An Approach to Designing Passive Self-Leveling Landing Gear with Application to the Lunar Lander

Troy B. Rippere* and Gloria J. Wiens*

Abstract

Once the lunar lander has touched down on the moon problems can occur if the crew module is not level. To mitigate, compliant landing gear provide a solution that would allow the module to be leveled once it has landed on some ground slope. The work presented here uses compliant joints, or flexures, for each leg of the module and optimizes the mechanics of these flexures such that the module can be passively leveled over a range of landing slopes. Preliminary results suggest that for landing on a slope of up to 12 deg the effective slope of the module can be reduced to a maximum of 2.5 deg.

Introduction

During the next lunar mission one of the challenges that will be faced is the possibility of landing on a slope of up to twelve degrees. Among other potential issues caused by landing on a slope, such as difficulties while offloading cargo, is the concern of "fly-out" problems during ascent from the lunar surface. By considering the lunar module and its four deployable legs as a single spatial mechanism, the legs can be designed in a novel manner with the objective of passive self-leveling in mind.

Additionally, to avoid issues associated with lubricating joints this problem can be approached using compliant joints, or flexures, which are thin members that provide the relative rotation between two adjacent rigid members through bending. Previous work has shown how flexures can be approximated as linear torsion springs with stiffness k [1]. From this, a variety of techniques can be applied to find appropriate flexure stiffness of each joint so that the combination of the weight of the lunar lander and the resistance to deflection from the flexures will effectively reduce the relative angle of the module with respect to the horizontal plane over the range of potential ground slopes.

Background

It should be noted that the approaches presented here model the module with four identical equally spaced legs. Furthermore, each leg is designed to have a joint connecting it to the module, a joint connecting it to the landing pad at the bottom of the leg, and a third joint spaced somewhere between the first and second joints.

<u>Assumptions</u>

For these analyses, the following initial assumptions were made:

1. The landing area of the lunar surface can be modeled as a plane. In other words, there are no craters or boulders that would affect a single leg.
2. There is a uniform probability of landing on any ground slope between zero and twelve degrees.
3. There is a uniform probability of landing at any twist angle. Here twist angle is defined as rotation of the module in the ground plane. A 0 deg twist angle means that the line between one set of two opposing pads is orthogonal to the local gravity field, or that those two pads are level. A 45 deg twist angle means that line between two adjacent pads is orthogonal to the local gravity field, or that they are level. It follows that the other set of two adjacent pads will also be level with respect to each other.

* University of Florida, Gainesville, FL

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4. Upon landing, the landing pads will remain fixed relative to one another as they were in the undeflected configuration; landing impact will not affect the placement of individual pads on the ground.

Leg Numbering System
Figure 1 and Figure 2 show the leg numbering system that is used to describe joint orientations.

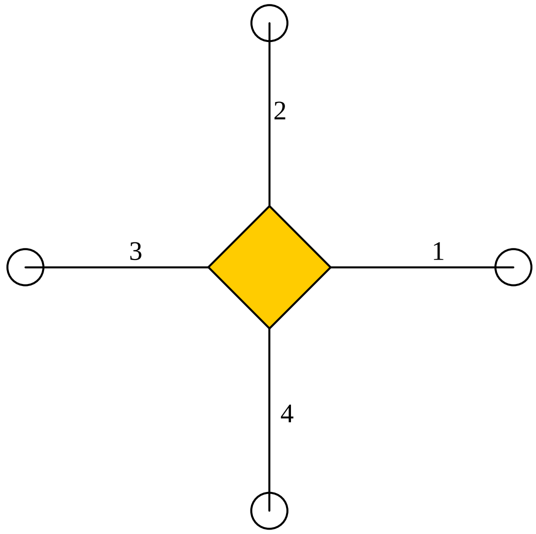

Figure 1. Top view of module with legs numbered

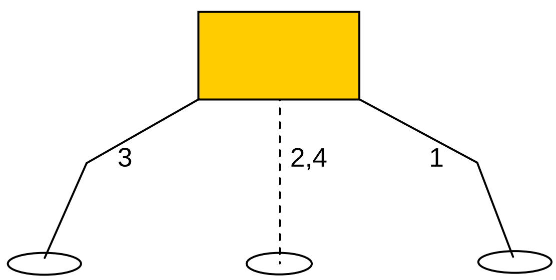

Figure 2. Front view of module with legs numbered

Joints
As mentioned previously, each leg will have a joint that connects the leg to the module. For the analyses presented here these joints will be flexures whose bending axes are perpendicular to the plane formed by that leg and the opposing leg when the landing gear is in its undeflected configuration. For example, in Figure 2 the flexure axes of legs 1 and 3 will be out of the page while the flexure axes of legs 2 and 4 will be parallel to the page. For each leg, this flexure will be called flexure *f*.

