Optimization of a Lunar Pallet Lander Reinforcement Structure using a Genetic Algorithm

Adam O. Burt[1] and Patrick V. Hull[2]
National Aeronautics and Space Administration, Marshall Space Flight Center, Alabama, 35812

This paper presents a design automation process using optimization via a genetic algorithm to design the conceptual structure of a Lunar Pallet Lander. The goal is to determine a design that will have the primary natural frequencies at or above a target value as well as minimize the total mass. Several iterations of the process are presented. First, a concept optimization is performed to determine what class of structure would produce suitable candidate designs. From this a stiffened sheet metal approach was selected leading to optimization of beam placement through generating a two-dimensional mesh and varying the physical location of reinforcing beams. Finally, the design space is reformulated as a binary problem using 1-dimensional beam elements to truncate the design space to allow faster convergence and additional mechanical failure criteria to be included in the optimization responses. Results are presented for each design space configuration. The final flight design was derived from these results.

Nomenclature

GA	= Genetic Algorithm
FEA	= Finite Element Analysis
RESOLVE	= Regolith and Environment Science and Oxygen and Lunar Volatiles and Extraction
RP	= Resource Prospector
X-TOOLSS	= eXploration Toolset for the Optimization of Launch and Space Systems
EC	= Evolutionary Computing
NASA	= National Aeronautics and Space Administration
MoS	= Margin of Safety
LPL	= Lunar Pallet Lander
λ	= Beam Location Weighting Parameter
Fx	= Primary Natural Frequency in the X direction
Fy	= Primary Natural Frequency in the Y direction
Fz	= Primary Natural Frequency in the Z direction
M_r	= Modal Effective Mass
$[M]$	= Mass Matrix
$\{\varepsilon\}$	= Eigenvector Scaling Factors

I. Introduction

In this paper, a system level spacecraft design optimization will be presented to determine a preliminary design for a Lunar Pallet Lander (LPL). A Genetic Algorithm is used at all levels to automate the search of the design space. First, a concept optimization trade study is performed to determine what type of structural concept has the highest probability of meeting design requirements. The types of concepts searched include: grid stiffened, flanged grid stiffened, open grid stiffened and metal-core-metal composite. The next stage of design was then to size and determine lay out of the LPL primary structure and finally determine the secondary and support structure required to produce positive margins of safety for the lay out design for the stiffened sheet metal concept.

[1] Aerospace Engineer, Space Systems Department, Thermal & Mechanical Analysis Branch/ES22 Marshall Space Flight Center, AL 35812
[2] Technical Assistant; Space Systems Department; Mechanical Design Branch/ES21; Marshall Space Flight Center, AL 35812

A. Mission Background

The design of the LPL is in support of the NASA Resource Prospector Mission to carry the RESOLVE instrument package to a lunar pole. The challenge in this design was to have a concept pallet lander that could be flexible to design changes and accommodate a fast-paced design schedule. The starting concept involved utilizing a thin, flat deck allowing the rover to simply roll off on the lunar surface. The LPL needs to be assessed to survive launch loads and meet the frequency requirements specified by the launch vehicle.

The proposed launch vehicle is the Falcon 9. From the Falcon 9 Payload Planners Guide it was determined that frequencies of the payload should be kept above 25 Hz to prevent detrimental coupling with the launch vehicle. From this it was decided that the optimization should try to achieve designs that have natural frequencies in the range of 35 Hz. This would provide additional robustness and allow for mass increases in the LPL subsystem design development without forcing a re-design of the LPL support structure.

Launch loads and factors of safety are included by providing a Margin of Safety (MoS) response in the objective function. In this way, primary structural requirements were all included in the optimization objective function to try and produce the most viable LPL structure design.

B. eXploration Toolset for the Optimization of Launch and Space Systems (X-TOOLSS)[6,7,8]

All levels of optimization presented here are driven by Genetic Algorithms (GAs). X-TOOLSS is an evolutionary computation (EC) toolset that includes GAs and other types of ECs. X-TOOLSS is currently being developed by NASA Marshall Space Flight Center and university partners. Genetic algorithms are based on the "survival of the fittest" concept. They develop optimal solutions for prospective designs by eliminating weak candidate designs. The power of EC techniques lies in their ability to discover unique, innovative, and often non-intuitive designs by thoroughly interrogating an entire design space. Furthermore, ECs drive the whole system to a global optimum and are able to avoid becoming "trapped" in local optima.

