfaces, climbing up or driving sideways on hills [9]. According to the team’s advisor, last year’s team had problems with turning during slope climb. Thus, the spring 2014 team can use those problems as a reference to improve the wheel’s capability of climbing up slopes. System architecture of a rover simulation environment can be a useful mechanism to test the wheels [8]. An iterative process has been developed and will be used to define the torsion stiffness properties, minimize rolling resistance and measure the static deflection of the elastic wheel due to a 5000 lb. load [4]. Once the number of springs is found, then the stiffness/dampening properties and the rolling resistance can be calculated following the methods described in section 3.3 of the study [4]. Due to the unpredictable nature of the moon and the fact that the conditions are very difficult to simulate physically, most of the testing is conducted using finite element models (FEM) simulation analysis with a 3-D model of the wheel created in Inventor [4, 6, 15]. There are several methods of testing that will be performed on the designed wheels including: computer-aided simulation; scaled replica testing on the tread design; and an engineered one-fifth wheel prototype that will undergo testing [4-7]. Each method of testing is intended to save time and money due to the lack of time and budget constraints [4, 6]. The methods will build on each other, each iteration refining and improving upon the previous design. This process reduces cost by eliminating poor design choices before constructing costly production prototypes manufactured with engineered materials [4, 6, 7, 14]. Few studies examine whether simulated testing of a lunar wheel design translates through scaled replicas to production model prototype testing [7]. After the preliminary research was completed, the geometry and dimensions were determined. The dimensions were based off the guidelines, and later changed due to the results of finite element analysis testing.
3.0 METHODS
3.1 DESIGN

The purpose of the lunar wheel design is for a wheel to be capable of handling different types of surface terrain. The wheel should be able to handle large point loads; this would typically come from large or jagged rocks, as well as be able to distribute its own weight over a large enough area to reduce the possibility of sinking in the moon’s regolith. The design of the wheel needs to be as open as possible to allow any regolith that happens to collect inside the wheel to be able to be expelled.

The team created and assembled all the parts of the design using AutoCAD 3D Modeling (Figure 3) (Autodesk, San Francisco, California) and Inventor (Autodesk, San Francisco, California). The assembled wheel is made up of three layers. The inner layer was designed as a rocking/hinged spoke. Originally the design had hinges on both sides of the wheel with steel C shaped springs in between (Figure 4). To allow for a less constrained rotation the design shifted to a singular series of spokes, stiffened by torsion springs and a mechanical tilt brake. Each of these components is centered on the hub allowing for the wheel to more capably move over angled rocks and other obstructions. The materials for the various parts were selected, discussed in section 3.2, and drawings were sent to the machine shop for production.

The middle section, also known as the C layer due to the fact it is comprised of circular springs, is similar in concept to the circular spring layer used on the new Michelin moon wheels. This layer was developed to deflect large loads from directly impacting the spokes and hub. These springs were originally designed in a unique kidney shape; however, once analytical analysis was complete it was determined that these would not be functional as they had been designed. The ideal springs used here would be elliptical shaped, but due to a budget constraint, circular springs will be used as shown in Figure 5.

Figure 3 - 3D Modeling Inventor File

Figure 4 - Inner Layer with Singular Series of Spokes

Figure 5 - Circular Springs in the middle layer
The outer section, also known as the gecko foot layer, is comprised of feet which are attached to springs (Figure 6). Our theoretical design includes a specially made tensile and compression wave spring. This allows the foot to absorb and or deflect point loads. For the prototype, compression springs will be used with a mechanical brake line to avoid application of tensile forces to the spring itself.

Each section has an important feature that contributes to the wheel as a whole. The wheel needs to be sturdy in the center giving little flex but at the same time be able to absorb loads and shocks such as a car suspension does. The tread will need to be the most flexible because it needs to be able to form around the terrain surface. This being said it is also the most difficult part of the design because this is where there is going to be the more shear and lateral forces being applied.

The tread design has been a great deal of work. The team had to come up with a tread that would allow the wheel to grip onto almost any surface (Figure 7). This tread design is called the “gecko tread,” and this allows each individual tread to have its own independent support to the wheel’s rim. This allows the tread to form around the surface that it is applied to thus allowing there to be more contact points to the surface. The “gecko tread” is designed to be able to take on large shear forces, and spread the point load.

The inner hub support was a very unique idea that was brought to the team as it allows relative movement between the hub and rim. Most hubs are stationary and do not allow any movement in the wheel. The design of the hub is to allow the wheel to angle more of itself parallel to the surface thus allowing more contact surface. The inner hub has a total of ten hinged supports, five on the outside and five on the inside.

The wheel as a whole is a very unique and complex design. This design has given the wheel more flexibility in overcoming different types of terrain, therefore helping the lunar vehicle to achieve durability. The team has applied FEM analysis to the finalized structure and has fabricated some of the unique parts. 3-D rapid prototypes were made for tread depth testing. Refer to appendix VII for wheel assembly of a 1/4th wheel design.
3.2 MATERIALS SELECTION

Aluminum is one of the lightest metals, with a density about one-third that of steel. The strength of aluminum alloys can approach 100 ksi (700 Mpa). The combination of high strength and light weight makes aluminum well suited to transportation vehicles [16]. Three aluminum alloys were examined to find the best material for the lunar wheels including aluminum alloy 6061, 6063, and 6066.

