Team Orbital

RPEP

Reduced Profile Electronics Packaging

FINAL REPORT
ME486

Mechanical Engineering Senior Design Project

College of Engineering and Natural Sciences
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1.0 Introduction

Northern Arizona University's College of Engineering and Natural Sciences requires that all students graduating with a degree in mechanical engineering complete a capstone design course. This design experience is meant to provide a valuable aid to understanding the engineering profession prior to entrance into said profession.

Four Mechanical Engineering students make up the design team dedicated to addressing the following design issue: Design an avionics housing for use in rockets designed to launch small satellites into space. The client for this project is Mr. Mark Whiting at Orbital Sciences Corporation, Launch Systems Group, in Chandler, Arizona. Mr. Whiting works as an electronics packaging supervisor at Orbital in Chandler. The current preferred avionics housing design is a brand new evolution of a past design. Despite its flexibility, the size of the design proved to be an issue on certain rocket designs. As such, Orbital required a design that decreased the appropriate dimensions, while optimizing numerous design parameters.

Our design team approached this problem in five steps:
Specifications and requirements gathering
Definition of Design Philosophy
Research
Design
Analysis
Through iterative implementation of the above five steps, our design team has designed an electronics packaging that reduces the head height of the current design while optimizing the specified parameters.

2.0 Specifications

Before any design problem can be properly approached, a definition of the problem is needed. Through correspondence with Orbital, a firm grasp on the problem was gained. The original requirements document from Orbital can be seen in Appendix B.

2.1.0 Hard Design Requirements

Orbital defined a set of hard design requirements that were absolutely necessary for the design.

1) The unit head height must be under 5 inches total. Head height is a particular dimension referring to the height of the modules with cable strain and electromagnetic interference (EMI) backshells of the connectors added on. The EMI backshell reduces electromagnetic interference on the connector interface. A quick hand analysis by Mark Whiting at Orbital estimates that current cable strain is about five inches. This includes an assumed cable diameter of 0.5 inches, an EMI backshell of 1.69 inches. The cable must go straight for 2 cable diameters before bending at a radius of 4 diameters to avoid excessive stress in the cable. Figure 2.1 below shows an example of an EMI backshell.

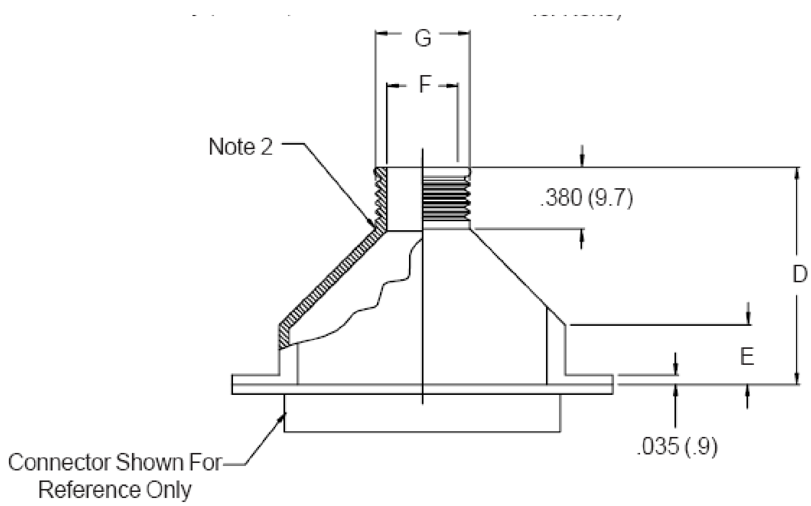
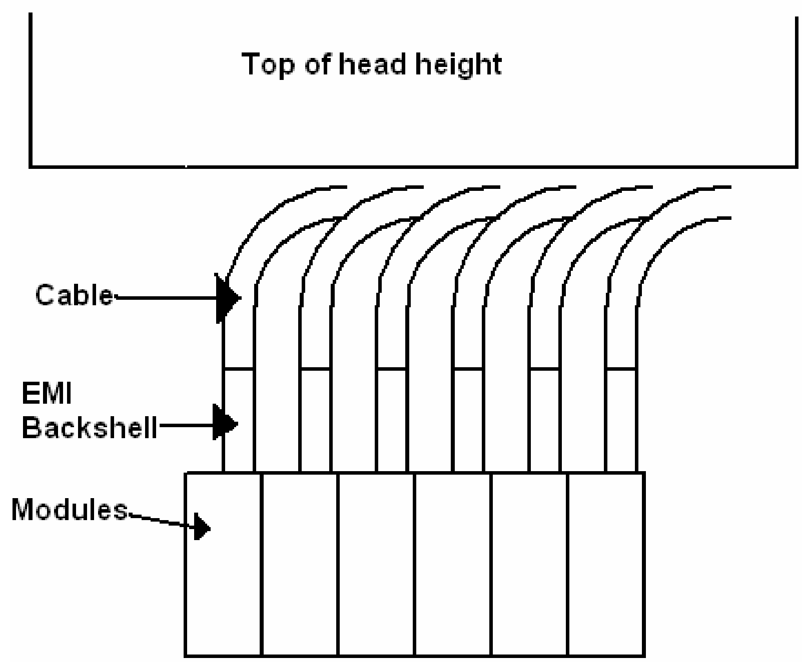


Figure 2.1: EMI Backshell

Below in Figure 2.2 is a sketch of what the head height with cable strain entails.



Note: Most of the head height is the cable strain and Emi backshell.
Figure 2.2: Head Height Illustration

The top of the head height is a limitation due to other rocket components. Essentially, the head height requirement requires our design team to reduce the current design's head height by at least 3 inches, to a total of 5 inches.

2) The new design must accommodate at least six modules. A module is currently a set of two boards (one a function board and one a BUS board) that serve a particular avionics purpose.

2.2.0 Optimization Requirements

Orbital has provided a list of nine optimization requirements for the project. While not absolutely necessary for the success of the design, these requirements should be optimized as much as possible for a more efficient design.

Board outline and connector locations should be as shown below in Figure 2.3.

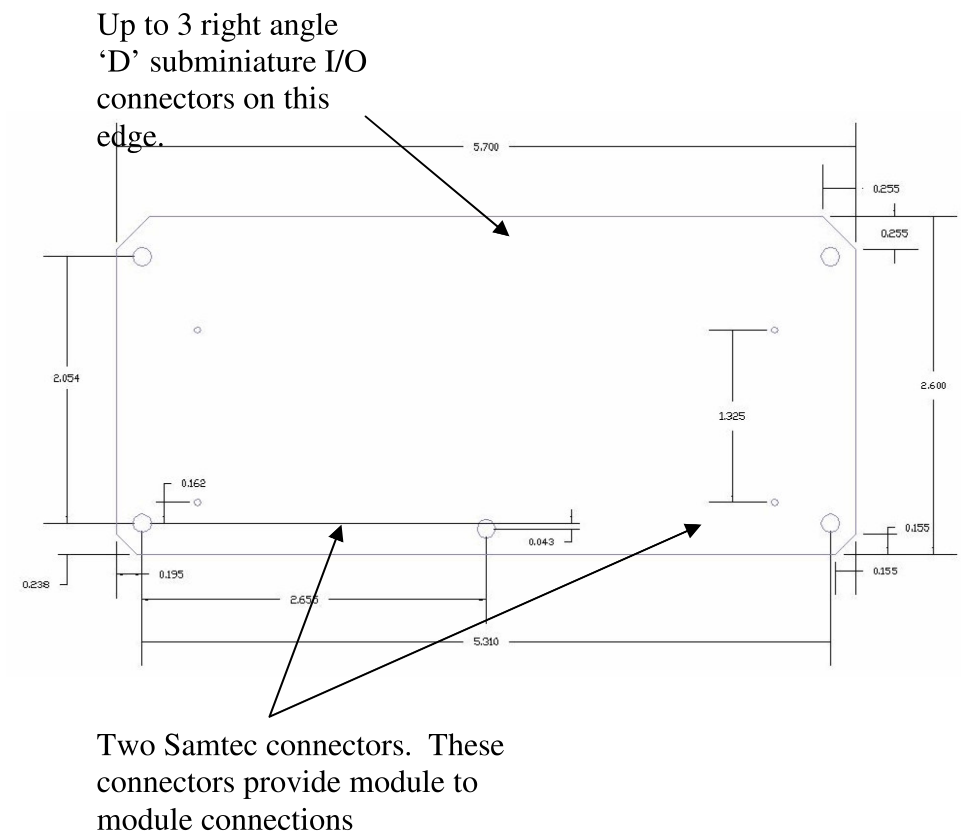


Figure 2.3: Board Outline and Connector Locations

1) Current board dimensions are 2.6 inches by 5.7 inches. This requirement was originally a hard requirement, but given Orbital's flexibility on the requirement, we decided it was actually a requirement to optimize. Orbital has allowed for some flexibility on board outline and connector location, if different mounting options are chosen.

2) I/O Connector locations should be on no more than two faces of the unit. Having connectors on only one face is desirable. This refers to the D-subminiature connectors used to connect other parts of the rocket avionics

to the unit being designed. For an example of a D-subminiature connector see Figure 2.4 below.

Note: These are high density D-sub connectors from Positronic.

Figure 2.4: D-subminiature Connector

3) The unit footprint shall be optimized. Obviously, reducing the unit head height will have an adverse effect on the unit footprint. By comparing the final unit head height reduction to the footprint increase, a feel for the efficiency of the design can be attained.

4) The board and unit resonant frequencies. Of utmost importance is that board resonant frequencies are close to or above 500 Hz. This is because low frequencies that are accompanied by high deflections due to resonance can cause failure. To avoid coupling of board and unit deflections at resonant frequencies, the unit resonant frequency should be at least an octave separated (higher) than board resonant frequency.

5) Based on resonant frequencies, safety factors for vibration and shock stress shall be calculated using the environments described in section 2.3.0 below. The safety factor for stress shall be 1.4.

6) Thermal conduction paths shall be understood and temperature rise optimized. Unit power dissipation at the center of the board shall be used to predict and optimize temperature rise.

7) The unit's mass shall be optimized so that it is comparable to the current unit, while still meeting all functional requirements.

8) Design for manufacturability. This means:
a) Reducing hardware (screws, nuts and washers) count and hardware types
b) top-down design
c) Reduce the number of piece parts

9) Minimize cost: Simplify parts to reduce machining costs.

2.3.0 Environments

Orbital has supplied standard test environment information to help our design team determine loads on the design during analysis.

2.3.1 Vibration environment

Upon implementation, the design will be put through random vibration testing to simulate vibration levels in flight per MIL-STD-1540. The random vibration spectrum can be seen in Figure 2.5 and Table 2.1 below.

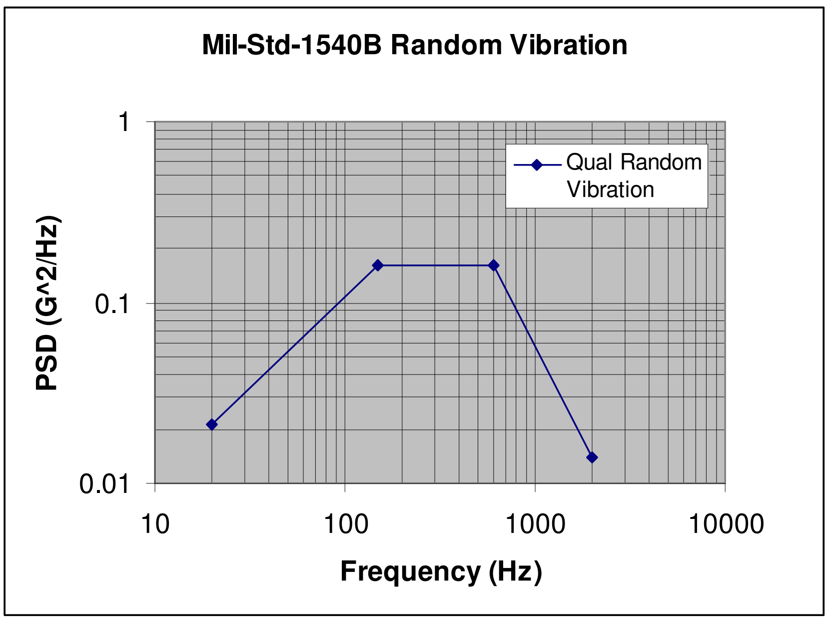


Figure 2.5: MIL-STD-1540 Random Vibration Spectrum

Table 2.1: MIL-STD-1540 Required Qualification Levels

Req'd Qual Levels	
Hz	PSD
20	0.021
150	0.16
600	0.16
2000	0.014
12.2 Grms	
3 min/axis	

Once the natural frequency is found, the MIL-STD-1540 levels can be used to look up the acceleration of the design at resonance during random vibration testing. This acceleration can be used to find loads on the design at resonance.