Additionally, the joints that connect the legs to the landing pads will be spherical joints and thus will not have any resistance to rotation.

Next, the joint in the middle of each leg will be the flexure equivalent of a universal joint with perpendicular axes. This is accomplished by having two flexures serially connected as depicted in Figure 1. One of the bending axes will be parallel to the bending axis of the joint which connects the leg to the module while the other axis will be in the plane formed by that leg and the opposing leg. For each leg, the flexure with the bending axis parallel to that leg's bending axis of flexure *f* will be called flexure *d* while the serially connected flexure will be called flexure *e*.

Because it has been shown that flexures can be modeled as linear torsion springs, subsequent modeling will assume that the flexures are traditional revolute or universal joints with torsion springs to resist rotation about the bending axes. For the model of each analysis, it is assumed that the bending axes of flexures *d* and *e* both pass through the same point.

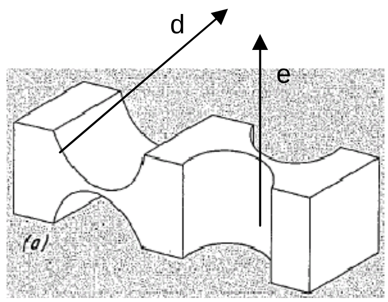

Figure 1. Universal flexure created by serially connecting two flexures [2]

Analysis

Two-Dimensional Analysis

The first analysis performed was a two-dimensional analysis in which the plane that was analyzed was that which is formed by two opposing legs in the undeflected configuration. The active flexures for the planar analysis of legs 1 and 3 are shown in Figure 4. It can be seen that legs 2 and 4 together form a single middle leg that has one flexure. Additionally, to account for the change in length of legs 2 and 4 projected onto the plane due to the out of plane motion, the middle leg is of variable length. This variable length is depicted in Figure 4 by a double headed arrow.

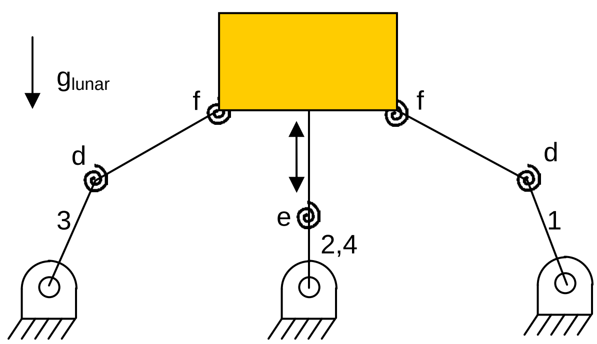

Figure 2. Joints and links for planar analysis

The three spring stiffnesses will be determined as follows:

1. Give an initial guess of spring constants for flexures *d*, *e*, and *f*.
2. For a given ground slope find the angle of each joint as described below.
3. Using static balancing equations find the required lunar weight of the module such that the combination of the weight of the vehicle and the resistance to bending in the springs will put the module in static equilibrium. For a given set of spring stiffnesses there exists one unique solution of required weight that will hold the module in static equilibrium at the desired level configuration.
4. Repeat steps 2 and 3 for n steps of ground slope between 0 deg and 12 deg.
5. Calculate the average required weight from the n steps in step 4.
6. Use forward divided differences with steps 4 and 5 to estimate the gradient of average required weight with respect to the stiffness of each spring.

7. Use steepest descent to find the set of stiffnesses (*d*, *e*, *f*) that minimizes the difference between the calculated average required weight and the known lunar weight of the module using the estimate of the gradient from step 6 for each iteration.

This planar mechanism has seven bodies (including the ground) and eight revolute joints resulting in two degrees of freedom. Addition of the variable length of the middle leg gives a total of three degrees of freedom.