In Table 1, some common aluminum alloys are compared based on strength, corrosion resistance, and cost [16]. The aluminum alloy 6061-T6 provides the best combination of strength, economy, and corrosion resistance of aluminum alloys and is widely used and easily available. The aluminum alloy 6063-T6 has a lower strength, higher corrosion resistance, and is slightly lower in cost compared to 6061. The aluminum alloy 6066-T6 has higher strength but is less corrosion resistant and is harder to extrude therefore, more expensive. Where strength is needed 6061-T6 should be considered, but if an even higher strength is needed 6066-T6 is a good choice. Where strength requirements are less demanding 6063 is well suited [16].

According to NASA, the temperatures on the light side of the moon reach 122°C whereas on the dark side of the moon go down as low as -156°C. Aluminum alloys of the 6xxx series are great for very low temperature applications because of the detailed documentation that their ductility, toughness as well as strength, increases at subzero temperatures including absolute zero (-273.15°C) [17]. Upon further research, aluminum alloy 6063 was found to not meet the temperature requirements of the lunar environment. Aluminum alloy 6063 was found to age at a temperature slightly below 122°C.

Having established operating parameters the materials group was able to move forward with the necessary research and calculations to choose the various materials that would be used to construct the prototype wheel. The materials group evaluated the options based on strength of material, corrosion resistance, malleability or difficulty associated with working the material, cost, and operating temperature tolerances. Having evaluated all options, the team decided on the aluminum alloy 6061-T6 as it has reasonable yield strength, relative low cost, and good corrosion resistance.

Other components in the wheel that required a greater strength, such as the springs and rods used in the spokes, were chosen to be made with 316 grade stainless steel. The 316 grade stainless steel with yield strength of 205MPa and a tensile strength of 515MPa makes it much
better suited for the spring and spoke applications [16]. Ideally titanium would be used in place of the steel rod but due to budget constraints stainless steel is more affordable and should still have the same performance for all conducted experiments.

An important aspect to the project is finding a solution to the clogging of mechanisms and adhesion of lunar regolith on lunar vehicles and surfaces. One promising result of this research is a coating made from the lotus flower that has self-cleaning properties. NASA teamed up with nGimat, a nano materials company, to develop a coating for lunar vehicles and surfaces [18]. This coating has been tested under lunar conditions and has shown promising results.

### 3.3 TESTING

Traction tests were conducted under both longitudinal and lateral slip conditions. A scaled, 3D printed, model of the tread and tread pattern was tested under both control and slip conditions. The control and slip tests were conducted utilizing scaled replicas over three surface types. The surfaces consisted of sand uses as a regolith simulant surface, a rocky surface, and a mixture of the two. The tests were conducted both over a flat surface of all three compositions to evaluate longitudinal traction performance as well as on a variable angled plane to evaluate lateral traction performance. A visual inspection method was established to determine amounts of lateral or down plane slip.

Compression testing was conducted on the proposed design. A point load test was conducted on components of the wheel (Figure 8). In the point load test the components was placed between two conical steel platforms and compressed until failure occurs. Furthermore, the production prototype was compressed between two flat steel plates until failure occurs.

One of the traction designs was plotted by a 3D printer at Old Dominion University. The design was analyzed with a finite element analysis to determine the thickness of the sheet metal required the best position of the rivets, the best positions of the cleats and which cleats was most effective.

### 3.4 TRACTION TESTING

To have an approximation of how our traction system was perform on the loose regolith of the moon four wheels, one-fifth the original scale, were created to test on a remote control car. The environment that the remote control car was tested on was a golf course bunker with varying sized granules of sand.
The model wheels were first drafted in Autodesk Inventor scaled down from their full size and slightly modified in size proportions so that there would be a 0.6 inch gap between the treads to allow for the proportional relative growth of the sand grains to the wheels traction system (Figure 9). A center hub was then designed to match the dimensions of the remote control car that the test would be conducted on. The rapid prototyping division at NASA Langley Research Center was willing and able to help our team produce 3D models out of plastic.

The tests were performed using a TRAXXAS E-MAXX RC car (Figure 10).

4.0 RESULTS

4.1 FINITE ELEMENT ANALYSIS

Finite element analysis was performed on components of the wheels. In figures 11-13 the finite element analyses of elliptical and circular springs as described in the design section with different point loads and numbers of rivets are shown. As shown in Figure 11, the analysis of the elliptical spring showed less stress concentration than the two circular springs (Figures 12 and 13). Due to budget constraints, elliptical springs were thrown out and the circular spring with four rivet holes was chosen because it had less stress than with three rivets.
Inventor (Autodesk, Inc. San Rafael, CA, USA) was used to create the thorough drawings, and to perform the analysis on parts of the wheel. The team created one fifth of the wheel to test and for used to demonstrate how the unique parts perform such as the gecko foot and the hinged center hub. Different types of loads at different angles were applied to the apparatus of the wheel to replicate the different types of conditions the wheel might be subjected to while on the lunar surface.