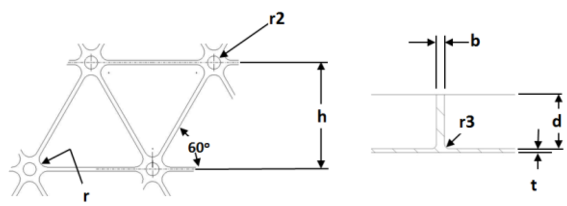

Figure 1. Example of Isogrid Stiffening

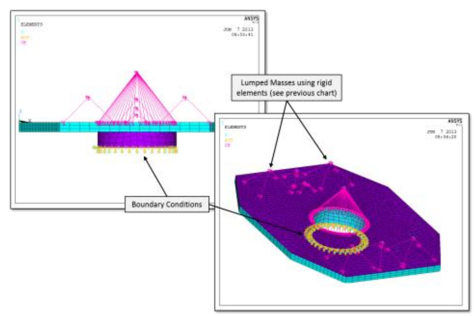

Figure 2. Optimization Model

II. Preliminary Concept Optimization

A. Structural Concept Optimization

The preliminary sizing tool used here was developed using GAs and ANSYS. The overall goal of this preliminary design approach is to create a mechanism to narrow down the disproportionate design space to the best construction method for further detailed optimization and analysis. Preliminary sizing of the LPL was approached using four possible construction methods: (1) Grid stiffened, (2) flanged grid stiffened, (3) open grid stiffened, and (4) metal-core-metal composite.

The first three options are grid stiffened structures, in essence integrally machined stiffened construction with a triangular configuration shown in Figure 1. Also checked in the following preliminary sizing was the allowable stress for each construction method. It became clear that the stress acting on the pallet lander was negligible and was not considered further in the objective function here. Metal-core-metal composite was used in this analysis; while there are many other types of composites available (i.e., solid laminate, laminate-core-laminate, etc.) those are not addressed in this work.

The preliminary structural optimization of a pallet lander deck requires assumptions of boundary conditions and lumped masses. The assumptions used in this analysis are shown in Figure 2.

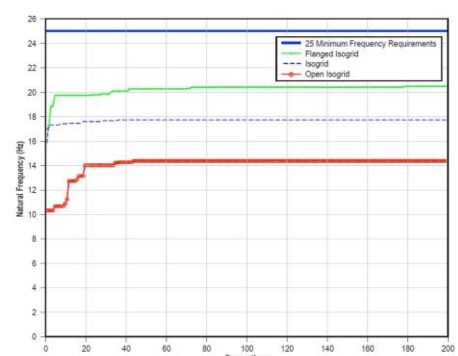

Figure 3. Concept Frequency Comparisons

This analysis uses a GA that deals with a mixture of mutation rate, population size, and stochastically selected initial states. This helps facilitate identifying construction types that will likely produce solutions closer to the global optimum. There are many options for applying evolutionary optimization..

The frequencies for each design option are given in the figures 3 and 4. The major difference between the isogrid options is the differing stiffness for each design (i.e., by geometry, the flanged isogrid design has a much higher stiffness than the open isogrid design). The composite design was optimized not for a frequency but for mass, as shown in Figure 4. As is the nature of composite structures, they possess high stiffness for relatively low weight; therefore, achieving the minimum natural frequency was easily obtainable absent of any optimization. With that in mind, it was decided to pursue the lightest weight structure and thus, the individual curve for the composite design. Notice on figure 3 the solid line at 25 Hz; that is minimum natural frequency required. It is also clear from figure 3 that none of the isogrid designs met the minimum required natural frequency. There are stiffening options that could be used to possibly achieve a higher natural frequency (e.g., higher frequencies and lower mass can be achieved with local geometry refinement of the grid stiffened designs).

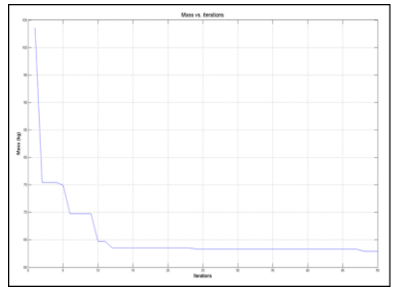

Figure 4. Composite Optimization Evolution

Concluding the preliminary analysis, it is clear that integrally stiffened structures are the least suited for high frequencies and low mass. While the composite or reinforced thin structures proves to be the lightest approach based on the preliminary analysis. The examples presented in this section demonstrate the versatility of this design method and its potential use in other applications where exceedingly large design space and complex geometries mandate the use of global optimization methods. Also, with the institution of additional design parameters, a more optimum solution exists. An illustration of this is shown as the paper progresses, i.e. parameterizing the stiffness and geometry throughout the structure.