2.3.2 Shock environment

During flight, the design will see a number of shock events. As such, a 2000 G shock test is required by Orbital. The spectrum for the shock test can be seen in Figure 2.6 and Table 2.2 below.

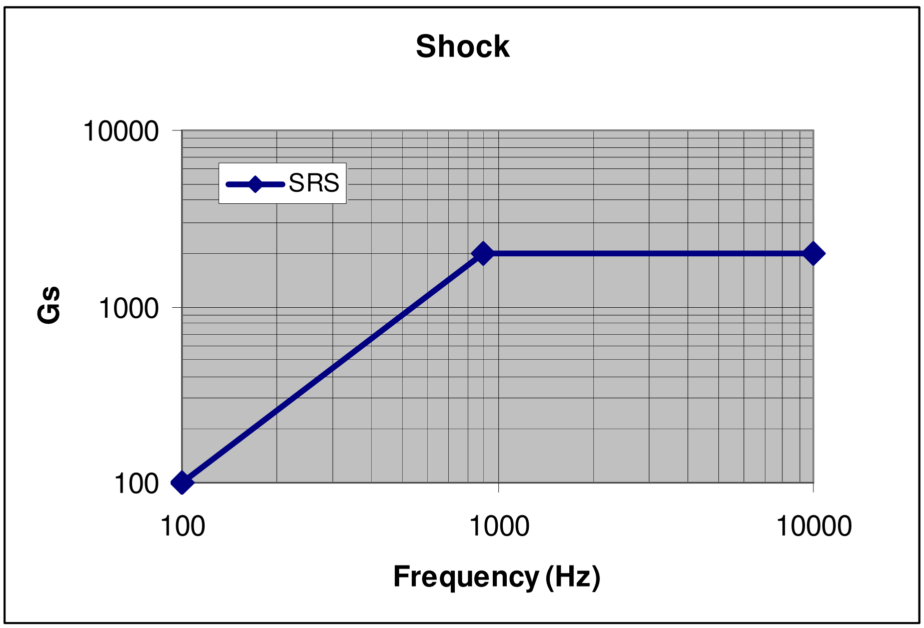


Figure 2.6: 2000 G Shock levels

Table 2.2: 2000 G Shock levels

Shock Response Spectrum	
Hz	Gs
100	100
900	2000
10000	2000
3 hits/axis	

As with the random vibration, the resonant frequencies can be used to find shock levels during testing for the unit.

2.3.3 Other environments

Other environmental concerns exist during flight. Orbital chose to not require our design team to specifically design for these environments, but encouraged us to keep them in mind. Some examples are below:

Thermal cycling: Orbital thermal cycle tests all units put into flight. This is to simulate the temperature variations that occur during flight. The team's thermal paths optimization addresses this environment somewhat, but not entirely.

Thermal Vacuum: In a vacuum environment like space, convection does not exist. As such, thermal response of a design in a vacuum can become a concern. The thermal conduction paths analysis only considers conduction, and therefore does something to plan ahead for loss of convection. Also, with high pressure changes during flight, venting of the unit can become a concern, to avoid blowout due to high pressure differences.

Electromagnetic Interference: While no analysis directly confirms any planning for EMI, a qualitative approach can be used to plan for EMI. Exposed electronic parts can be a problem for EMI, so this is something Orbital encouraged the design team to consider during the design process.

3.0 Design Philosophy

The design philosophy involves a mixture of originality and trade study. Orbital encouraged our design team to try to come up with original ideas that we could take ownership of. We were very careful not to let Orbital's design ideas wholly influence our design decisions. Orbital's valued input on the design served as a guideline we followed to increase design efficiency, not to define our design. In this way, the chosen design shows some originality, while still using industry standards to ease manufacturing and design.

Based on this philosophy, the following design and analysis processes were used:

3.1.0 Design Process

The first step in design is the proposal. This required find a customer that has a need (Orbital), contacting the customer, determining their requirements, and conducting initial research of the subject to determine project feasibility. Then a proposal document was written which stated the details of the service that would be provided to the customer. Orbital and the design team agreed on the terms of the proposal and the next stage began.

The second stage was brainstorming general design ideas and choosing a design using a decision matrix. After more in-depth research of the State-of-the-Art (SOTA) the team produced a list of all design possibilities, feasible or not, and grouped them into subsystems. Then all possible combinations of subsystems were determined to produce system designs. Next, these system designs were evaluated with a decision matrix to eliminate the designs that were not feasible, or inferior. Some hand analyses and simple CAD modeling were performed to help in decision matrix development. The decision matrix determined the general form of the final design.

Once the final design had been chosen, more in depth hand calculations and computer modeling took place. All throughout the process, documentation was compiled by the design team including time reports updating the amount of time devoted to the project, frequent status reports updating the customer on project progress, a design report outlining the final design and intermediate analysis results, a final report giving detailed information about the final design, all analysis results, detailed drawings of the design, and any prototypes constructed during the course of the project.

3.2.0 Vibration Analysis

Hand calculations estimated a natural frequency for the housing(s) and a natural frequency for the PWB(s) mounted in the housing. Mass, size, material, and mounting information were used to perform a Steinberg analysis finding the natural frequency of the PWB(s). Common single degree of freedom methods were used to find the fundamental natural frequency of the housing

For the computer modeling COSMOS/M Geostar was used to find the natural frequencies and confirm the hand calculations. The packaging and PWB's were modeled in COSMOS, and a mesh created with the known material properties. The A_Frequency and R_Frequency commands performed and ran the frequency analysis, outputting natural frequencies.

3.3.0 Shock Analysis

Hand calculations were used to find the natural frequency as mentioned for the vibration analysis. This shock level at the natural frequency was used to determine the force acting on certain critical design elements. A subsequent stress analysis was performed.

3.4.0 Stress/Strain Analysis

The accelerations due to natural frequencies determined from the vibration and shock analyses were used to find the forces for the stress analyses. Computer modeling in COSMOS was used for a Finite Element Analysis to verify maximum deflections and stress concentrations over the PWB(s) and housing components.

3.5.0 Tolerance Stack-ups

Tolerance stack-up analysis was performed in critical locations where clearance could be a problem. This includes, but is not limited to, fastener locations, connecting part interfaces, connector interfaces, housing interfaces, and low board clearance locations. Tolerances of the components involved in the stack-up were obtained for stack-up analysis. For only two components, the tolerances were added and the range of possible values measured in that way. For three or more components a root-sum-square (RSS) method was implemented to statistically account for individual errors.

4.0 Research

In order to familiarize ourselves with the problem and understand what existing design solutions existed, research was conducted. Detailed below are the research subjects most pertinent to the chosen design. The full State-of-the-Art research can be found in Appendix C.

1) Card-Loks

Card-loks are manufactured and sold by a company named Calmark. They are a method of attaching the boards to the housing. A screw runs through wedged pieces of aluminum, increasing the height of the unit and using pressure to secure the board. Consequently, an edge boundary condition between fixed and simply supported is attained. For its natural frequency-raising ability, it was selected for use in our chosen design. Figure 4.1 below shows a card-lok.

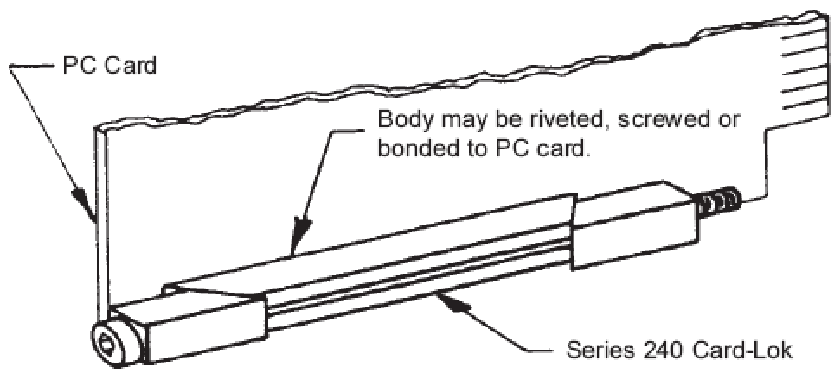
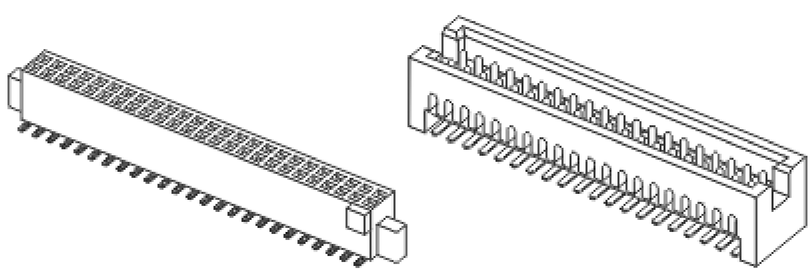


Figure 4.1: Card-lok

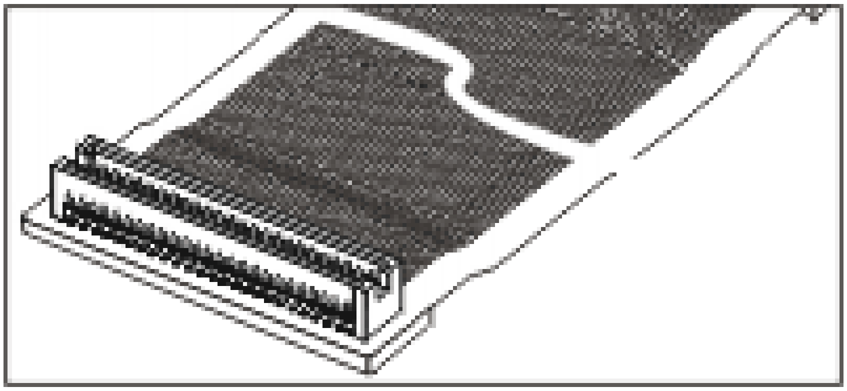
2) www.samtec.com

Samtec is a company that produces and sells connectors and flex cables used on the current Orbital design. Our chosen design uses the same and similar connectors and flex cables. Samtec connectors and flex cables provide a flexible interface between boards and modules and can be seen below in Figures 4.2 and 4.3.



Note: These are Samtec's SFM and TFM connectors, used in the design

Figure 4.2: Samtec Connector



Note: This Samtec Flex cable can have the TFM or SFM connector on the end.
Figure 4.3: Samtec Flex Cable

3) Positronic Industries
Positronic Industries provides the current and chosen designs' D-sub connectors. These are shown above in Figure 2.4.

4) Dave S. Steinberg's Vibration Analysis for Electronic Equipment
Steinberg's book is very useful in performing hand vibration calculations for printed wiring boards (PWBs). It was used for hand calculations of natural frequency for the chosen and current design.

5.0 Design

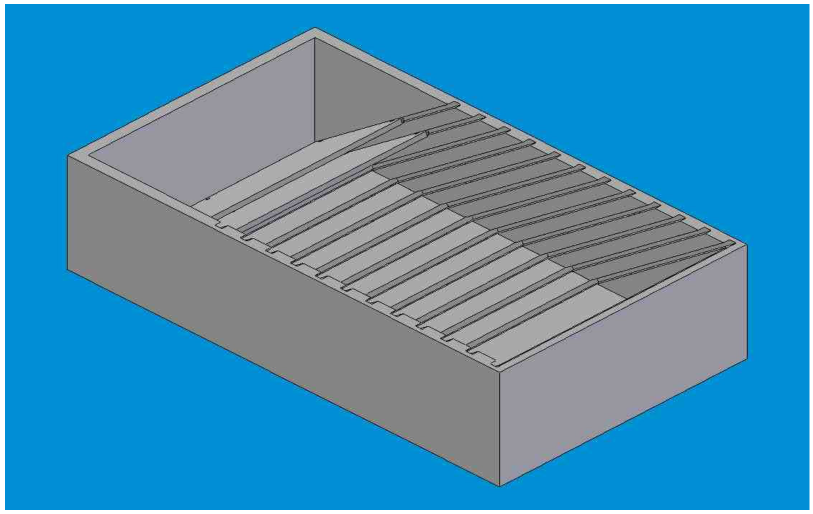
As per the design process described in section 3.1.0, numerous design ideas were brainstormed and evaluated. Next, a final design was chosen based on preliminary modeling and analysis. Finally, further modeling introduced more and more design decisions and considerations.