![Figure 11 - Finite Element Analysis on Elliptical Springs with 1000 lbs. point load](image1)

![Figure 12 - FEA with 3 Rivets hole on 1050 lbs. point load](image2)
The gecko tread, circular springs, the hinged center hub, and the two rim lays were subjected to loads ranging from 500lbs to 1050 lbs. under multiple conditions (Table 2). From the FEM the team was able to calculate the maximum Von Mises stress, strain in multiple directions and maximum displacement for the sub-assemblies (Table 3-7). Figures of the finite element model FEM models Von Mises stress, and maximum displacement for each sub-assembly are shown in figure 11, 12, and 13. Pictures of other Finite Element Analysis results are in Appendix.

<table>
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<tr>
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<th>Force</th>
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<tbody>
<tr>
<td>Vertical Static Point Load</td>
<td>500 – 1050 (lbs)</td>
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<tr>
<td>Vertical Static load</td>
<td>21-57 (psi)</td>
</tr>
<tr>
<td>Shear Load</td>
<td>200 (lbs)</td>
</tr>
<tr>
<td>30 Degree Slope</td>
<td>500 (lbs)</td>
</tr>
<tr>
<td>30 Degree Slope Lateral</td>
<td>500 (lbs)</td>
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</table>

Table 2 - Loading Conditions and Forces

<table>
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<th>Maximum Displacement (in)</th>
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<td>500 (lbs.)</td>
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<tr>
<td>30 Degree Slope Lateral</td>
<td>500 (lbs.)</td>
<td>798.8</td>
<td>13.04</td>
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Table 3 – Von Mises Stress and Maximum Displacement for Inner Rim

Figure 13 - FEA with 4 Rivets hole with 1050 lbs. point load
### Table 4 – Von Mises Stress and Maximum Displacement for Gecko Tread

<table>
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<th>Loading Type</th>
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<th>Maximum Displacement (in)</th>
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<td>.2263 – .4751</td>
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<td>.4444 - 1.206</td>
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<td>Shear Load</td>
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<td>30 Degree Slope</td>
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<td>30 Degree Slope Lateral</td>
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### Table 5 – Von Mises Stress and Maximum Displacement for Circular Spring

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<th>Von Mises stress (ksi)</th>
<th>Maximum Displacement (in)</th>
</tr>
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<tbody>
<tr>
<td>Vertical Static Point Load</td>
<td>500 – 1050 lbs</td>
<td>133.5 – 280.4</td>
<td>.01865 – .03916</td>
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<tr>
<td>Vertical Static Load</td>
<td>1050 lbs</td>
<td>69.19</td>
<td>.02439</td>
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<tr>
<td>Shear Load</td>
<td>57 psi</td>
<td>4.056</td>
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### Table 6 – Von Mises Stress and Maximum Displacement for Hinged Center Hub

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<td>2.503e-005</td>
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<tr>
<td>Shear Load</td>
<td>57 psi</td>
<td>4.056</td>
<td>9.078e-004</td>
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### Table 7 – Von Mises Stress and Maximum Displacement for Outer Rim

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<tr>
<td>Vertical Static load</td>
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<td>.00092</td>
</tr>
<tr>
<td>30 Degree Slope</td>
<td>500 lbs</td>
<td>2040</td>
<td>Flat (37.26)</td>
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<tr>
<td>30 Degree Slope Lateral</td>
<td>500 lbs</td>
<td>1786</td>
<td>23.37</td>
</tr>
</tbody>
</table>
4.2 3-D MODEL TESTING

The team decided to test the 3-D printed model wheels at three different inclines, 20, 35, and 45 degrees. The team started the tests at 20 degrees and found that the wheel when placed as shown in figure 14, just rolled backwards because the wheels did not lock. At 20 and 35 degree inclines the wheel when placed laterally, as shown in figure 15, stayed in place. However, at 45 degrees the wheel moved slightly down the slope.

This testing was not the most accurate testing of the wheel’s traction because when the wheel was placed vertically along the incline the wheels turned and did not lock in place to allow for accurate reading of slip in this direction. Also the tread design did not have the same compression as the springs would have allowed for. However, the lateral testing showed more useful results. The tread seemed improved traversing across an incline compared to the previous lunar wheel team.

4.3 COMPRESSION TESTING

Two strain gauges, in x and y directions, were placed on one of the circular springs from the middle layer. The results are shown in appendix VI. The circular spring tested was 2” wide and 3.5” thick and was found to have a peak load of 1051.423lbf, peak stress of 0.2ksi, and has a modulus of 14.158 ksi. The test showed that the circular springs had the strength to support the weight of a 5000 lb. truck.

The results of the compression testing were compared to the values resulting from finite element analysis. The experimental values of deflection were 0.023 in, with a max deflection of 0.025 in, while the finite elemental analysis gave a
value of 0.0244 inches. The values were very close. The finite element analysis pictures for this test are shown in appendix VI, in figures 51 and 52.