B. Sheet Metal Grid Stiffened Concept Preliminary Study

Based on the results of the preliminary study, it was decided that the sheet metal concept would indeed be the best solution to pursue as it would be both low cost, easier to manufacture and provided design change flexibility. The composite option was not pursued as it does not provide enough flexibility to change required in this design.

The approach utilized was incremental so as to truncate the design space for the final optimization. While this may eliminate the possibility of determining a globally optimal solution, it does significantly reduce the design space and guarantees a more attractive solution from a manufacturing and assembly perspective. The first step was to determine an appropriate nominal gauge thickness and beam cross section for the LPL support structure. Based on availability and cost, it was determined that channel section would provide the cheapest and highest flexibility option. By inspection it can be determined that the highest strength, and greatest impact on primary frequencies would be to maximize the web depth of the beam. For this design, that number is ultimately constrained by the launch vehicle and landing gear.

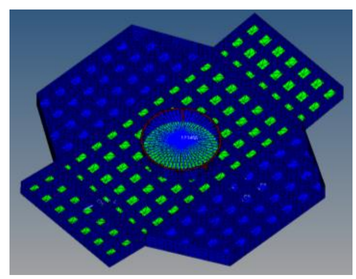

Figure 5. Baseline Design. *The baseline design is based upon the minimum pocket dimension generated uniformly over the lander*

Next, the nominal thickness needed to be chosen. A gauge optimization study provided insight into the effect of thickness on natural frequencies and ultimate strength in the beams. It was determined that from this study that the thinner beams had the highest the potential to produce the highest natural frequencies as well as maintain the ultimate strength requirements for the LPL. From this, the lead designer selected a commercially available channel stock section to reinforce the LPL. This cross section is then utilized for the remainder of the optimization and eliminates that variable from the design space.

Finally, a baseline design needed to be assessed. The best starting solution would be to reinforce the LPL structure using the minimum pocket size and determine the mass and frequency associated with the design. This would provide insight to determine which designs are improvements. Figure 5 shows this baseline concept. The mass of this design was obtained and used as a metric to compare against in the optimization. The natural frequencies for this concept are in the range 15-20Hz in the X, Y and Z directions. Because of this it is clear that non-uniform pocket geometry must be explored to determine a design that meets the natural frequency requirements.

III. Non-Uniform Pocket Sizing with a Continuous Design Space

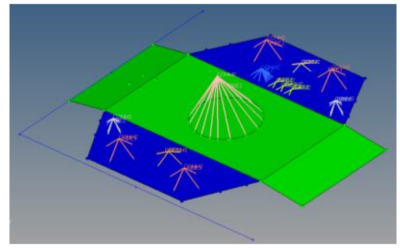

Figure 6. Pallet Lander Starting Design Space

The concept optimization described in the previous section yielded two results. First, it provided a starting nominal thickness and secondly, it showed that a uniform pocket design would not be adequate and that looking into tuning the structure to be of non-uniform pockets would be required to raise the natural frequencies further.

To accomplish this, a tcl/tk script was written for Hypermesh v12.0[3] to take the general design space (the top deck of the LPL) and automate creating the reinforcing beams in a way that could be readily parameterized. The script used a baseline geometry shown in Figure 6. Next, lines were translated along the top deck and surfaces extruded from them creating baseline geometry for the reinforcing structure. Next, all surfaces were remeshed and point masses representing the primary propulsion and avionics systems were equivalenced to output a NASTRAN input deck for a normal modes analysis. X-TOOLSS is then used to drive the parameterized script in a batch process that creates the unique beam design each iteration.

A. Design Space

To trim the total design space, a minimum pocket size was determined based upon tooling and intended manufacturing processes at the time of the design. This resulted in the determination that no more than 15 beam across the lateral and longitudinal directions would be allowed. These two parameters are used as design variables and were given a discrete, integer range from 2 to 15. Secondly, each beam was assigned a floating point value, λ, given a range from -0.8 to 0.8. This provided a percentage that the beam would be moved from the nominal position (with a negative sign indicating direction along an axis). In this way, the design space is as large as possible without violating manufacturing constraints. From this the total number of parameters in the design space is 30. The calculation for the position of each beam is given as:

$$Lateral_{Postion_i} = Lateral_{position(i-1)} + \frac{Lateral_{length}}{N}(1+\lambda_i) \quad (1)$$

$$Long_{position_i} = Long_{postion(j-1)} + \frac{(Long_{length})}{M}(1+\lambda_j) \quad (2)$$

Where N is the number of lateral beam, M is the number of longitudinal beam and i and j represent each beam in the lateral and longitudinal directions, respectively. Figure 7 shows a sample design after the tcl/tk scripts has been executed and the top surface (lander deck) has been removed.