5.1.0 Brainstormed Design Ideas

As a result of the brainstorming process, a variety of possible designs, orientations, connector locations, and attachment methods were discussed. Five of the resulting design concepts are discussed below.

5.1.1 Slanted Unit

The slanted unit design concept shown below in Figure 5.1 consists of a single housing that can accommodate up to six function boards and six controller boards.



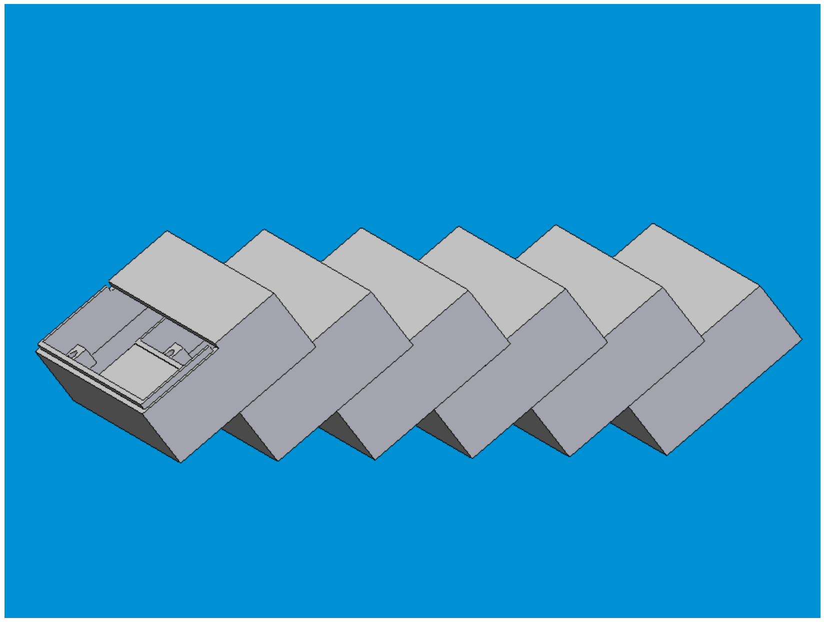
Note: Two boards have been placed in housing

Figure 5.1: Slanted Unit Design Concept

This design accomplishes the reduced head height requirement by orienting the boards at an angle. To meet the requirement of an overall head height under five inches, the angle of the boards with respect to the bulkhead would have to be about 30 degrees. This would result in greater horizontal spacing between boards to ensure there would be no interference. There would also be a significant amount of wasted space on the ends of the housing. Another issue is cable strain. All of the cables and backshells would have to come out of the top of the housing, creating more head height, and the angle at which the boards are oriented may create issues with connector placement and spacing.

5.1.2 Slanted Module

The slanted module design retains the modularity of the current housing design, only orienting the modules at an angle to reduce the head height. A rough representation of this is shown in Figure 5.2 below.

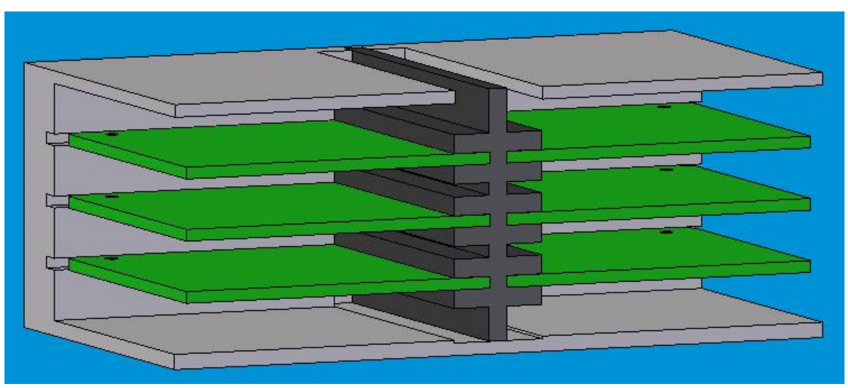


Note: Two boards would be mounted inside each module
Figure 5.2: Slanted Housing Design Concept

In this design, the boards would be attached to each other with standoffs and one would be attached to each module using bolts. This design would require a significant amount of machining, and the geometry would be complex with all the required lips and inter-modular connections. Another difficulty would be mounting each module to the bulkhead due to the angled ends.

5.1.3 Motherboard

The motherboard design concept would utilize a single motherboard in place of the six controller boards. This concept is shown if Figure 5.3.

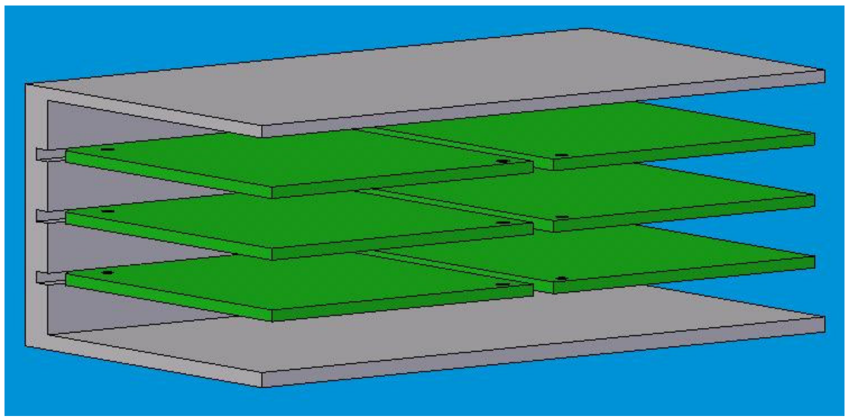


Note: This cutaway view shows that function boards are connected to both sides of the motherboard
Figure 5.3: Motherboard Design Concept

The motherboard design provides the simplest geometry and requires that only six boards be accommodated instead of twelve. The function boards would be attached to both sides of the motherboard, and attached to the housing with wedge-locks. This design would require that a motherboard be designed which would take a significant amount of work and add to the overall cost. Another disadvantage of this design is the relatively large footprint required due to the need for connectors, cables, and backshells on both sides of the module.

5.1.4 Back-to-Back

The Back-to-Back design concept features two modules, each capable of holding three function boards and controller boards. Each function board and its corresponding controller board lie in the same plane. A cutaway view of this concept is shown in Figure 5.4 below.



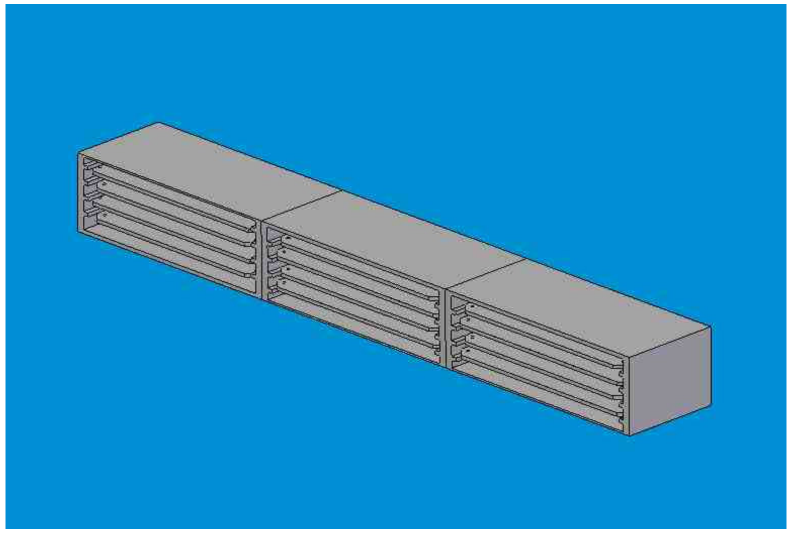
Note: There are two modules with three pairs of boards each in this design

Figure 5.4: Back-to-Back Design Concept

While the head height could be reduced significantly with this concept, cables, connectors, and backshells would again be required on the front and back faces of the modules making the overall footprint the largest of all of the designs.

5.1.5 Two-Stacks

The two-stacks design is shown in Figure 5.5. This design consists of modular housings each containing two complete function and controller PWB pairs. The modules would be connected mechanically and electrically.



Note: Each stack of two function and control boards would be contained in a modular housing

Figure 5.5: Two-Stacks Design Concept

The original brainstorming concept was called two-Stacks, and each module contained two pairs of boards as shown above, but after calculating the amount of space available by moving all the cables from the top face to the sides, it was determined that three pairs of boards could be put in each module, leading to the chosen design of three-stacks. This design also features a more efficient footprint due to the possibility of attaching all cables, connectors, and backshells to one face of the housing.

5.2.0 Hand Analysis

To aid in selecting a design from the initial concepts, some hand analysis was performed. This allowed for objective methods of evaluating the design concepts. Although this analysis could be placed in the analysis section of this report, it fits in the design section as a method for choosing a final design.

5.2.1 Vibration Analysis

Overview: The general idea of the hand vibration analysis was to use Steinberg analysis methods to try and predict the natural frequency of the printed wiring boards (PWBs) in certain mounting configurations. MATLAB was used to calculate the results of the appropriate Steinberg plate natural frequency equations.

Input: The driving inputs for the program were the appropriate Steinberg equations, which can be seen below.

For wedge-clamp calculations, both simply supported and fixed on all sides boundary conditions were needed. Wedge-clamp mounting is a mixture of the two, and Steinberg wedge-clamp calculations need both values.

For a plate simply supported on all sides:

$$f_s = \frac{\Pi}{2}\sqrt{\frac{D}{\rho}}\left(\frac{1}{a^2}+\frac{1}{b^2}\right)$$

where: f_s = the simply supported natural frequency
ϱ = the mass density per unit area of the board
a = length of the long side of the board
b = length of the short side of the board

$$D = \frac{Eh^3}{12(1-\upsilon^2)}$$

where: E = Young's modulus
h = board thickness

υ = Poisson's ratio

For a plate fixed on all sides:

$$f_f = \frac{\Pi}{1.5}\sqrt{\frac{D}{\rho}\left(\frac{3}{a^4}+\frac{2}{a^2+b^2}+\frac{3}{b^4}\right)}$$

where: f_f = the fixed edge natural frequency

For a plate wedge-clamped on the two short sides and simply supported on the long sides:

$$f_w = \frac{f_s + 1.10(f_f - f_s)}{1+0.001(f_f - f_s)}$$

where: f_w = the wedge-clamped natural frequency

For the current PWB bolted on 4 corners, boundary conditions of 4 sides simply supported were used as an approximation.

The values used for base inputs were as follows:
$\varrho = 8.282 \times 10^{-5}$ slug/in^2
a = 5.7 in.
b = 2.6 in.
E = 3.9×10^6 psi (Polyimide)
υ = 0.12 (Polyimide)

Method: The above values and equations were put into a MATLAB program so that changes could be easily made and implemented. The source code for this MATLAB program can be found in Appendix 2.

Expected Output: The output expected from the vibration analysis was a relative comparison of wedge-clamp versus screwed board fastening methods. Typically, wedge-clamped connections have a higher natural frequency than simply supported edge conditions. As such, the expectation was that the natural frequency would be highest for the fixed edge condition, and lowest for the simply supported edge condition, with the wedge-clamped edge condition in the middle.

Results: The program output showed the following natural frequency values:

ff =

1.0441e+003

fs =

502.3686

fw =

712.3626

As can be seen above, the simply supported condition of the current design shows a board natural frequency of 502.37 Hz. For the planned wedge-clamped edge condition, the predicted board natural frequency is 712.36 Hz.

Conclusion: As a general rule, wedge-clamps improve vibration characteristics of PWBs. For this reason, wedge-clamps became a desired feature in the design. Some issues that are not covered by this analysis are total unit natural frequency and standoff effects. The housing natural frequency is difficult to determine from such a raw analysis. As such, the mass was taken as a controlling factor for preliminary total unit natural frequency calculations. This value is found in the FEA analysis described in section 6. The standoffs provide additional stiffness both to the current design and the future design. These effects were neglected for both designs to ensure that a fair comparison was made. This issue will also be resolved in the FEA analyses of Chapter 6.