5.0 DISCUSSION

5.1 PURPOSE OF THE PROJECT

The purpose of the Lunar Wheel Project is to develop a new efficient lunar/planetary wheel design that will be placed on a Lunar Rover Vehicle. The design will be able to incorporate an independent traction motor, and support a load of 5000 lbs. The wheel will be airless and flexible and have a designed tread that will provide traction on regolith while also being capable of traversing obstacles to navigate. The project design will resolve the clogging issues on the previous GM LRV wheel due to the zero atmosphere and low gravity field on the moon.

5.2 LIMITATIONS OF THE STUDY

The limitations of the project are the ability to test the regolith in the same gravitational state as the moon. The clogging testing is limited due to there being an atmosphere and not being in a vacuum state. There are limitations for the team to be able to experiment with extreme temperature fluctuations ranging from 40 K to 400 K to help represent some of the situations the lunar wheel will deal with in the lunar environment[3]. There is also no capability to test with high amounts of solar radiation to analyze the effect on the selected materials.

Other limitations of the study include only being able to produce and test a section of the wheel, about one-fourth of the design, due to a limited budget of $1800 and time constraint. The testing of the 3D printed models will test the basic design of the tread of the wheel but will not be the selected material or have the capabilities full scale models would have had. The original NASA competition was canceled so the tests had to be conducted on the beach which is not as close to the lunar environment as NASA’s testing environment.

5.3 PRESENT AND FUTURE IMPLICATIONS

Due to low budget, the team was not able to produce a full scale prototype. In the future, if finance permits, the team can produce a full scale wheel and test it under 5000 lbs. load. Lab testing on the full scale prototype as well as testing under lunar conditions will be conducted as if the competition is still going on.

6.0 CONCLUSION

Overall, the team did not produce a full scale wheel to test them on the 2000 Chevy Silverado. However, the team has successfully created a one-fourth wheel prototype as a final product. The team’s advisor was proud of the lunar wheel team and the progress they made over the past year. The conducted testing on the 3-D model of the tread design showed promising results in the wheel’s performance for lateral slippage, however, was not very useful for determining vertical slippage up an incline. The compression testing showed that the middle
spring layer met the required load. Overall, after all the testing that was performed on the components, actual and finite element analysis results, the design proved to be promising for further study.

7.0 APPENDICES

Appendix I – Spring Design

<table>
<thead>
<tr>
<th>Nomenclature</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>CRR</td>
<td>Compliance rolling resistance of FW suspension</td>
</tr>
<tr>
<td>$C_n$</td>
<td>Damping coefficient of unit #n</td>
</tr>
<tr>
<td>$D_n$</td>
<td>Displacement of unit #n</td>
</tr>
<tr>
<td>$f$</td>
<td>Vertical-mode natural frequency of a PSV with FW suspension</td>
</tr>
<tr>
<td>$k_L$, $c_L$</td>
<td>Effective rotational stiffness and damping of FW suspension in a magnitude of $L$, respectively</td>
</tr>
<tr>
<td>$k_{Ln}$, $c_{Ln}$</td>
<td>Effective rotational stiffness and damping of unit #n in a magnitude of $L$, respectively</td>
</tr>
<tr>
<td>$k_T$, $c_T$</td>
<td>Effective vertical stiffness and damping of FW suspension, respectively</td>
</tr>
<tr>
<td>$k_{Tn}$, $c_{Tn}$</td>
<td>Effective vertical stiffness and damping of unit #n, respectively</td>
</tr>
<tr>
<td>$k_X$, $c_X$</td>
<td>Effective longitudinal stiffness and damping of FW suspension, respectively</td>
</tr>
<tr>
<td>$k_{Xn}$, $c_{Xn}$</td>
<td>Effective longitudinal stiffness and damping of unit #n, respectively</td>
</tr>
<tr>
<td>$k_\alpha$, $c_\alpha$</td>
<td>Effective translational stiffness and damping of FW suspension in an angle of $\alpha$, respectively</td>
</tr>
<tr>
<td>$k_{\alpha n}$, $c_{\alpha n}$</td>
<td>Effective translational stiffness and damping of unit #n in an angle of $\alpha$, respectively</td>
</tr>
<tr>
<td>$K_n$</td>
<td>Stiffness of unit #n</td>
</tr>
<tr>
<td>$L_0$</td>
<td>Static vertical deflection</td>
</tr>
<tr>
<td>$P_n$</td>
<td>Power consumption of unit #n</td>
</tr>
<tr>
<td>$P_S$</td>
<td>Power consumption of FW suspension</td>
</tr>
<tr>
<td>$R$</td>
<td>Radius of the FW suspension</td>
</tr>
<tr>
<td>RPF</td>
<td>Rotational property factor of FW suspension</td>
</tr>
<tr>
<td>TPF</td>
<td>Translational property factor of FW suspension</td>
</tr>
<tr>
<td>$V$</td>
<td>Rover forward speed</td>
</tr>
<tr>
<td>$V_n$</td>
<td>Velocity of unit #n</td>
</tr>
<tr>
<td>$\xi$</td>
<td>Vertical-mode damping ratio of a rover with FW suspension</td>
</tr>
<tr>
<td>$\omega$</td>
<td>Angular velocity of $OB$</td>
</tr>
</tbody>
</table>