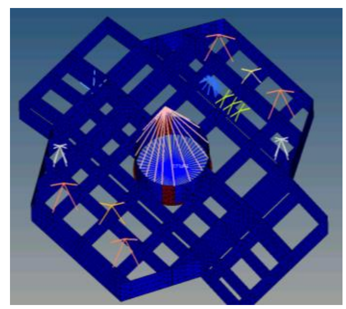

Figure 7. Sample Design After Beam Generation. *This figure shows a sample lander design after the beams have been generated and the top skin removed. This mesh is generated each step in the analysis.*

As seen in Figure 7, this method is able to create unique, pocket geometries that will allows stiffness and mass to be tuned allowing the LPL to meet the launch vehicle natural frequency requirements.

B. Objective Function

The objective of the optimization is to raise the primary natural frequencies of the lander above 35 Hz. To do this Modal Effective Mass is used to identify globally acting modes. Modal Effective Mass was obtained from the NASTRAN results and is defined as:

$$M_r = \{\varepsilon\}^T[M]\{\varepsilon\} \quad (3)$$

Where ε scaling factor for the eigenvectors and M is the mass matrix as given by Ref. 4. This parameter gives an indication of what direction the mass for a given mode is moving and guidance as to how much of that mass participates in a given mode. Because the reinforcement design is thin and deep, there are many low frequency panel modes discovered by a normal modes analysis. Therefore, modal effective mass provides a useful metric to determine frequencies that are more likely to have a substantial structural response.

The objective function is written such that the natural frequency requirements are treated as a system identification problem to target the desired natural frequency. In this way, constraints can be communicated through the objective function. The objective function used for this optimization is given as:

$$obj = min\left[\left(\frac{\frac{(Fx-35)^2}{a^2}+\frac{(Fy-35)^2}{b^2}+\frac{(Fz-35)^2}{c^2}}{3}\right)+\frac{mass}{mass\_target}\right]$$

Where

$$\begin{aligned} &if\ Fx \geq 35\ Hz : a = 100, if\ Fx < 35\ Hz\ a = 1 \\ &if\ Fy \geq 35\ Hz : b = 100, if\ Fy < 35\ Hz\ b = 1 \\ &if\ Fx \geq 35\ Hz : c = 100, if\ Fz < 35\ Hz\ c = 1 \end{aligned} \tag{4}$$

and the target frequency is 35 Hz. This provides a fitness function surface shape as a modified parabaloid as shown in Figure 8 for a given value of *Fx*. The minimum solution for this problem is *Fx*, *Fy* and *Fz* equal to 35 Hz and the target mass reached which yields a value of 1.0. Frequencies above or below 35 Hz will result in a higher value, but the function can be weighted such that higher solutions are not penalized as greatly as the lower ones. This is controlled by the values of a, b and c after reaching the 35 Hz objective. For this case, a,b and c were chosen to be 1 or 100. Therefore solutions have a decreasing sensitivity as 35 Hz is approached in each direction and no longer drive the solution if higher frequency solutions are found. Once the optimal frequencies are reached the only way to further minimize the objective function is to reduce mass.

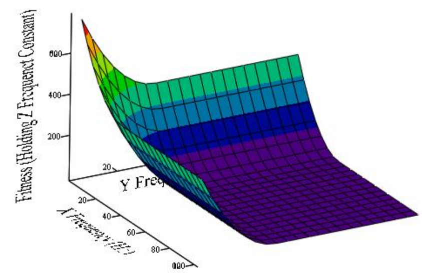

Figure 8. Parabaloid Fitness Function Surface

C. Results

The final solution was found in 883 generations and obtained a fitness value of 1.107. The lowest primary natural frequency for this solution is 34.8 Hz and corresponds to motion in the axial direction; all other directions exceed the requirement. Figure 9 shows the bottom view of the final design.

This solution was compared to results obtained using a full design space as described in Section IIB. By using a non-uniform pocket arrangement the natural frequencies were able to be increased to meet the requirements of the launch vehicles as well as reduce the overall mass of the spacecraft. By using less beams, cost due to manufacturing and weight of the secondary structure has also been reduced.