5.2.2 Thermal Analysis

In meetings with Orbital, the client expressed that it would be most conservative to assume the only method of heat transfer was through conduction. Therefore, effects due to convection and radiation were neglected. It was also assumed all heat to be at the center of the printed wiring board and conducted through and off the board. The board material is a poor conductor. It was therefore assumed that optimized conduction paths were designs that had maximum surface area contact with the printed wiring board and aluminum housing, as aluminum is a good heat conductor. It was decided that wedge locks were preferable to fastening by bolts. The wedge locks had higher surface area contact with the board and housing for better conduction paths.

5.2.3 Manufacturability Analysis

A qualitative analysis of the relative manufacturability of each design was conducted with five main categories: Quantity of machined pieces (part count), assembly requirements, tolerance limitations and requirements, geometric complexity, and inter-modular connection. Other factors unique to each design were also considered. Below is a description of the characteristics of each design concept and a manufacturability summary table (Table 5.1).

1) Slanted Unit
    - Part count – Requires at least six face plates with slots or flanges for circuit board connection
    - Assembly – Each side would have to be fastened to each other side and wedge-locks, bolts, or another form of connector would have to be used to secure each board.
    - Tolerance – Slots would have to follow close tolerances in order to maintain proper clearances. Also, the angle of the slots would directly affect the clearance between boards and must be maintained constant.
    - Complexity – Moderate complexity of machining the slots and proper lips to prevent EMI interference would be encountered.
    - Connections – No inter-modular connectors would be necessary, only connection points for each board and to the spacecraft.

2) Slanted Modules
    - Part count – Each module could be made from a single piece, so part count would be limited to number of modules.
    - Assembly – Easy assembly and no post machining assembly required.
    - Tolerance – Bolted connections for all boards would require less strict tolerancing.
    - Complexity – Complex geometry would require CNC coding and would have to be turned over to finish part geometry.
    - Connections – Inter-modular connections would be required as well as a method for fastening slanted modules to the spacecraft.

3) Motherboard
    - Part count – Fewer parts than modular options, but still more than non-modular concepts.

- Assembly – Fewer boards than other concepts would require less assembly time.
- Tolerance – Tolerance on slots would be critical, yet more room clearance would be available due to reduced number of boards.
- Complexity – Geometry is not complex.
- Connections – No inter-modular connections required. Wedge-locks or fasteners required for boards.

4) Back to back
    - Part count – The modular nature of this design results in more parts.
    - Assembly – Post-machining assembly would be required for each module.
    - Tolerance – Tight tolerances on slots.
    - Complexity – Simple part geometry would result in ease of machining.
    - Connections – Connections between modules and to spacecraft would be required.

5) Two-stacks
    - Part count – This design would result in the most machined pieces. Each side of each module would have to be fabricated separately.
    - Assembly – Post-machining assembly would be required for each module.
    - Tolerance – Tight tolerances on slots.
    - Complexity – Simple geometry would result in ease of machining.
    - Connections - Connections between modules and to spacecraft would be required.

Table 5.1: Summary or relative manufacturability of each design concept

Category	Design				
	Slant. Unit	**Slant. Mod.**	**Motherboard**	**Back-2-Back**	**Two-Stack**
Part count	(–)	(+)	(–)	(–)	(– –)
Material	(–)	(–)	(+)	(–)	(–)
Tolerance	(–)	(+)	(–)	(–)	(–)
Complexity	(–)	(–)	(+)	(+)	(+)
Connections	(+)	(–)	(+)	(–)	(–)
Total	**-3**	**-1**	**1**	**-3**	**-4**

5.2.4 Attachment search/development – Methods and devices for securing the circuit boards to the housings were researched and a list of alternatives is given below. Although not a true analysis, this research was integral as a method in evaluating the design concepts.

1) Wedge-locks – These mechanisms produced by Calmark (also known as Card-loks) are basically three or five wedges attached to a screw that when tightened forces the wedges together decreasing the length between each wedge and forcing them apart against the housing and the board. An example of a wedge-lock is shown below in Figure 5.6.

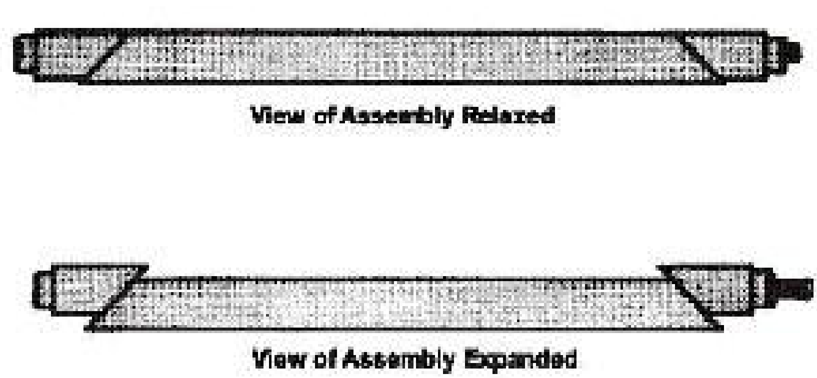
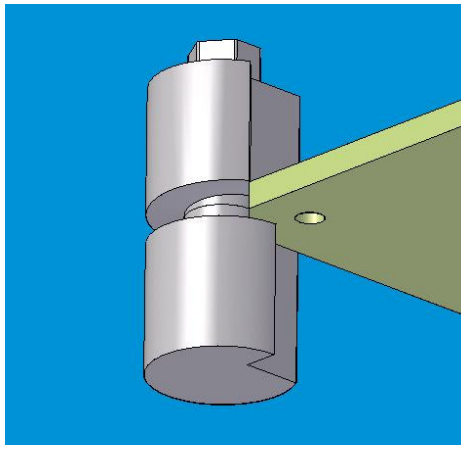


Figure 5.6: Wedge-lock

2) Bolts – By far the simplest method of attachment, bolts provide a force to a concentrated area of the board equal to the area under the head of the bolt. Using a washer or as a bar spanning two bolts can increase this area and provide greater stability.
3) Cylinder-Lock –Our design team conceived this idea while considering fastening options. The design comprises of a cylindrical piece of material with a tapering slot that constricts on the PWB as the cylinder is turned on its axis. A transverse groove will be cut along the axis of the cylinder to allow for the board to slide down to the groove, and a lip on the bottom side of the slot will assure that the board will be in the correct position. These cylinders could be stacked in order to accommodate more than one board. Figure 5.7 below shows a model of the cylinder lock.



Note: Tapered slot constricts PWB as cylinder rotates

Figure 5.7: Cylinder Lock Device used to secure PWBs

5.3.0 Decision Matrix

In order to narrow the designs to a final design, a decision matrix was put together. Table 5.2 below shows the decision matrix. Along with the five brainstorming designs, the current design used by Orbital was also included. This was to ensure that the redesign was an improvement from the original design. The needs used in the matrix were mostly the requirements set forth by the client as hard requirements and optimization requirements. An importance level was assigned to each need so the final score was weighted to the most important needs. The most important needs in the matrix included: Accommodating 6 modules, Maximum head height under 5 inches, and Board outline remaining unchanged. The results of the matrix show the Three-Stack design (a variation on the two-stack brainstorm idea) to be superior, with the Motherboard design coming in a close second. The matrix also shows that the redesign process is a success, as the chosen design easily beat out the current used design.

Table 5.2: Decision Matrix

Decision Matrix Needs	Importance (1 = low, 10 = high)	Current Design	Slanted Unit	Slanted Module	Motherboard	Three Stacks	Back to Back
Board outline unchanged	10	5	5	5	5	5	5
Connector locations unchanged	6	5	5	5	5	5	2
Accomodates 6 modules	10	5	5	5	5	5	5
Max head height under 5 inches	10	0	2	2	5	4	4
Footprint optimized	4	5	3	4	1	2	4
Connector locations optimized	6	3	2	2	4	5	5
Desirable board resonant frequencies	10	4	4	4	5	5	5
Adequate safety factor	9	5	3	3	3	4	4
Optimal thermal conduction paths	6	2	4	2	2	4	3
Optimal mass	7	5	1	2	5	4	3
Optimal manufacturability	6	4	3	4	5	1	2
Optimal cost	7	4	3	4	1	3	3
Total		352	311	323	369	373	356

Note: Each column score is determined by multiplying importance by given category score

5.4.0 Chosen Design: Three-stacks

Based on hand analysis and the decision matrix shown above, the design group chose the three-stacks design concept as the preferred design option. The two-stacks design was changed to a three-stacks design because of cable strain calculations. As a result of moving the connectors from the top face to a side face, a great deal of space was gained for head height. As such, three sets of boards can fit in each of the two housing units necessary to accommodate 6 modules. This section focuses on the numerous design decisions and features of the chosen design.

5.4.1 Overview

The chosen design is semi-modular, housing three modules (sets of boards) in two separate housings. The housings are made of 6061 T651 Aluminum and are connected with a flex cable designed by Team Orbital. Both housings connect to the vehicle bulkhead using bolts run through gusseted feet on three sides of the housing. A board set is held together using standoffs. The boards set is then slid into the housing and secured with card-loks. Figure 5.8 below shows an overview of the design. The aforementioned and additional design features will be explained and investigated in the following sections.

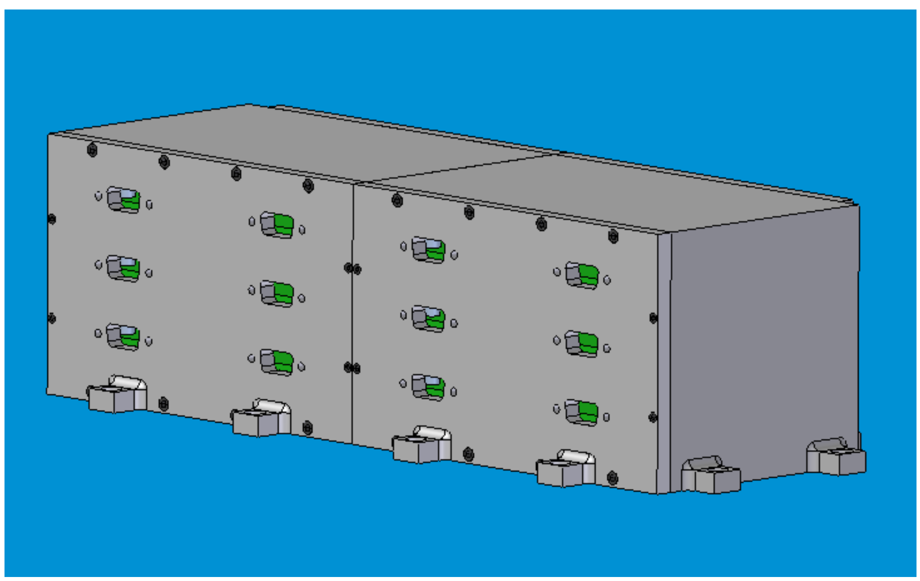


Figure 5.8: Final Design, Overview

5.4.2 Housing Design

For the housing it was imperative to keep head height below the maximum allowable head height of five inches. The final design has an overall head height of 3.6 inches. The 4.4 inch reduction in head height is due largely to moving the cable strain from the top of the unit to the front face.

Material selection for the housing was fairly simple. Orbital currently uses 6061 T651 Aluminum as the material for many of their avionics boxes. T651 provides acceptable strength, conducts heat well, and is lightweight, while remaining economically feasible. While the design philosophy had the design team considering other materials, nothing provided a set of strength, weight, conductivity and cost gains significant enough to replace the current standard choice.

The housing was developed as one solid piece machined out of a solid block of aluminum, with a front cover. Next, it was realized that a back and front cover should be added after board installation to allow for easy routing of cable between modules and housings. While milling a "shell" out of a solid block of aluminum introduces higher costs, in the aerospace industry it is preferable to using many different pieces. It improves manufacturability and helps with EMI and thermal environments.

The slots for the boards were created so that each set of boards could easily slide into the housing with the card-loks disengaged, and be secured with the card-loks engaged. The card-lok information sheets provided the necessary spacing needed for a card-lok to sufficiently engage in a slot. The slots have a small lip on the front edge, to ensure that the card-lok stays within a confined space even under unforeseen circumstances of unintentional disengagement or slippage. This lip can be seen below in Figure 5.9.