Because the mechanism has three degrees of freedom, three choices regarding its configuration must be made. The first choice, which should be the most obvious, is that the module be level with respect to the local gravity field. The second choice is the length on the adjustable leg which will account for any ground slope going into the page. Lastly, the angle of flexure *d* on the uphill leg is assumed to be constant at an angle of 180 deg. This assumption remains valid for any configuration that would otherwise try to extend the angle beyond 180 deg if the leg is designed with a mechanical stop preventing a greater angle. An example of a configuration after these three choices is shown in Figure 5.

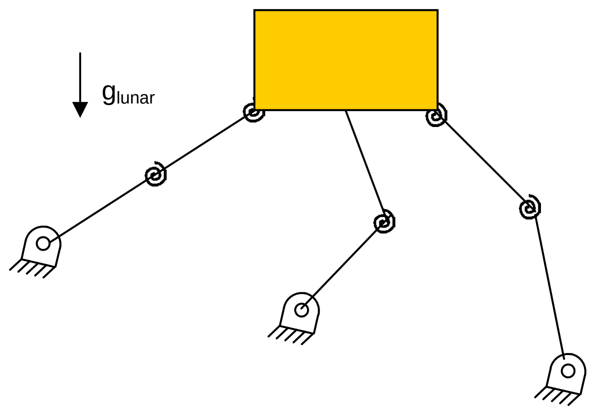

Figure 5. Configuration of leveled module

Three-Dimensional Analysis

The second analysis performed was a three-dimensional analysis. This analysis looks at the mechanism as a whole instead of just a projected plane while also providing a slightly different approach to finding the optimized spring stiffnesses.

This approach is as follows:

1. Give an initial guess of spring constants.
2. For a given ground slope and twist angle find the mechanism configuration as described below.
3. Using static balancing equations find the required lunar weight of the module such that the combination of the weight of the vehicle and the resistance to bending in the springs will put the module in static equilibrium. For a given set of spring stiffnesses there exists one unique solution of required weight that will hold the module in static equilibrium at the desired level configuration.
4. Use forward divided differences with step 3 to calculate the gradient of required weight with respect to the stiffness of each spring. While there exists a unique solution of required weight for a set of spring constants, for the inverse problem when given the required weight there exists a plane of solutions for the set of spring stiffnesses. The calculated gradient is the normal vector to all of the planes of solutions for any given required weight.
5. Calculate the point that is closest to the initial guess of spring constants and on the plane of solutions when the required weight is equated to the given weight of the module.
6. Using the point found in step 5 and the normal vector, calculate the equation of the plane.
7. Repeat steps 2 through 6 for m steps of twist angle between 0 and 45 deg. Because of symmetry any twist angle can be modeled by a twist angle between 0 deg and 45 deg.
8. Repeat step 7 for n steps of ground slope between 0 deg and 12 deg.

9. Using linear least squares, find the point that minimizes the L^2-norm of the vector of the distance between this point and each plane found in step 8. This point is defined to be the set of optimum spring stiffnesses.

This spatial mechanism has ten bodies (including the ground), four spherical joints, four universal joints, and four revolute joints. This gives six degrees of freedom, which means that the module itself can be positioned and oriented in any manner within its workspace.

Similar to the planar analysis, it is first necessary to find the angles of each joint to then find the optimum spring stiffnesses. Also similar to the planar analysis, the orientation is assumed to be level. This reduces the degrees of freedom by two because the module must be level about two orthogonal horizontal axes. The third choice of configuration is that the module is at some assumed height which is below its undeflected height on a level surface. Essentially this is saying that on a level surface, the weight of the module would cause some sort of deflection in the joints and would sink by some amount. This analysis assumes that the height of the module will be at 90% of its undeflected height.

After these two constraints on orientation and one constraint on position, three degrees of freedom still remain. It is assumed that the preferred leveled equilibrium configuration of the module will be one which minimizes the deflections in the flexures, where here minimization is defined as minimization of the L^2-norm of the vector of the twelve flexure deflections. Using the Matlab fmincon function, these remaining degrees of freedom, two of which are position in the horizontal plane and the third is the rotation in the horizontal plane, can be found which minimize the deflections.