Keywords:
Planetary surface vehicles (PSVs);
Flexible-wheel (FW) suspension;
Compliance rolling resistance (CRR);
Appendix II – Gantt chart

APPENDIX III – OVERALL BUDGET

<table>
<thead>
<tr>
<th>Item</th>
<th>Amount</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steel Springs</td>
<td>$504.01</td>
</tr>
<tr>
<td>Steel Rods</td>
<td>$108.53</td>
</tr>
<tr>
<td>CNC work</td>
<td>$0</td>
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<tr>
<td>Aluminum Plates</td>
<td>$200.27</td>
</tr>
<tr>
<td>Aluminum Sheets</td>
<td>$451.33</td>
</tr>
<tr>
<td>Steel Tube</td>
<td>$61.55</td>
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<tr>
<td>Bolts and Rivets</td>
<td>$44</td>
</tr>
<tr>
<td>Miscellaneous</td>
<td>$430.31</td>
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<tr>
<td><strong>Total</strong></td>
<td><strong>$1,800</strong></td>
</tr>
</tbody>
</table>
APPENDIX IV – DESIGN CALCULATIONS

Stress $\sigma = 80,438.835 \frac{N}{m^2}$

Strain $\epsilon = 0.00000116747$

AXIAL LOADED DEFORMATION

$A = 0.75 \times 4.5 = 3.375 \text{ in}^2$

$E = 10,000 \text{ ksi}$

$L = 10 \text{ in}$

$P_{int} = \frac{V}{A} = \frac{700.0138 \text{ lbf}}{3.75 \text{ in}^2} = 186.6703 \text{ psi}$

$\Delta = \frac{P_{int}L}{AE} = \frac{186.6703 \text{ psi} \times 10 \text{ in}}{3.375 \text{ in}^2 \times 10,000,000 \text{ psi}} = 55.30972 \times 10^{-6} /\text{in}$

Shear $\tau = \frac{1050 \text{lbf}}{3.375 \text{ in}^2} = 311 \frac{\text{lbf}}{\text{in}^2} = 2,145,035.604 \frac{N}{m^2}$, 1 ft full load

Shear $\tau = 155.56 \frac{\text{lbf}}{\text{in}^2}$, 1 ft for half load

$\tau_{mx} = \left( \frac{V (\frac{h}{4})(\frac{bh}{2})}{(\frac{1}{12})bh^3} \right) = \left( \frac{V (0.75 \text{ in})(4.5 \text{ in} \times 0.75 \text{ in})}{(\frac{1}{12}) \times 4.5 \text{ in} \times (0.75 \text{ in})^3} \right) = 311.111 \frac{\text{lbf}}{\text{in}^2}$

$V = 700.0138 \text{ lbf}$, for 1 foot at full load

BEAM SHEAR

$\tau = \frac{VQ}{It}$

$Q = \bar{y}'A' = \left( \frac{1}{2} \right) \left( \frac{0.75^2}{2} \right) 4.5$

$Q = 0.6328 \text{ in}^3$

ALLOWABLE STRESSES

AEROSPACE INDUSTRY USUALLY USES F.S = 1.5
\[\sigma_{allow} = \frac{\sigma_{mx}}{F.S} = \frac{80438.835}{1.5} \frac{N}{m^2} = 53625.89 \frac{N}{m^2}\]

\[\sigma_{allow} = 0.0536 \text{ Mpa}, \text{1 ft at full load}\]

\[\tau_{allow} = \frac{\tau_{mx}}{F.S} = \frac{2145035.604}{1.5} \frac{N}{m^2} = 1430023.736 \frac{N}{m^2}\]

\[\tau_{allow} = 1.43 \text{ Mpa}, \text{1 ft at full load}\]

MARGIN OF SAFETY

\[M.S = \frac{\text{Failure load}}{\text{Design load} \times \text{Design S.F}} - 1\]

Failure Load = \(\sigma_{\text{Fail}} = 245 \text{ Mpa}\)

Design Load = \(\sigma_D\)

\[\sigma_D = \frac{F}{A} = \frac{1050 \text{ lbf}}{4.5 \text{ in} \times 10 \text{ in}} = \frac{4670.633N}{2(0.1143 m \times 0.254 m)} = 80438.84 \left(\frac{N}{m}\right) = 80438.84 \text{ (N/m)}\]

\[M.S = \frac{245 \text{ MPa}}{0.08043854 \text{ MPa} \times 1.5} - 1 = 2030.53, \text{Very High}\]

\(\rightarrow\) This was modeled as a spring flat plate

\(\rightarrow\) Alternate \(M.S\) Calculation

\[M.S = S.F - 1 = 1.5 - 1 = 0.5 = 50\%\]

The 0.5 margin suggests the design will perform up to 0.5 times more than the design load and so the design passes.