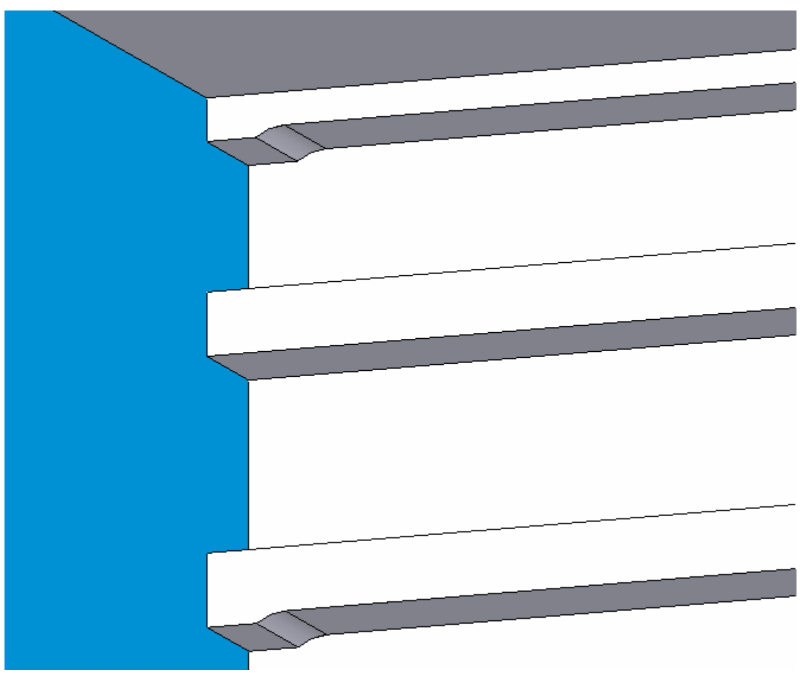


Figure 5.9: Housing Slot Lip

The space between the boards in a module is driven by the Samtec connector mated height of 0.320 inches. This means that a standoff length of 0.328 inches between the two boards will preserve a sufficient connection, while disallowing additional stress caused by the connectors over-connecting. To see a view of this interface, see Figure 5.10 below.

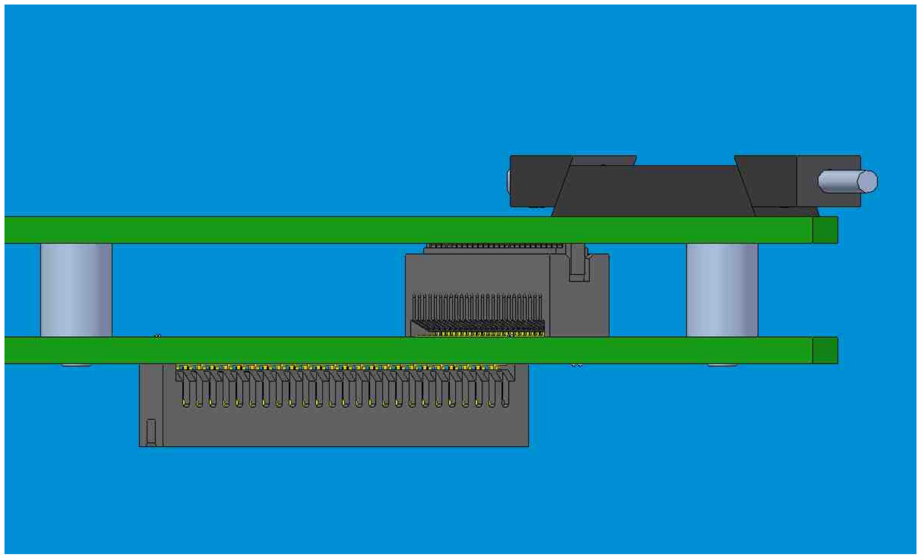


Figure 5.10: Board-Standoff Interface

The space between sets of boards is driven by Orbital's suggestion that we allow for a tallest part on the boards of 0.25 inches and a dynamic deflection of 0.020-0.025 inches. This dynamic deflection is based on the current Orbital design values. It is used by our design team conservatively since the final design should have better board frequency response than the current design. Assuming the "top" board of a module has its highest component over the tallest component on the "bottom" board of the next module, and maximum board deflections we get a total of 0.55 inches of space needed between board sets. For a view of this dimension, see Figure 5.11 below.

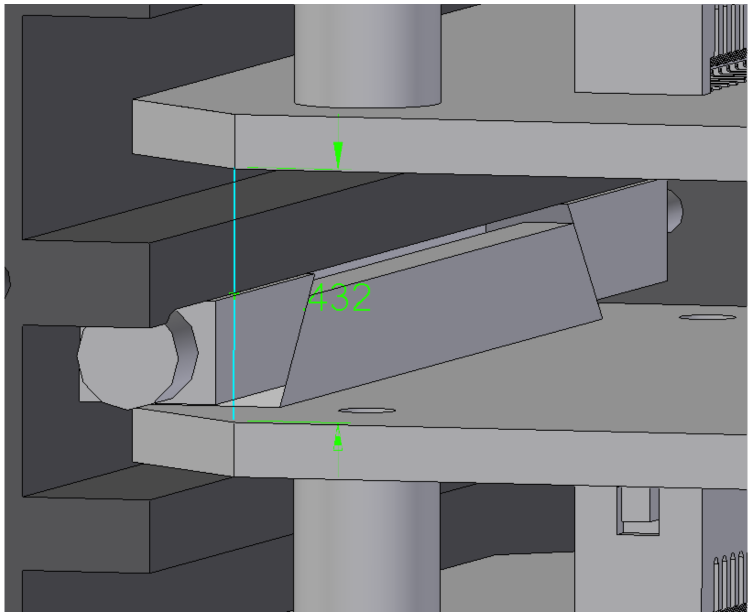


Figure 5.11: View of the Inter-modular Spacing Dimension

The space between the bottom set of boards and the bottom of the housing is larger due to connector necessities. The flex cable that connects the two housings together must be plugged into the bottom of a board in the next housing. As such, space for the full unmated heights of the Samtec connectors used is necessary. Of course, this space is only needed on one of the housings, but as a general design, the final housing design will be sufficient. To see this dimension, see Figure 5.12 below.

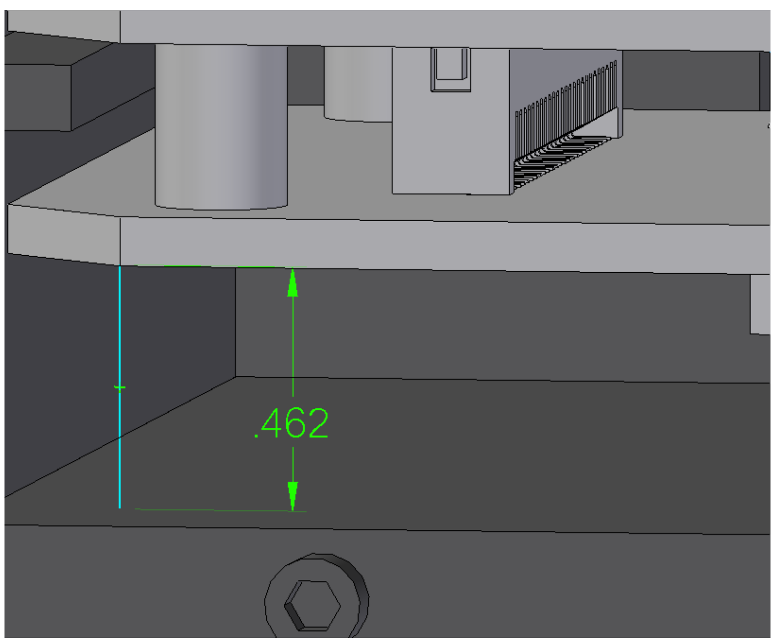


Figure 5.12: View of the Housing Bottom-Bottom Module Spacing Dimension

5.4.3 Board Mounting Design

Obviously a main goal of the design was to preserve board layout as much as possible. However, given the "slide-in" nature of the housing design, some changes were necessary. To accommodate card-loks, additional screw holes had to be added to one of the boards in each set. Luckily, the current board interface using standoffs was able to be preserved, thus minimizing the amount of work necessary for Orbital to implement the mounting method.

The main decision for mounting came down to choosing how to mount the boards in the housing. With the current design, boards are "laid" into the housings, thus allowing for use of standard fasteners like screws. While simple, cheap, and easy for the current design, the final design does not accommodate screw mounting as well. To do so, additional pieces would need to be used, leading to a loss in manufacturability. For this reason standard "slide-in" fastening methods were considered. Of the options available, card-loks were the best option. They are replaceable pieces of hardware that allow for individual fastening of the boards to the housing. Placing card-loks on two sides of the board and then simply supporting the other two sides would even lead to improved response in vibration over the current design. Card-loks on two sides with simply supported edge conditions on the other sides gives total edge conditions somewhere between all sides simply supported and all sides fixed. Figure 5.15 below shows the board set with card-loks attached.

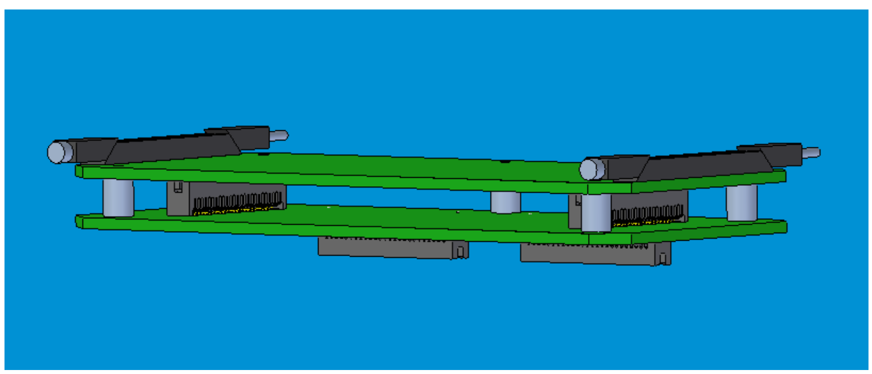


Figure 5.15: Board Set

One problem was that card-loks are commonly available with a minimum total wedge length of 2.8 inches. This is longer than the board edge. However, the only wedge that touches the board is the middle wedge, which is well under 2 inches in length. Thusly, the design team justified that the card-lok could easily be fastened to the board and no adverse stress conditions would be created.

Simply supporting the other two sides became the major concern. For the front edge, it was determined that making a faceplate to fit snugly around the connectors would create a condition close to simple support. For the back board edge, tight-fitting slots were the instinctual choice. However, tolerance stack-up analysis shows this to be unfeasible. Based on trade studies and some knowledge of industry practice, the design team realized that foam is sometimes used to attain a tight fit when tolerancing seems to be a problem. As such, the back edge slots would contain foam that allowed for a sort of preload to help simply support the board.

5.4.4 Inter-housing Flex Cable Design

To connect the housings together, it was necessary to design a flex cable that could easily connect the IBUS of the top module on one housing to the function card of the bottom module on the next housing. The beneficial thing about flex cable is that it can have a very sharp bend radius in a particular plane. This means that it can be used in rather tight spaces without worrying about cable strain issues, as long as it is bent in the correct plane. The design challenge to connect the housings was to create a flex cable that could connect the two housings with a minimal amount of cable twisting. The design attained is shown below in Figure 5.17.

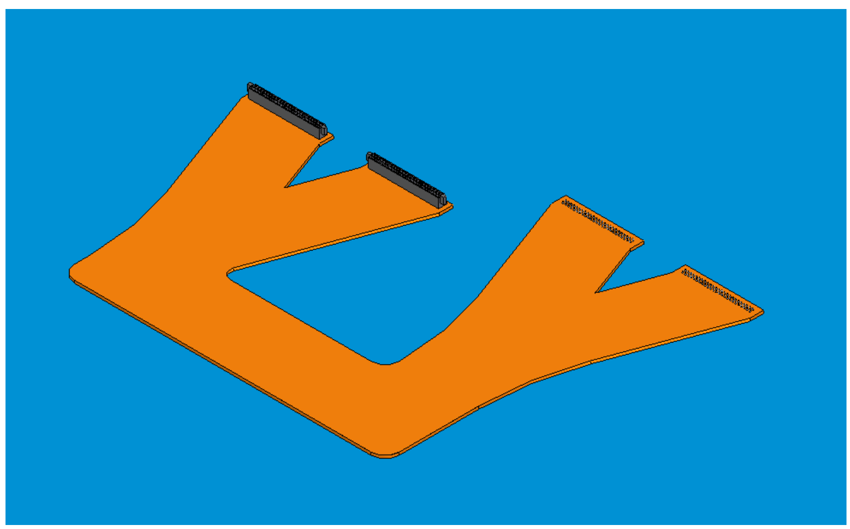


Figure 5.16: Inter-modular Flex Cable

5.4.5 Final Design Specifications

For use in general comparison, the important geometric and mass specifications have been gathered for both the current and final design. Table 5.3 below shows these results in tabular form.