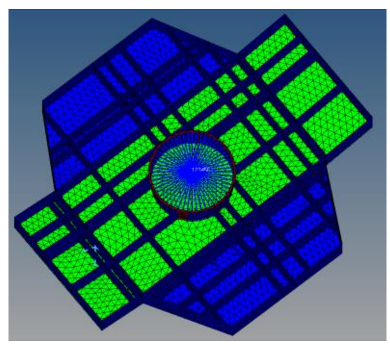

Figure 9. Final Optimal Design

There are two shortcomings to this approach. First, as mentioned previously, the cross section chosen was channel section. However, for modeling purposes on the web length was used to represent the beam and the flanges were not included in the model. Therefore, the predictions with regards to natural frequency are likely lower as adding the flange increases the stiffness. This is not detrimental to the process as this is a conservative approach given the objective. Secondly, because the number of beams are allowed to vary for each iteration and the total length of the vectors λ_i and λ_j are fixed, repeated solutions are introduced into the optimization. For example, if the sample design only has 10 beams, in each direction, then *i* and *j* greater than 10 have no impact on the objective function. This dramatically increases the size of the design space and inhibits quicker convergence.

The latter discussion provided impetus toward re-formulating the design space as binary problem with a fixed number of beams

IV. Non-Uniform Pocket Sizing with a Binary Design Space

The results from the previous study were used to begin initial LPL concept design. However, the optimization process was updated to include structural responses as well as frequency and mass and re-stated as a binary problem to reduce the size of the design space.

This step in optimization occurred later in the design process, after several subsystems acquired more definition. Because of this, model properties such as mass and stiffness are different. As can be seen in Figure 10, the design space has been updated to include the addition of a cut out between the ramps as well as an additional point mass.

A. Design Space Setup

1. Binary Formulation

The lander structural design space was reformulated as a binary problem. The total number of possible beams was decided up front to be 20 in the lateral direction and 20 in the longitudinal direction. This was based on the minimum reasonable pocket size to allow for easier assembly and resulting mesh quality for FEA. The surface shown in **Error! Reference source not found.** was trimmed in a process similar to the method described in Section III. Next, each new line was meshed as a 1D beam element creating the overall design space shown in Figure 10. Each beam has a design variable associated with it that can have the value of either 1 or 0 (corresponding to masked or unmasked). During the optimization a script is executed to determine which beams have been removed and then removes the corresponding elements lying along that beam, this process is described in more detail in Section *2.*

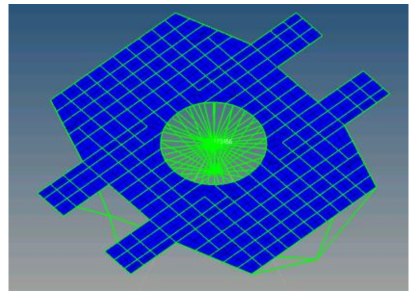

Figure 10. Initial 1D Beam Element Binary Design Space

2. Beam and Bay Identification

At each stage in the optimization it must be determined what elements should be removed from the model based upon which beams have a design parameter corresponding to a value 0 for that iteration. To do this, every beam element in the model is search and associated with a beam id, based upon their coordinates. As all beams are orthogonal in the design space, it is a simple matter to detect the vector of the axial direction of the beam. The next step is simply to mask those elements associated with the beam, creating the new design. Once this step is completed an algorithm is used to detect the bay generated for this candidate design. A bay is defined as the length in between two intersecting beams as is shown in Figure 11 using 3D visualization of the 1D element. The dimensions of these bays are used to determine the supporting structure design later discussed in section D. To generically determine each bay in the lander, the maximum and minimum node along each beam is identified. This is used as a starting and stopping criteria for the connectivity march. The goal is to search for every intersection along a beam, and then obtain the elements in between each intersection, forming the entire bay. Once all the elements in a bay have been identified, a bay id number is given to that bay and a key-value pair constructs is created to retain the data. The element information obtained from each bay is then used to determine the structural responses. This analysis is performed every iteration.

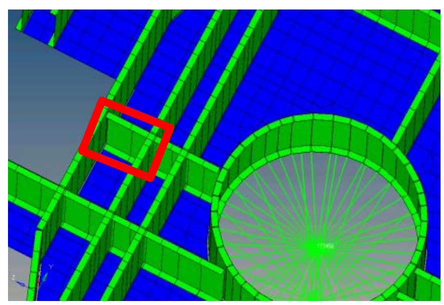

Figure 11. Example Bay. *Bay information will be used to determine the design of the reinforcing secondary structure*

B. Optimization Flow

A multilevel approach was used for this final optimization process. First, the genetic algorithm suite provided by X-TOOLSS is used to drive the beam design by turning off and on different beams. For every candidate design, the primary natural frequencies, structural margins of safety, rivet and stiffener requirements, and mass are determined. These are then compiled into an objective function to be returned to the genetic algorithm. This method provides a top level design and uses closed form solutions to characterize the strength and secondary stiffening requirements of that design. It also captures additional mass needed to make the candidate design viable from a strength perspective. The optimal design should have frequencies above 35 Hz, with the minimum number of rivets and stiffeners and with positive margins of safety.