APPENDIX V – MATERIAL DATA FOR ALUMINUM 6061-T6

Subcategory: 6000 Series Aluminum Alloy; Aluminum Alloy; Metal; Nonferrous Metal

Composition Notes:
Aluminum content reported is calculated as remainder.
Composition information provided by the Aluminum Association and is not for design.

Key Words: al6061, UNS A96061; ISO AlMg1SiCu; Aluminium 6061-T6, AD-33 (Russia); AA6061-T6; 6061T6, UNS A9601; ISO AlMg1SiCu; Aluminium 6061-T651, AD-33 (Russia); AA6061-T651
### TABLE 8 – COMPONENTS’ PERCENT WEIGHT

<table>
<thead>
<tr>
<th>Component</th>
<th>Wt. %</th>
<th>Component</th>
<th>Wt. %</th>
<th>Component</th>
<th>Wt. %</th>
</tr>
</thead>
<tbody>
<tr>
<td>Al</td>
<td>95.8 - 98.6</td>
<td>Mg</td>
<td>0.8 - 1.2</td>
<td>Si</td>
<td>0.4 - 0.8</td>
</tr>
<tr>
<td>Cr</td>
<td>0.04 - 0.35</td>
<td>Mn</td>
<td>Max 0.15</td>
<td>Ti</td>
<td>Max 0.15</td>
</tr>
<tr>
<td>Cu</td>
<td>0.15 - 0.4</td>
<td>Other, each</td>
<td>Max 0.05</td>
<td>Zn</td>
<td>Max 0.25</td>
</tr>
<tr>
<td>Fe</td>
<td>Max 0.7</td>
<td>Other, total</td>
<td>Max 0.15</td>
<td></td>
<td></td>
</tr>
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</table>

### TABLE 9 – MATERIAL PROPERTIES

<table>
<thead>
<tr>
<th>Physical Properties</th>
<th>Metric</th>
<th>English</th>
<th>Comments</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density</td>
<td>2.7 g/cc</td>
<td>0.0975 lb/in³</td>
<td>AA; Typical</td>
</tr>
</tbody>
</table>

**Mechanical Properties**

<table>
<thead>
<tr>
<th>Property</th>
<th>Metric</th>
<th>English</th>
<th>Comments</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hardness, Brinell</td>
<td>95</td>
<td>95</td>
<td>AA; Typical; 500 g load; 10 mm ball</td>
</tr>
<tr>
<td>Hardness, Knoop</td>
<td>120</td>
<td>120</td>
<td>Converted from Brinell Hardness Value</td>
</tr>
<tr>
<td>Hardness, Rockwell A</td>
<td>40</td>
<td>40</td>
<td>Converted from Brinell Hardness Value</td>
</tr>
<tr>
<td>Hardness, Rockwell B</td>
<td>60</td>
<td>60</td>
<td>Converted from Brinell Hardness Value</td>
</tr>
<tr>
<td>Hardness, Vickers</td>
<td>107</td>
<td>107</td>
<td>Converted from Brinell Hardness Value</td>
</tr>
<tr>
<td>Ultimate Tensile Strength</td>
<td>310 MPa</td>
<td>45000 psi</td>
<td>AA; Typical</td>
</tr>
<tr>
<td>Property</td>
<td>Value</td>
<td>Details</td>
<td></td>
</tr>
<tr>
<td>--------------------------------</td>
<td>-------------</td>
<td>-------------------------------------------------------------------------</td>
<td></td>
</tr>
<tr>
<td>Tensile Yield Strength</td>
<td>276 MPa</td>
<td>40000 psi, AA; Typical</td>
<td></td>
</tr>
<tr>
<td>Elongation at Break</td>
<td>12 %</td>
<td>12 %, AA; Typical; 1/16 in. (1.6 mm) Thickness</td>
<td></td>
</tr>
<tr>
<td>Elongation at Break</td>
<td>17 %</td>
<td>17 %, AA; Typical; 1/2 in. (12.7 mm) Diameter</td>
<td></td>
</tr>
<tr>
<td>Modulus of Elasticity</td>
<td>68.9 GPa</td>
<td>10000 ksi, AA; Typical; Average of tension and compression. Compress</td>
<td></td>
</tr>
<tr>
<td>Notched Tensile Strength</td>
<td>324 MPa</td>
<td>47000 psi, 2.5 cm width x 0.16 cm thick side-notched specimen, K_t =</td>
<td></td>
</tr>
<tr>
<td>Ultimate Bearing Strength</td>
<td>607 MPa</td>
<td>88000 psi, Edge distance/pin diameter = 2.0</td>
<td></td>
</tr>
<tr>
<td>Bearing Yield Strength</td>
<td>386 MPa</td>
<td>56000 psi, Edge distance/pin diameter = 2.0</td>
<td></td>
</tr>
<tr>
<td>Poisson's Ratio</td>
<td>0.33</td>
<td>0.33, Estimated from trends in similar Al alloys.</td>
<td></td>
</tr>
<tr>
<td>Fatigue Strength</td>
<td>96.5 MPa</td>
<td>14000 psi, AA; 500,000,000 cycles completely reversed stress; RR</td>
<td></td>
</tr>
<tr>
<td>Fracture Toughness</td>
<td>29 MPa-m½</td>
<td>26.4 ksi-in½, K_c, TL orientation.</td>
<td></td>
</tr>
<tr>
<td>Machinability</td>
<td>50 %</td>
<td>50 %, 0-100 Scale of Aluminum Alloys</td>
<td></td>
</tr>
<tr>
<td>Shear Modulus</td>
<td>26 GPa</td>
<td>3770 ksi, Estimated from similar Al alloys.</td>
<td></td>
</tr>
<tr>
<td>Shear Strength</td>
<td>207 MPa</td>
<td>30000 psi, AA; Typical</td>
<td></td>
</tr>
<tr>
<td>Electrical Properties</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Electrical Resistivity</td>
<td>3.99e-006 ohm-cm</td>
<td>3.99e-006 ohm-cm, AA; Typical at 68°F</td>
<td></td>
</tr>
<tr>
<td>Thermal Properties</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>CTE, linear 68°F</td>
<td>23.6 μm/m-°C</td>
<td>13.1 μin/in-°F, AA; Typical; Average over 68-212°F range.</td>
<td></td>
</tr>
<tr>
<td>CTE, linear 250°C</td>
<td>25.2 μm/m-°C</td>
<td>14 μin/in-°F, Estimated from trends in similar Al alloys. 20-300°C.</td>
<td></td>
</tr>
<tr>
<td>Specific Heat Capacity</td>
<td>0.896 J/g-°C</td>
<td>0.214 BTU/lb-°F</td>
<td></td>
</tr>
<tr>
<td>Thermal Conductivity</td>
<td>167 W/m-K</td>
<td>1160 BTU-in/hr-ft²-°F, AA; Typical at 77°F</td>
<td></td>
</tr>
</tbody>
</table>
### TABLE 10 – STAINLESS STEEL AIRCRAFT CABLES