Table 5.3: Design Specification Comparison

		Current Design	Final Design
Head Height	in^2	8	3.6
Overall Footprint	in^2	57.75	126.1
Housing Weight	lb	4.5	4.7

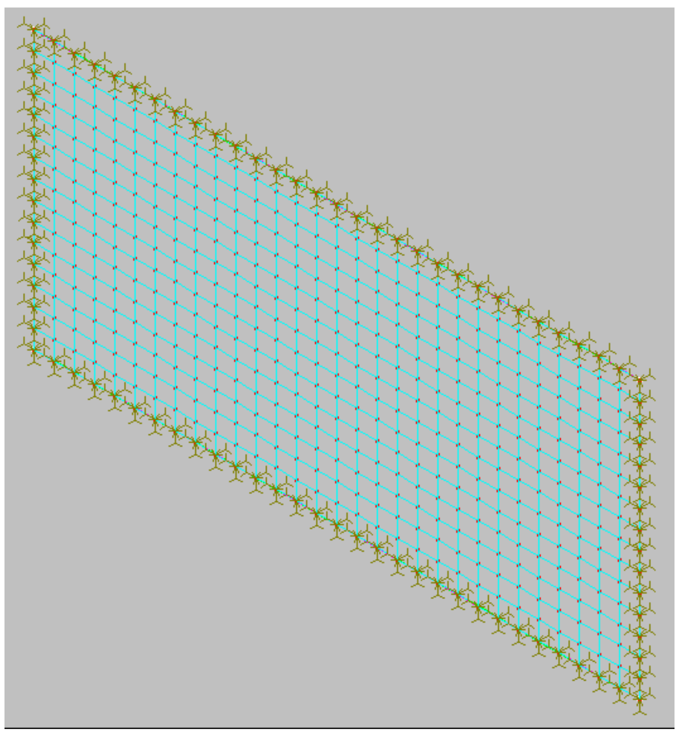
6.0 Analysis

6.1.0 Finite Element Frequency Analysis

The finite element frequency analysis was used to find the resonant frequencies of both the board pair and the housing. This helps determine stresses from vibration and shock events, and lets the design team check that there is sufficient separation between board and housing natural frequencies. All FEA was done in COSMOS.

6.1.1 Sanity Check

The first frequency analysis performed was a sanity check to ensure that the hand calculations previously performed were reliable. This just involved modeling a plate element (using Shell4 element type) and finding the frequency under different edge conditions. The first edge conditions were simply supported on all edges. The original geometry and mesh can be seen below in Figure 6.1.

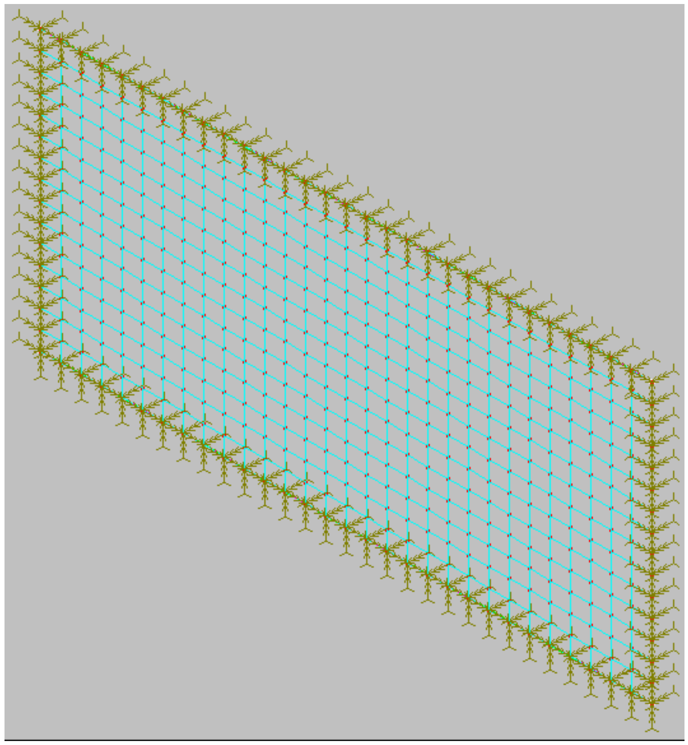


Note: The board is simply supported on all edges.

Figure 6.1: Simply Supported Mesh and Boundary Conditions

This analysis yielded a natural frequency of 502 Hz. This is virtually the same as the natural frequency with simply supported edges predicted by the hand analysis.

The next boundary conditions tested were the fixed on all edges boundary conditions. Figure 6.2 below shows the original mesh and boundary conditions for this finite element model.



Note: The board is fixed on all edges.

Figure 6.2: Fixed Edge Mesh and Boundary Conditions

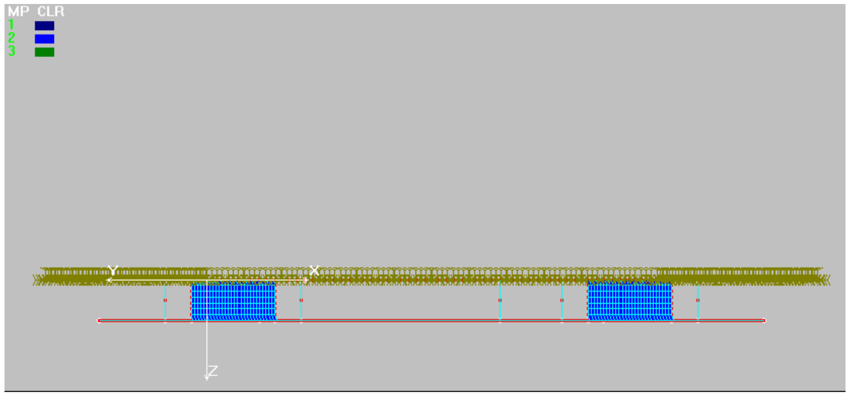
This analysis yielded a natural frequency of 1,013 Hz. This is very close to the 1,044 Hz found in the hand analysis. These two analyses act as a very good sanity check for the hand analysis. This lends a great deal of credence to the conclusion of the hand analysis that card-loks are a superior fastening mechanism to screwing the PWB in on four corners. Notably, this parity between hand analysis and the finite element results provides a good deal of faith in future finite element analyses.

6.1.2 Board Set Natural Frequency Analysis

The next step in the finite element frequency analysis was to model the boards as they are mounted in the housing. This includes modeling a set of two boards, one supported by the housing and the other held to the first board with standoffs. Just as done with one board, this analysis would be done with the top board simply supported on all sides in one case and fixed on all sides in another case. This gives a preliminary idea of the range of values the natural frequency of the board set falls in.

Just as before, both boards were modeled as plates using Shell 4 elements. The boards are connected together using beam elements to represent the standoffs. Also, plate elements were added to represent stiffness introduced by the connectors. Appropriate stainless steel material properties were used for the standoff beams.

The connector plate elements used the material properties of liquid crystal polymer (LCP). A picture of the mesh and geometry can be seen below in Figure 6.3 with simply supported boundary conditions.

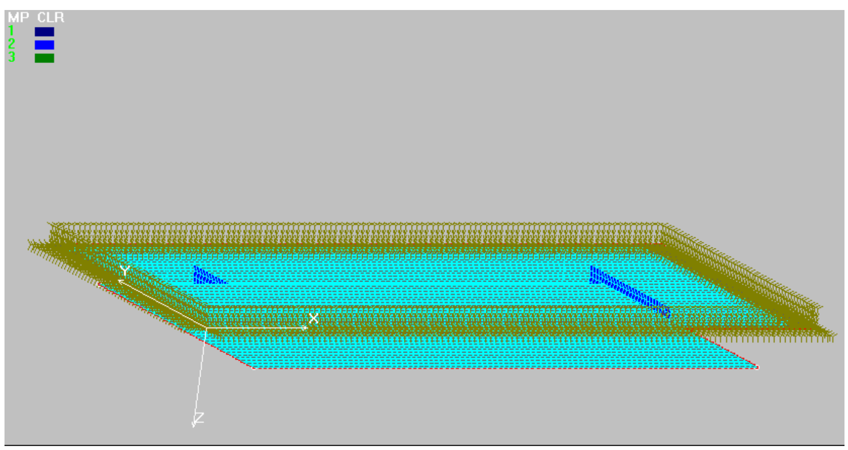


Note: This is a skewed side view of the boards that shows the five standoffs and the connectors.

Figure 6.3 Set of Boards with Simply Supported BCs

This frequency analysis yielded a natural frequency result of 448 Hz.

Next, the board set was modeled with the top board using fixed boundary conditions. This mesh geometry and set of boundary conditions can be seen below in Figure 6.4.



Note: This view shows a 3D view of the boards (light blue) with a small amount of the connectors showing (dark blue).

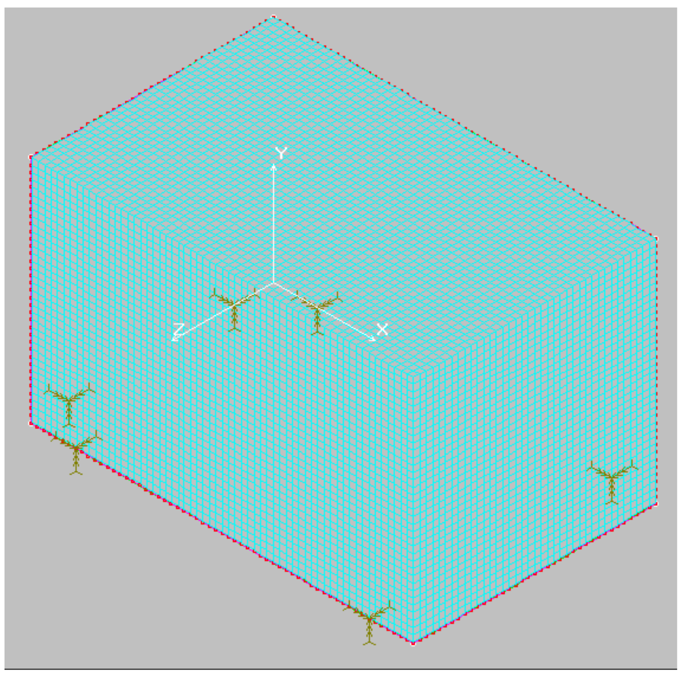
Figure 6.4: Set of Boards with Fixed BCs

This frequency analysis yielded a natural frequency result of 459 Hz. Using this result and the simply supported result, the natural frequency can safely be estimated as above 450 Hz for the set of boards. While this natural frequency is lower than desired for the final design, Orbital has assured the design team that these values are acceptable for a first cut at an untested design. Normally, with time permitting, testing would be done to verify analysis results and refine the design.

6.1.3 Housing Natural Frequency Analysis

The next section of the frequency analyses was meant to estimate the natural frequency of the housing. This will allow us to later check for failure in housing bolts joints and for one octave of separation between board and housing natural frequency.

The housing was modeled as a box composed of six plates. This assumes that bolted joints for the housing are sufficiently strong to achieve continuity between plate elements. The housing is fixed on the bottom at nodes near where the mounting feet will be. This represents the bolts holding the housing to the bulkhead. 6061 T651 Aluminum material properties and a plate thickness of 0.125 inches was used. The original mesh and geometry can be seen in Figure 6.5 below.



Note: The boundary conditions behind the visible elements can be seen through the model.