C. Finite Element Analysis Results Post Processing

One advantage to using 1-Dimensional elements to represent the supporting beams is that the post processing of results is simplified. As all of the elements in each bay have been identified, the maximum axial load, shear load in both planes, bending moments in both planes and torsion loads can be determined by reading the NASTRAN output file generated for each design using linear static analysis. Using these loads, a shear flow analysis of the cross section is performed to extract the peak line loads in the beam web and beam flanges for each bay[2]. These peak lines loads can then be used to design the supporting structure (i.e, rivets and stiffeners for each bay of the lander design).

The primary natural frequencies are determined the same way as described in Section III. B.

D. Additional Structural Responses

To support the stiffened sheet metal concept, several responses in addition to just material allowable strength are required. Primarily, stiffeners will be potentially required for each bay (see Figure 11) to provide resistance to shear buckling. Also included is a determination of the minimum rivet diameter and number of rivets needed to attach both the skin of the top deck of the LPL to the beams structure as well as attach the stiffeners to the channel section flanges. These three parameters were directly solved for each bay and were included the mass response as well as the margin of safety response.

1. Rivet Size and Spacing Design

For the reinforcement structure, it was decided that rivets would be the appropriate choice for the bulk of the joints present in the LPL. The number of rivets in the lander will drive cost on assembly as well as total assembly time. The minimum number of rivets is determined uniquely for each bay of the LPL structure based upon the design methods presented by Ref. 2 and Ref. 5.

To find the minimum rivet diameter, the maximum rivet spacing for the smallest diameter rivet is determined based upon the load in the flange. If this spacing is less than 4 times the diameter of the rivet, then the diameter of the fastener is increased to the next size (this is a general rule of thumb).

To solve for the required rivet spacing, two parameters are required. First the shear allowable for the rivet must be calculated or provided and compared against line load in the in flange or web. Secondly, as a lower bound for spacing, inter-rivet buckling must be taken into account. Using these two parameters, a maximum and minimum spacing is determined. The maximum spacing is chosen for each bay, as this will minimize the number of rivets. If the maximum value is less than the minimum value, a negative margin of safety is carried as a response. When looking at the final design, this would indicate that riveting may not be an adequate joint design for that area and should be handled in the detailed design work.

Once rivet spacing is determined, the bay length is divided by the spacing to determine the number of rivets required, rounding to the next nearest integer. This number is included in the mass response of that candidate design. Finally, to determine the margin of safety, the number of rivets in a given bay is divided by the bay length to determine the true spacing (this will be a different value than previously found due to rounding). A margin of safety is written against this value.

This same process is also repeated to find the minimum number of rivets to attach the stiffeners to the beams. Instead of bay length, the web length of the beam is used to determine the correct number of rivets.

2. Stiffener Design

The primary failure mode for the LPL beam structure is buckling to due to shear. The assumptions in the loads analysis is that the bending moment is reacted into the flanges only, thus placing the beam into a state of pure shear. The shear stress in each bay is compared against the critical shear stress for a flat plate[9] to determine if additional stiffening is required to prevent buckling.

If this is the case, the proper stiffener spacing is determined based upon empirical equations provided by Ref. 1. Ref. 1 proposes a limiting value for the buckling factor, K based upon the stiffener spacing as well as the moment of inertia of the stiffener. Once the minimum stiffener spacing is chosen, the required stiffener moment of inertia can be directly solved based upon the spacing, modulus of elasticity and flexural rigidity of the web. More detail can be found in Ref. 1 and will not be presented here. Once the stiffener spacing is determined, it is divided by the bay length, to determine the number of stiffeners needed to reinforce a bay. This calculation assumes a linear distribution of stiffeners.

The stiffeners used are assumed to be comprised of an angle section. By making this assumption, the dimensions of this angle can be directly solved based upon the moment of inertia determined in the previous calculation. Although there are a variety of options for stiffening panels, with this simplification, the design space is truncated and is more favorable for detailed design.

3. Other Margins of Safety

The previous section discusses several design aspects required for each iteration to become closer to a viable solution. There are several other margins of safety needed to characterize the structural integrity of the design concept. This includes ultimate, yield and shear strength for the web and flange of the beam as well as the stiffener, shearout, and bearing for rivets going from flange to skin as well as from stiffener to beam web, local buckling of the flange of the channel section, crippling of the channel section and stiffener, sheet wrinkling and inter-rivet buckling. A Margin of Safety is developed for each of these parameters and is included in the objective function.