<table>
<thead>
<tr>
<th>Diameter in Inches</th>
<th>Construction</th>
<th>Minimum Break Strength/Lbs.</th>
<th>Rated Load/Lbs.</th>
<th>WT. Per 100 Ft./Lbs.</th>
</tr>
</thead>
<tbody>
<tr>
<td>3/16</td>
<td>7 x 7</td>
<td>480</td>
<td>100</td>
<td>.8</td>
</tr>
<tr>
<td>5/32</td>
<td>7 x 7</td>
<td>920</td>
<td>180</td>
<td>1.6</td>
</tr>
<tr>
<td>1/8</td>
<td>7 x 7</td>
<td>1,700</td>
<td>340</td>
<td>2.8</td>
</tr>
<tr>
<td>1/8</td>
<td>7 x 19</td>
<td>1,760</td>
<td>350</td>
<td>2.9</td>
</tr>
<tr>
<td>5/32</td>
<td>7 x 7</td>
<td>2,600</td>
<td>520</td>
<td>4.3</td>
</tr>
<tr>
<td>5/32</td>
<td>7 x 19</td>
<td>2,400</td>
<td>480</td>
<td>4.5</td>
</tr>
<tr>
<td>3/16</td>
<td>7 x 7</td>
<td>3,700</td>
<td>740</td>
<td>5.2</td>
</tr>
<tr>
<td>3/16</td>
<td>7 x 19</td>
<td>3,700</td>
<td>740</td>
<td>5.5</td>
</tr>
<tr>
<td>7/32</td>
<td>7 x 19</td>
<td>5,000</td>
<td>1,000</td>
<td>8.6</td>
</tr>
<tr>
<td>1/4</td>
<td>7 x 19</td>
<td>6,400</td>
<td>1,280</td>
<td>11.0</td>
</tr>
<tr>
<td>5/32</td>
<td>7 x 19</td>
<td>7,800</td>
<td>1,560</td>
<td>13.9</td>
</tr>
<tr>
<td>5/16</td>
<td>7 x 19</td>
<td>9,000</td>
<td>1,800</td>
<td>17.3</td>
</tr>
<tr>
<td>3/8</td>
<td>7 x 19</td>
<td>12,000</td>
<td>2,400</td>
<td>24.3</td>
</tr>
</tbody>
</table>
APPENDIX VI

Figure 16 – Gecko Tread Maximum displacement 21 psi

Figure 17 – Gecko Tread Vertical Static Load 57 psi
Figure 18 – Gecko Tread Maximum displacement 57 psi

Figure 19 – Gecko Tread Vertical Static point 500 lbs.
Figure 20 – Gecko Tread Maximum Displacement 500 lbs.

Figure 21 – Gecko Tread Vertical Static Point Load 1050 lbs.
Figure 22 – Gecko Tread Maximum Displacement 1050 lbs.

Figure 23 – Shear force
Figure 24 – Maximum Displacement Shear Force

Figure 25 – Gecko Tread 30 deg. Angle load
Figure 26 – Gecko Tread angle displacement

Figure 27 – Gecko Tread 30 deg. angle Lateral load
Figure 28 – Circular Spring Vertical Static load 500 lbs.