Figure 6.5: Housing Geometry and Mesh with BCs

Running the frequency analysis on this housing model yielded a natural frequency result of 1,345 Hz. Luckily, this is much greater than the natural frequency of the boards. However, it is still necessary that a check be done to ensure one octave of separation. This separation ensures that no coupling of adverse resonant effects occurs. To check for an octave of separation, the higher frequency should be halved, and the lower frequency doubled. If the higher frequency halved is larger than the lower frequency and the lower frequency doubled is smaller than the higher frequency, there is an octave of separation.

Halving the frequency of the housing gives 672.5 Hz. This is well above the predicted board natural frequency of 450 Hz.

Doubling the frequency of the boards gives 900 Hz. This is well below the predicted housing natural frequency of 1,345 Hz.

Clearly at least one octave of separation exists between the frequencies. This ensures that at natural frequencies, there will be no adverse coupling effect to the vibration displacements and loads.

6.2.0 Stress Analysis

The stress analysis was performed as two separate analyses. The first analysis investigated the deflections and stresses in the boards and the housing at resonant frequency. The second analysis took the mass of the box and ensured that fastener failure would not occur at resonant frequency.

6.2.1 Finite Element Board Set Stress Analysis

The first step to calculating the acceleration deflections and stresses at resonance is to calculate the acceleration in both shock and random vibration loading. At the board natural frequency of 450 Hz, the shock levels in Figure 2.6 show an acceleration of 900 G's (900 times the acceleration due to gravity). A transmissibility knockdown factor of 0.6 can be used twice on this G level to account for bolted interface transmissibility effects. This makes the final G level 324 G's. For random vibration, the PSD level from Figure 2.5 is 0.16. Using Miles' Equation shown below for converting PSD to G's, the 3-sigma peak G's were found. This is an acceleration representative of the maximum levels to be statistically expected in a random vibration test using a particular PSD level.

$$3\sigma PeakG's = 3\sqrt{\frac{\pi}{2}(PSD)(Q)(f_n)}$$

where: PSD=Power Spectral Density level

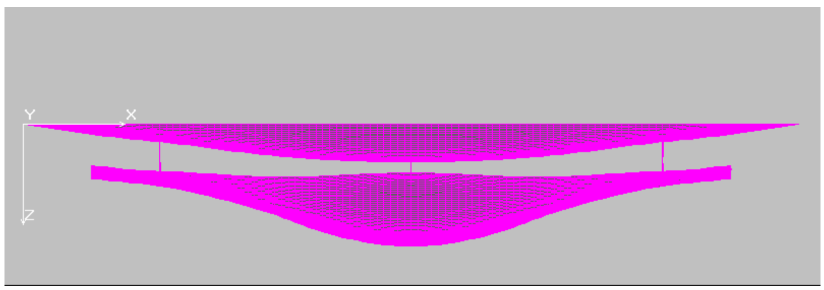
Q= Transmissibility
f_n= natural frequency
Usually, when Q is unknown, the square root of the natural frequency is used, as long as it is above 20. Using the above calculations:

$$3\sigma PeakG's = 147G's$$

Once these accelerations are known, they can be input into COSMOS as a gravity load in units of inches per second squared. This unit selection is based on the units used for geometry creation and material properties in COSMOS. This gives a shock acceleration of 125,064 in/s^2 and a random vibration acceleration of 56,742 in/s^2. After specifying a gravity load case for analysis options and then running the static analysis, results can be plotted.

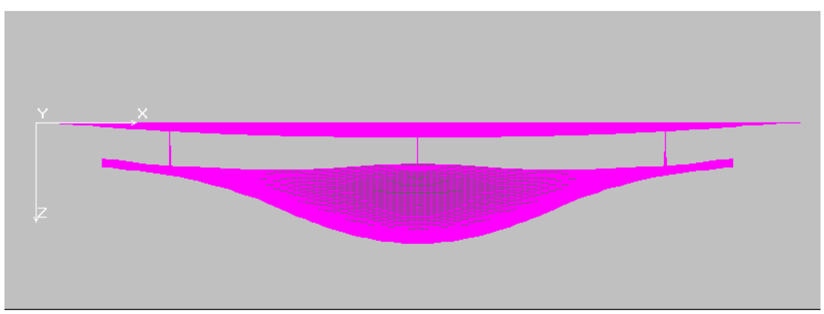
Since shock and random vibration are performed in each of the three axes, there are a great number of results. The main body of this report will only deal with stresses and displacements found in the maximum cases. Appendix F shows the rest of the results.

For the board set, the maximum load case is obviously acceleration applied in the Z axis direction, as shown in Figures 6.3 and 6.4 above. In this direction the "vertical" flexibility of the boards is taken into account. Figures 6.6 and 6.7 below show the deformed shape of the boards in shock loading for both simply supported and fixed edge condition scenarios.



Note: The above drawing is scaled from the actual deflections.

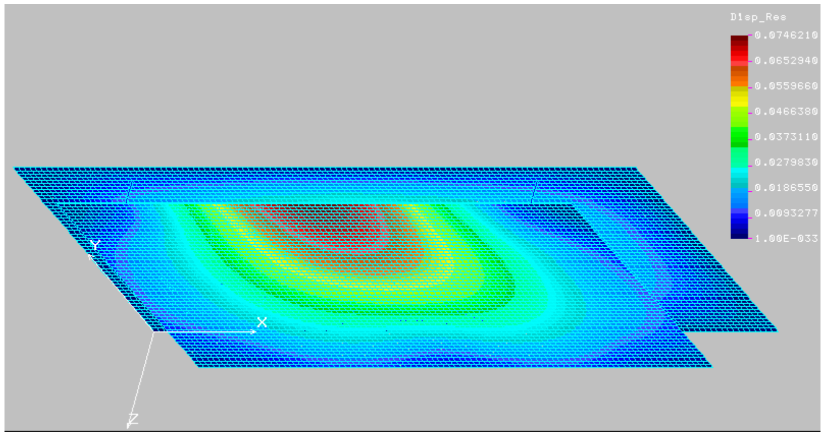
Figure 6.6: Simply Supported BC Deformed Shape



Note: The above drawing is scaled from the actual deflections.

Figure 6.7: Fixed BC Deformed Shape

Notice that the top board deforms much more in the simply supported case. This is to be expected, as the top board has inferior support in that setup. Peculiarly, the bottom board deflections do not change as much, as shown in Figures 6.8 and 6.9 below.



Note: The bottom board sees the maximum deflections.

Figure 6.8: Simply Supported BC Deflection Plot

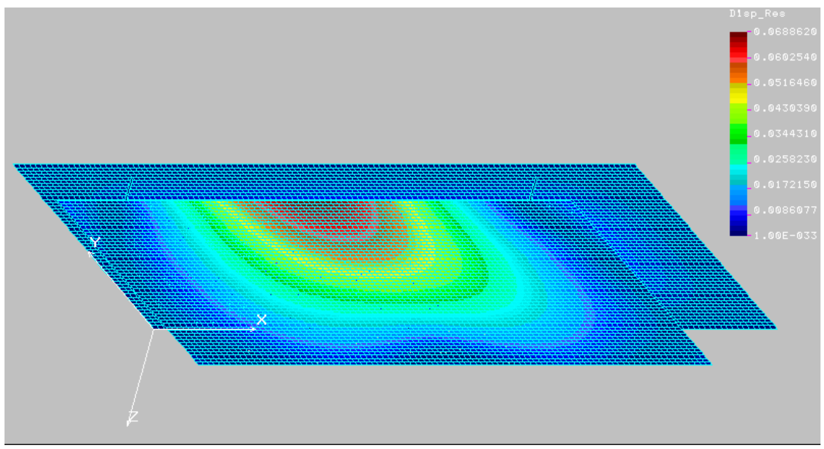
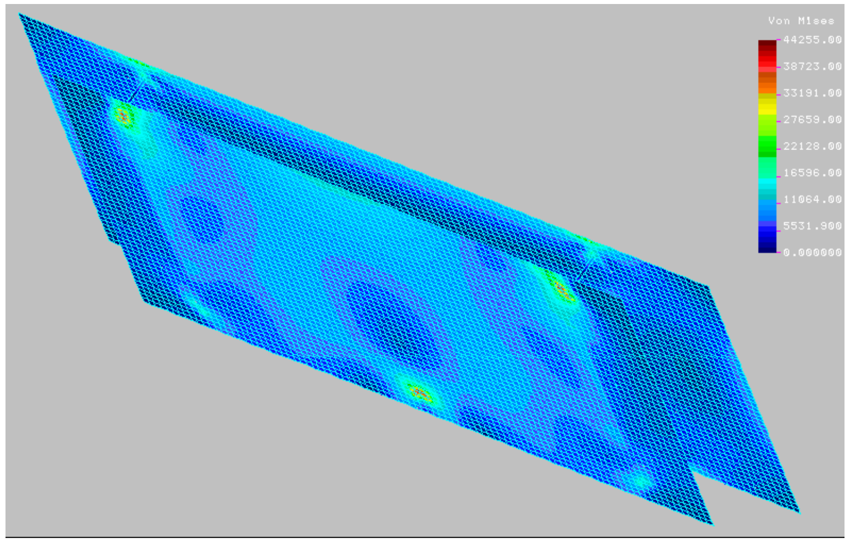


Figure 6.9: Fixed BC Deflection Plot

The maximum deflection seen for the simply supported case is 0.027 inches. The maximum deflection seen for the fixed case is 0.025 inches. Surprisingly, there is not a great deal of difference between the two cases. This is a common trend found in analysis, that seems to point to the conclusion that "hanging" one board from another using standoffs is harmful to vibration response.

The next step in the analysis was to look at the stress results. Figure 6.10 below shows the stress plot for the fixed boundary conditions. The plot is similar with different values for the simply supported case.



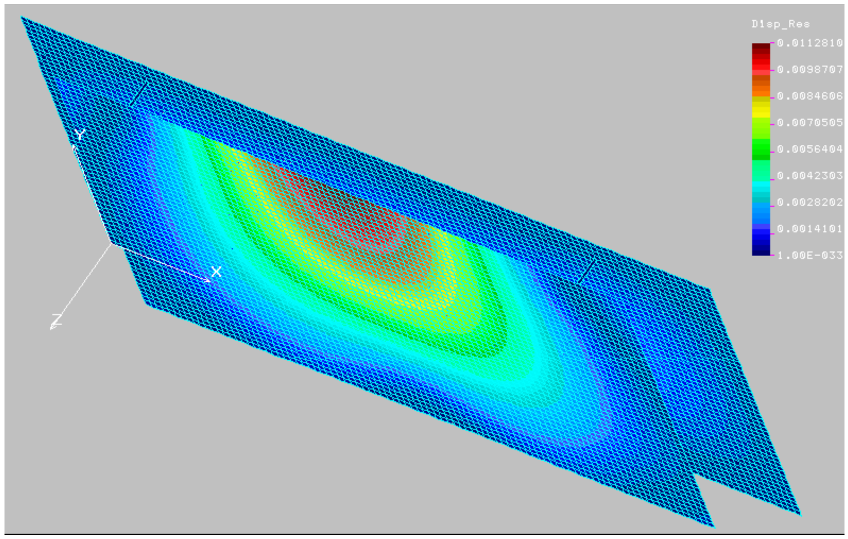
Note: The maximum stresses occur near where the standoffs contact the boards

Figure 6.10: Fixed BC Stress Plot

The stresses obtained for shock loading were 15.9 ksi for fixed boundaries and 17.5 ksi for simply supported boundaries.

It should be noted that these deflections and stresses are far higher than expected. They are even higher than is seen in testing for the current design. The extraordinarily high values of the shock analysis results lead the design team to believe that there is some error in the method of shock analysis. Shock is notably difficult to simulate in a finite element analysis. Therefore, testing of a prototype is really the best way to get more meaningful results. Fortunately, the random vibration analysis seems to give much more realistic results.

In random vibration, the deformed shapes are obviously similar. The displacement plot for the simply supported boundary case can be seen in Figures 6.11 below. It is similar for the fixed boundary case.



Note: The maximum deflections occur near the front edge of the bottom board.