As has been demonstrated, each solution is not only being assessed for natural frequency and mass, but detailed strength characteristics have been established to help determine the viability of the design solution.

C. Objective Function

All structural responses, mass and primary frequencies are included in the object function. For this optimization, a new objective function was generated to include the structural responses. The objective function is given as:

$$obj_{MoS} = \sum_{i}^{m}\sum_{j}^{n}\left(-MoS_{ij} + MoS\_Target\right)\Phi(-MoS_{ij} + MoS\_Target)$$

$$Obj_{Frequency} = \frac{\frac{(Fx-35)^2}{a^2} + \frac{(Fy-35)^2}{b^2} + \frac{(Fz-35)^2}{c^2}}{3}$$

$$Obj_{Mass} = \frac{mass}{mass\_target}$$

$$Obj = \min(Obj_{MoS} + Obj_{Frequency} + Obj_{Mass}) \tag{5}$$

Where Φ represent the Heaviside step function.

The final objective function is simply to minimize the sum of the three secondary objectives. If all MoS are positive and the frequencies meet their requirements then the only remaining objective will be to minimize mass. In this way solutions, converging toward this goal are rewarded while solutions that exceed requirements are not rewarded or punished.

For the MoS objective function, m is the total number of bays in the lander design (this value will change for every candidate design), n is the total number of margins calculated for each bay (n remains constant). This objective function will act to push the margins of safety to positive values equal to the MoS target. Once each individual margin obtains a value greater than the MoS target, the value is set to zero thus removing the sensitivity of the design to that parameter and lowering the total sum in the objective function. Figure 12 shows a plot representative of the objective value as a function of the MoS. In addition, this formulation also acts to minimize the total number of bays in the lander (i.e reducing mass). Because it is a non-weighted summation, as fewer bays are presents, the margin of safety sum becomes less, thus becoming a more attractive solution. Therefore this one parameter helps to produce two desired outcomes; minimal mass and positive margins of safety. Once all margins of safety are positive they no longer have any impact on the objective function.

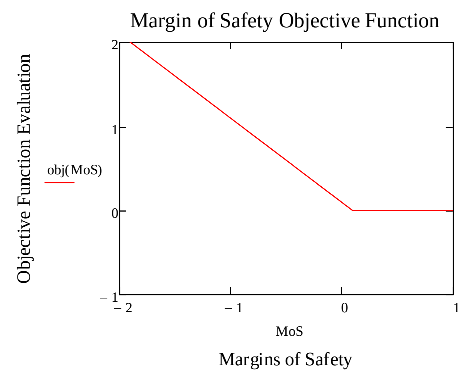

Figure 12. MoS Objective Function

Equation 4 is used as an objective function for the primary natural frequency. A surface plot with *Fx*_held constant is shown in Figure 8. Because the frequency objective function is parabolic, that objective will have priority over the other responses, with that priority decreasing as the required frequencies are approached. Once all frequencies meet the requirements, the only remaining methods to minimize the objective function are to minimize mass and increase the margins of safety to achieve the target value.

D. Results

The minimum fitness found at the end of the optimization run was 21.567, with minimum frequency obtained equal to 27.3 Hz, which is below the desired 35 Hz value. This is primarily due to the design mass increase encountered due to the higher fidelity design in subsystems. However, the optimization did produce a similar solution to the first optimization giving a higher confidence that the preliminary concept is working toward a good solution and the optimum frequency is still above the 25 Hz requirement as specified by the launch vehicle. Figure 13 shows the design that yielded the minimum fitness value.

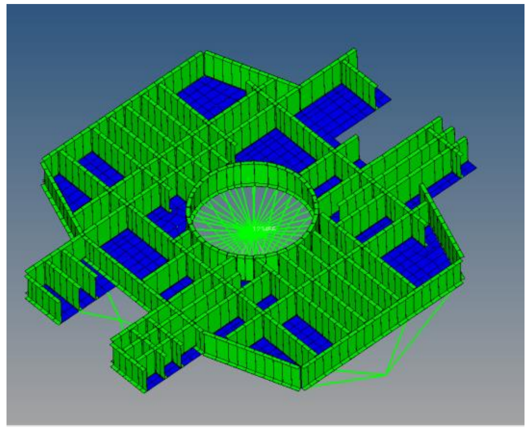

Figure 13. Design Produced by Optimization

The minimum margin of safety is -0.967 and corresponds to local buckling of the flange of the channel section. With the majority of the margins of safety being positive, this is a good staring design from a static strength perspective. The negative margins of safety can be communicated back to the designer as the areas that will need the most significant improvement during the detailed design process.