Figure 29 – Circular Spring displacement 500 lbs.
Figure 30 – Circular Spring 1050 lbs.

Figure 31 – Circular Spring displacement 1050 lbs.

Figure 32 – Circular Spring Shear Force
Figure 33 – Circular Spring Shear Displacement

Figure 34 – Inner Rim Vertical Static Load

Figure 35 – Inner Rim Vertical Static displacement
Figure 36 – Inner Rim 30 deg. slope Lateral

Figure 37 – Inner Rim 30 deg. slope lateral displacement

Figure 38 – Inner Rim 30 deg. Slope
Figure 39 – Inner Rim 30 deg. angle slope displacement

Figure 40 – Outer Rim Vertical Static load
Figure 41 – Outer Rim Vertical Static load Displacement

Figure 42 – Outer Rim Static Load

Figure 43 – Outer Rim 30 deg. angle slope
Figure 44 – Outer Rim 30 deg. Slope displacement

Figure 45 – Outer Rim 30 deg. Slope Lateral

Figure 46 – Outer Rim 30 deg. Slope Lateral displacement
Figure 47 – Hinged Center Hub with visuals of Von Mises Stress around the Rod

Figure 48 – Hinged Center Hub Displacement
Figure 49 – Hinged Center Hub Shear Force

Figure 50 – Center Hinged Hub Displacement
APPENDIX VII

Assembly of 1/4th Wheel Instructions

Bottom up Assembly

1. Connect the bottom center hinge to the base with three 9/16th diameter x 3 inch long bolts
2. Attach the 10 degree stops to the sides of the top and bottom hinges using ten 0.5 inch diameter x 1.438 inch long bolts
3. Connect bottom center hinge to top center hinge with pin while placing springs in the middle
4. Three washers are to be put on the end of the pin with the bolt to secure the pin. Bolt size: 7/16 diameter x 1 inch long
   (Possible problem tightening the bolt on the pin due to lack of leverage on other side of the pin, note that the other end of the pin could have been made to fit a socket)
5. Drill through holes in the center of the inner semi-circle to lineup with predrilled holes in the top of the center hinge
6. Use two 0.5 inch diameter x 1.438 inch long bolts to connect the inner semi-circle to the top center hinge

Top Down Assembly

1. All upper pull rivets are to be pulled from the outer most distance from the center of the wheel (The smoother surface of the rivet will then be on the outer most surface with the weaker, more vulnerable to deformation side on the inner side to help prevent damage from terrain. This will make it less exposed to the ground, and will also make the wheel look more appealing.)
2. Place upper spring restraints on diamond tread in the correct pattern and mark where the holes will need to be drilled for the rivets. These holes will be .13” Diameter holes for the .125” Diameter, 0.501” - 0.625” gage length rivets.
3. Drill holes
4. Insert the cable into the groove of the upper spring restraint and through the center hole
5. Apply adhesive to the wire spring and or recessed circular cut out and place the spring in the recessed circular cut out pulling the cable through the center of the spring
6. Feed the cable through the center of the bottom spring restraint and place the cable in the groove heeding towards the closest edge of the diamond
7. Apply adhesive to the wire spring and or recessed circular cut out in the bottom spring restraint
8. Make sure the two blocks are parallel to one another and the curvatures match the surfaces they are going to be attached to
9. Do this for all the wave springs
10. Wait 24 hours for the adhesive to cure
11. Place the springs in the positions that they will be in on the upper semi-circle when they are to be riveted and secure them until all holes are marked in the upper semi-circle. Use the diamonds to line up the correct position of the springs.
12. Mark where the holes for the rivets will need to be in the upper semi-circle and drill holes
13. Line up the tube springs with the holes on the upper semi-circle to mark holes (not all holes will be able to be marked this way; there will still be one hole that will need to be added to the upper semi-circle and the circular springs that is 0.55” towards the largest amount of tubing and drill holes).
14. The bottom holes in the tube springs do not have to be the same as the top but they can be; if they are all the same there will be a few holes that will be covered up by the hinge assembly so it is recommended using three evenly spaced holes. On the tubes that would still have holes that interfere with the hinge, the holes will have to be shifted towards the outside of the wheel and space will need to be reduced between holes.

15. Place the wire luge on the end of cable on the bottom spring restraint and loop a small portion back on itself and crimp with a set of pliers.

16. Keeping the cable pulled tightly, wedged in between the diamond tread and the top of the upper wire restraint, keep the luge on the outside edges of the diamonds to prevent any interferes with other luges or springs and pull the rivets as previously stated.

17. Pull cable tightly to the point that the spring compresses ever so slightly and crimp the luge on the end of the cable.

18. Place the circular springs on the inner semi-circle mark were holes need to be drilled and drill holes.

19. Rivet the inner semi-circle to the bottom of the circular springs, pulling the rivets from underneath these will be the .251 to .375 gage length rivets.

20. Celebrate and admire your creation.
8.0 REFERENCES