Results and Conclusions

For both the two-dimensional and 3-dimensional analysis, the flexure has the dimensions as depicted in Figure 6. The stiffness for flexure can be approximated in terms of its geometric dimensions as follows:

$$k \approx \frac{2Ebt^{5/2}}{9\pi R^{1/2}} \tag{1}$$

Two-Dimensional Analysis

The optimum spring stiffnesses as determined by the first analysis were found to be:

$$\begin{bmatrix} k_d \\ k_e \\ k_f \end{bmatrix} = \begin{bmatrix} 3.841\times10^3 \\ 5.562\times10^4 \\ 5.840\times10^4 \end{bmatrix} \frac{N\cdot m}{rad} \tag{2}$$

From this and the spring stiffness approximation of equation (1), the dimensions of the flexures as shown in Figure 6 are on the order of:

$$\begin{bmatrix} b \\ t \\ R \end{bmatrix} = \begin{bmatrix} 0.1 \\ 0.01 \\ 0.5 \end{bmatrix} m \tag{3}$$

Using these spring stiffnesses and the given weight of the module a static analysis was performed for a range of ground slopes between 0 deg and 12 deg. The maximum angle of the module in this range was 2.5 deg at a ground slope of 12 deg.

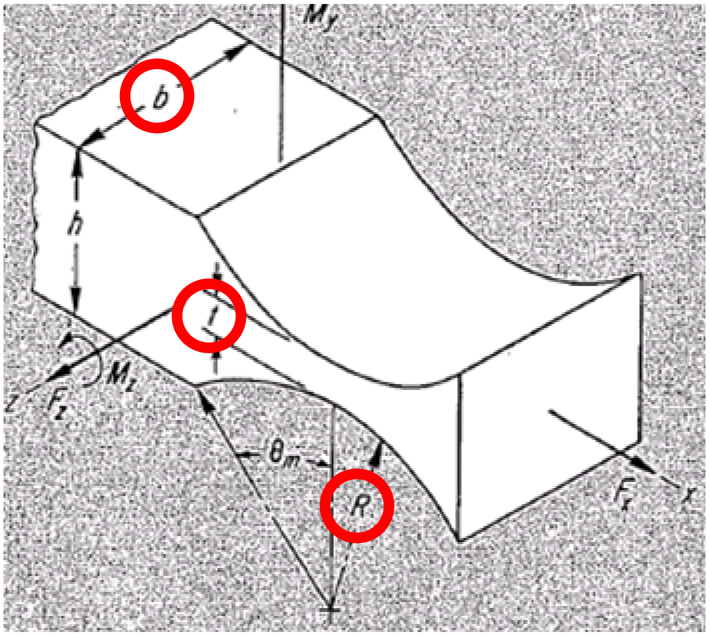

Figure 3. Flexure dimensions [2]

Three-Dimensional Analysis
The optimum spring stiffnesses as determined by the second analysis were found to be:

$$\begin{bmatrix} k_d \\ k_e \\ k_f \end{bmatrix} = \begin{bmatrix} 1.077 \\ 1.811 \\ 2.162 \end{bmatrix} \times 10^5 \frac{N \cdot m}{rad} \tag{4}$$

From this and the spring stiffness approximation of equation (1), the dimensions of the flexures as shown in Figure 6. Flexure dimensions are on the order of:

$$\begin{bmatrix} b \\ t \\ R \end{bmatrix} = \begin{bmatrix} 0.2 \\ 0.01 \\ 0.05 \end{bmatrix} m \tag{5}$$

The static analysis that was performed for the calculated spring stiffnesses for the two-dimensional analysis has not yet been performed for the three-dimensional analysis. This will be completed in future work.

Comparing the results of the two analyses it is seen that the optimum stiffness as found by the two-dimensional analysis is only about one-fourth as that obtained from the three-dimensional analysis. This discrepancy is likely due to the different assumptions of the two analyses, specifically the length of the adjustable leg and the fixed angle of the uphill leg in the two-dimensional analysis. Additionally, because of the assumption that the bending axes of flexures *d* and *e* pass through the same point, it follows that the flexure radius *R* must be small because the flexures are actually serially connected. Because of this, the three-dimensional analysis might prove more accurate because of this much smaller dimension.

Future Work

In addition to the three-dimensional static analysis for the calculated spring stiffnesses there are a few other .tasks that can improve the robustness of this method. These include calculating and minimizing the stresses in the flexures and taking into account the change in module weight once the payload has been unloaded. Additionally, minimizing the effect of single-leg disturbances, such as one leg landing on a boulder or in a small crater or being displaced by landing impact, would increase the robustness. Also, for more insight into the actual bending mechanics, a finite-element analysis might prove more accurate than the linear spring approximation. Lastly, looking at different flexure shapes, as described in [3], might provide a better flexure system.

Acknowledgments

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