E. Comparison of Results

Two different optimization methods have been used to determine the optimal pocket geometry by operating beam placement. Figure 9 and Figure 13 show the results of the two optimization methods. While the fitness functions are not objectively comparable due to difference in mass properties, commonalities in the two designs can clearly be seen.

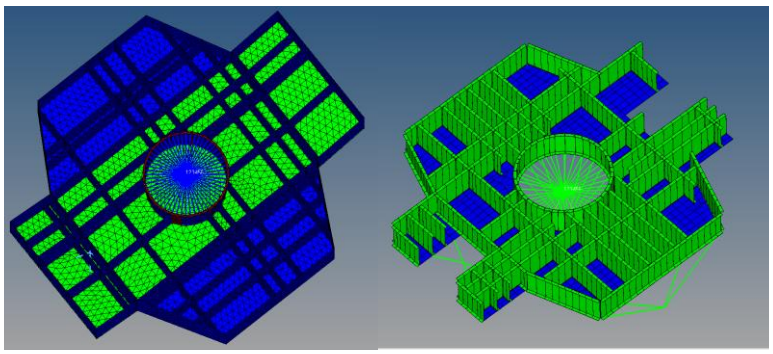

Figure 14. Comparison of Results

First, both designs have produced a tight grouping of beams along the radius of the central ring. The beams are located directly underneath the propellant tanks and provide a central load path to the central ring (running in the lateral direction). In the first optimization, 3 beams are present and in the second method 5 are present. The additional beams are likely present in the second method as margins of safety are taken into account. Secondly, each solution shows a beam present, attaching into the corners of the lander near the ramps (running in the longitudinal direction) and at least two beams in the lateral directions creating smaller pockets closer to the center.

V. Conclusion

F. Final Lander Structure Design

Based on the results of this optimization, a final lander structure that met frequency requirements and had well characterized strength margins was developed. The final lander design was derived from the optimized results. The final design created by the optimization lacked symmetry, which is undesirable from a manufacturing and structural robustness stand point. Therefore, the final design adhered to the final optimization as close as possible, but maintained symmetry. This final design became the base line concept for the LPL structure and was carried forward into the preliminary design cycle.

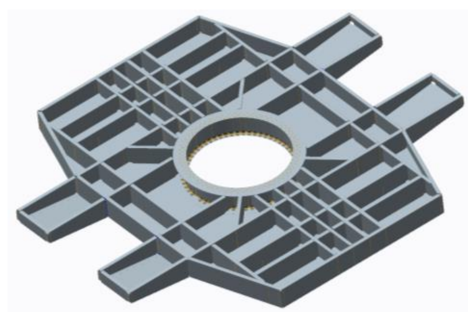

Figure 15. Final Pallet Lander Structure Design

The changes to from the optimal solutions acted to lower the primary frequencies, but as the design continues this can be addressed with higher fidelity analysis.

G. Discussion and Future Work

Two methods for optimization were discussed in this paper. The first represented a way to find the optimal beam placement using a continuous design space, while the latter showed how to reformulate the problem in terms of a binary space and include more structural responses.

Using modal effective mass as a way to determine the primary natural frequency has proven to be successful for this design effort. However, during the optimization it was noted that many of the modes are very low in terms of percentage participation and at or below 5% when compared to the total mass. In general, this makes using modal effective mass very sensitive, depending on where the next mode is located. For example a solution could show 5 Hz with 5% mass as the primary frequency, with the next highest being 30 Hz at 4%. Future work for this process should determine a more rigorous method for characterizing the primary natural frequencies when the number of modes in a frequency range is very high and effective mass percentage is similar. Future work could also include modifying the 1D mesh optimization and parameterization flow back to a continuous design space to see if better results could be obtained for this method.

As noted earlier, another desirable outcome for manufacturing purposes is to have a solution that maintains symmetry about both planes. An additional step that had to be taken from the optimized design to starting concept design was to create this symmetry which leads to the starting concept being a sub optimal solution. An interesting future task would to explore adding symmetry constraints and compare the resulting designs.

One benefit of performing optimization in this way (directly operating on the mesh), is that the analyst is left with a function model of the concept design. This reduces model setup time and allows the analyst or designer to begin immediately honing in the optimized design into a more realistic solution

In conclusion, the processes presented in this paper represent two distinct methods to optimize the LPL support structure. While what has been presented here is specific to the Lunar Pallet Lander, this process could extend elsewhere in industry where non-uniform structural reinforcement is desired to gain weight savings over other designs.

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