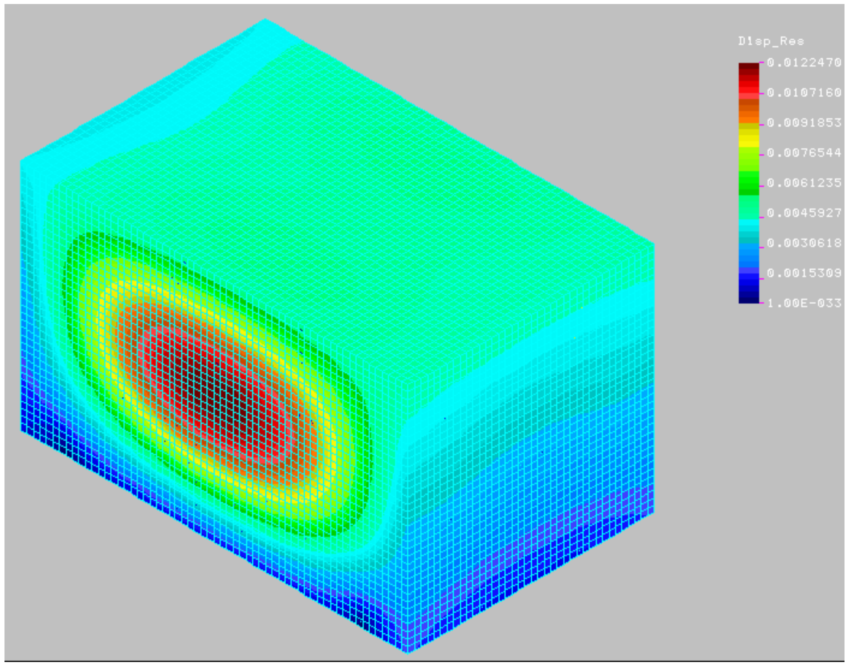
Figure 6.11: Simply Supported BC Deflection Plot

For the simply supported boundary case the maximum deflection was 0.0113 inches. For the fixed boundary case the maximum deflection was 0.0112 inches. These values are much closer to Orbital's predicted 0.020-0.025 inch deflections. The stresses yielded were also more realistic.

Figure 6.10 above shows a stress plot similar to that seen in random vibration. The simply supported boundary case had a maximum stress of 7.33 ksi. The fixed boundary case gave a maximum stress of 7.2 ksi. Using the simply supported case and polyimide yield strength of 17.4 ksi, this yields a factor of safety of 2.38. This is well above Orbital's desired 1.4 factor of safety to yield.

6.2.2 Housing Finite Element Stress and Deflection Analysis

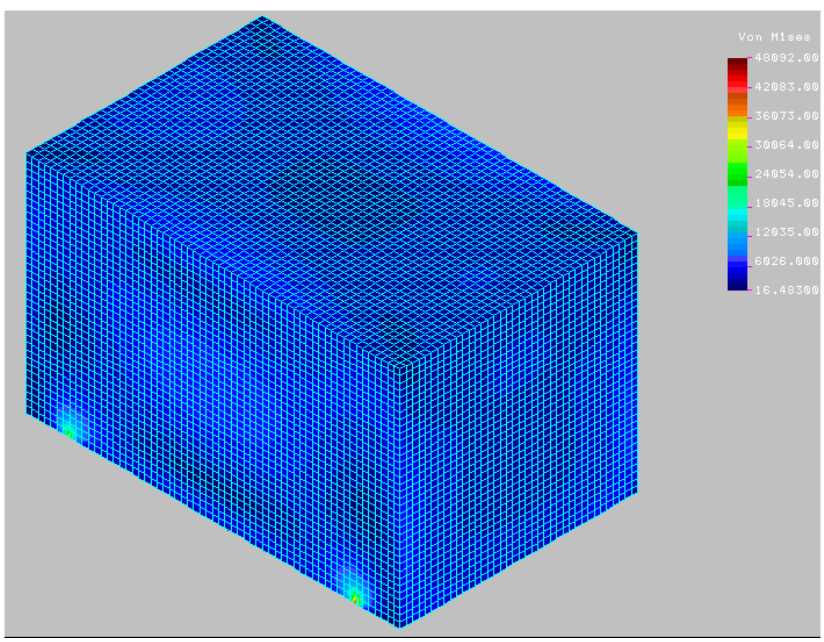
For the housing, modeled with particular nodes fixed where the feet would be, the only goal was to ensure that no large stresses or deflections occurred in the middle panels of the housing. Analyses used a shock acceleration of 2000 G's (772,000 in/s^2) and a random vibration acceleration of 145 G's (55,970 in/s^2). Figure 6.12 below shows the deflection plot attained for the maximum load case (shock, in the Z direction, as shown in Figure 6.5).



Note: Maximum deflections happen in the middle of the front panel.

Figure 6.12: Housing Deflection Plot (Shock)

The maximum shock deflection in the housing is 0.0122 inches. This is a very small deflection for the acceleration levels being introduced. Random vibration only showed a deflection of 0.00089 inches when acceleration was applied in the same direction. Figure 6.13 below shows the stress plot under the same loading.



Note: The highest stresses occur at the fixed nodes.

Figure 6.13: Housing Stress Plot (Shock)

While the initial results of stresses above 48 ksi were initially disconcerting, it makes sense put into context. Instead of constructing feet with bolts through them for finite element analysis, this analysis simply fixed the nodes near where the bolts would line up. As such, the stress in the housing at the feet would actually be spread over a number of nodes where the feet attach. This finite element analysis was just seeking high stresses in the middle of the sides of the box. No part of the box away from the fixed nodes sees stresses above about 18 ksi. This gives a factor of safety over 2, even in shock conditions.

To address fastener issues, hand analyses were performed for the bolts at the interfaces.

6.2.3 Bolt Analysis

A stress analysis was done on the bolts to determine whether the accelerations experienced by the launch vehicle would cause bolt shear or tensile fracture of the bolts. An acceleration of 2000 g. was used throughout the calculation as a conservative estimate. The weight of the entire unit was found to be 2.338 lb. Six size eight, 32 threads per inch bolts are used to secure the unit to the base. Therefore, the force on each bolt was determined to be approximately 780 lb.

Based on the SAE specifications for steel bolts, a minimum SAE grade 4 bolt would need to be used. This would give a minimum shear yield strength of 92 kpsi for each bolt. The force was assumed be applied in single plane shear loading as shown in Figure 6.14 below, causing the maximum shear stress to be about 56 kpsi. This would give us a factor of safety of 1.65.

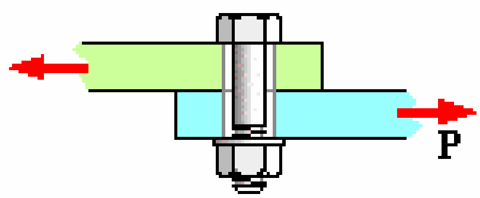


Figure 6.14: Single Plane Shear Assumed for Analysis

7.0 Conclusions

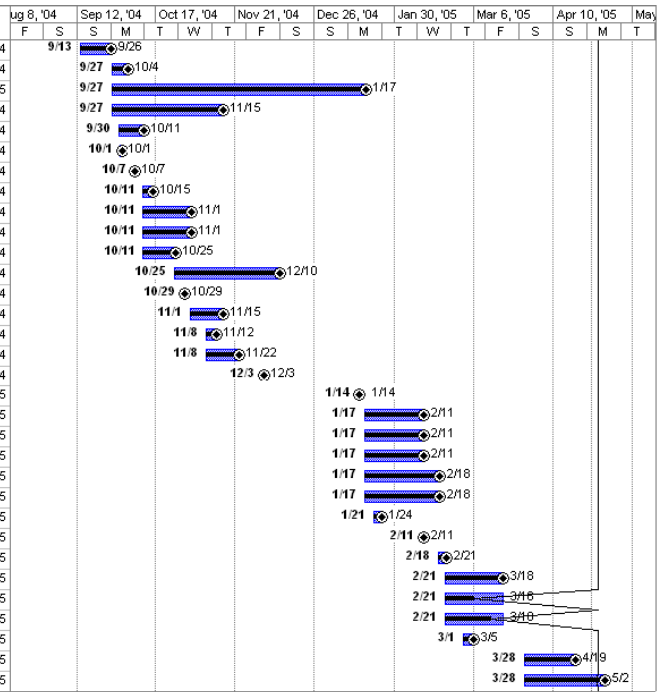
Team Orbital has provided in RPEP an electronics packaging design that addresses the specified optimization requirements and meets the hard requirements. The design fully addresses the main issues of head height and number of modules accommodated. For the optimization parameters, the board layout, I/O connector, mass, manufacturability, and footprint optimization have been fully optimized.

For the vibration requirement, further testing is needed to verify analysis and identify any further issues. For the stress requirement, an assessment of the shock analysis done is necessary to determine if changes to analysis method are needed. Random vibration stresses exceed the factor of safety of 1.4. The thermal requirement requires further investigation. While qualitatively, Team Orbital believes it chose a good design for thermal properties, the thermal finite element analysis was waived with the approval of Orbital. Further Orbital development and testing of the design should provide the necessary insight.

Based on the above results Team Orbital has provided and analyzed a design that Orbital can now implement and test at their behest. Team Orbital has provided a fresh perspective on electronics packaging design while investigating and using industry standards.

8.0 Schedule

ID		Task Name	Duration	Start	Finish
1	✓	Project Memo	14 days	Mon 9/13/04	Sun 9/26/04
2	✓	Project Plan	8 days	Mon 9/27/04	Mon 10/4/04
12	✓	Research	113 days	Mon 9/27/04	Mon 1/17/05
17	✓	Web Page Template	50 days	Mon 9/27/04	Mon 11/15/04
3	✓	Requirements Gathering	12 days	Thu 9/30/04	Mon 10/11/04
7	✓	Time Report 1	1 day	Fri 10/1/04	Fri 10/1/04
4	✓	1st Sponsor Meeting	1 day	Thu 10/7/04	Thu 10/7/04
5	✓	Status Report 1	5 days	Mon 10/11/04	Fri 10/15/04
10	✓	Initial Proposal Presentation	22 days	Mon 10/11/04	Mon 11/1/04
13	✓	Initial Proposal Compilation	22 days	Mon 10/11/04	Mon 11/1/04
15	✓	Brainstorming	15 days	Mon 10/11/04	Mon 10/25/04
14	✓	Design	47 days	Mon 10/25/04	Fri 12/10/04
8	✓	Time Report 2	1 day	Fri 10/29/04	Fri 10/29/04
16	✓	Final Proposal	15 days	Mon 11/1/04	Mon 11/15/04
6	✓	Status Report 2	5 days	Mon 11/8/04	Fri 11/12/04
11	✓	Final Proposal Presentation	15 days	Mon 11/8/04	Mon 11/22/04
9	✓	Time Report 3	1 day	Fri 12/3/04	Fri 12/3/04
31	✓	Initial Downsizing	1 day	Fri 1/14/05	Fri 1/14/05
18	✓	Hand Vibration Analysis	26 days	Mon 1/17/05	Fri 2/11/05
19	✓	Hand Thermal Analysis	26 days	Mon 1/17/05	Fri 2/11/05
20	✓	Hand Shock Analysis	26 days	Mon 1/17/05	Fri 2/11/05
24	✓	Modeling (CAD)	33 days	Mon 1/17/05	Fri 2/18/05
25	✓	Design Document	33 days	Mon 1/17/05	Fri 2/18/05
26	✓	Status Report 3	4 days	Fri 1/21/05	Mon 1/24/05
32	✓	Final Downsizing	1 day	Fri 2/11/05	Fri 2/11/05
27	✓	Status Report 4	4 days	Fri 2/18/05	Mon 2/21/05
21	✓	Modeling Vibration Analysis	26 days	Mon 2/21/05	Fri 3/18/05
22		Modeling Thermal Analysis	26 days	Mon 2/21/05	Fri 3/18/05
23		Modeling Shock Analysis	26 days	Mon 2/21/05	Fri 3/18/05
28	✓	Status Report 5	5 days	Tue 3/1/05	Sat 3/5/05
29	✓	Design Presentation	23 days	Mon 3/28/05	Tue 4/19/05
30	✓	Final Report	36 days	Mon 3/28/05	Mon 5/2/05



9.0 Project Time Expenditures

The total hours put in by team Orbital can be summed up in Table 9.1 below. The majority of the time spent on the design portion of the project was in the CAD modeling and analyses of the designs.

Table 9.1: Project Time Expenditures

Total Hours Spent	
Task	
Travel	24
Project Plan	4
Meetings	99
Research	55
Initial Proposal	10
Final Proposal	5
Web Page	14.5
Developing and Choosing a Design	20
Design Document	7
Analyses	41
Modeling	63
Final Report	23
Final Presentation	15
Total	380.5