Cislunar Explorer Senior Design Report

Structural Redesign and Analysis of 12U Bus

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1.Abstract

This report serves to look into a preliminary structural analysis of the 12U bus for Cislunar Explorers, focusing on the stiffness of the structure with modal analysis on ANSYS. Redesigns of the +/-Y and +/-Z rails, as well as the +/-X shear panels, were completed, where an assembly featuring tentative subsystem components and placements was used for the analysis. This first analysis did not meet the frequency requirements as many of the individual subsystem components reached resonance at much lower frequencies than expected rather than the structure itself. Due to this, it was discovered that proper mounting of components such that they are flat against an existing structure plays a large role in the overall stiffness and modal behavior of the system. This led to the preliminary design of a bulkhead concept, which lies flat against the propulsion tanks, which both increased the overall stiffness of the structure while also provided a surface to allow for proper mounting of the components. Another modal analysis was ran with this concept, where the first two frequency mode was 85.825 and 101.9 Hz, but again this was caused by the incomplete mounting of the flame arrestor, meaning that the actual frequency should be much higher once this is resolved. The first frequency mode that featured entire portions of the structure deforming was mode 3 at 184.65 Hz, indicating that the redesign of the rails, shear panel, and the addition of the bulkhead was a success as this is out of the range of low-frequency vibrations that will be experienced at launch. Thus, the newly designed structures should allow the 12U bus to successfully survive loading in terms of stiffness and it can be concluded that internal mounting structures are necessary.

2. Introduction

Structural analysis of spacecraft is different from traditional ones as the structure isn't static. It isn't ever rigidly attached to anything unless you look at a component locally and once a thruster fires, you can't just simply find the reaction forces and thus the internal forces like you normally would by assuming the net forces equal to zero as there is now an acceleration. This isn't to say there aren't reaction forces at all, as these forces are caused by the thrusters and anything else that isn't the ground. Because of this, a spacecraft is rarely statically-determinate and must employ elastic methods. Luckily, for a Cubesat, these cases don't have to be considered and instead, it is common to ensure that the spacecraft survives launch as that is where the most loads will be faced. This is more true with Cislunar as water propulsion doesn't have a high thrust value, so the structure isn't expected to fail under propulsion firing.

Given that Cislunar is back in the redesign phase, with nearly every subsystem up in the air in terms of design and placement, the main focus of this report will be on the bus structures, including the rails and shear panels. A simplified model of this entire structure was first created to ensure that it could quickly be iterated upon as the placements of certain subsystems become more clear. More analysis is encouraged to be completed with the actual structure down the road to ensure that nothing was overlooked.

After verifying the current state of the bus, the shear panels and rails were redesigned to accommodate thrusters/nozzle as well as combine the two 6U. While this report goes into much more detail on structural analysis, including strength and stability, this is not the main focus, these were only included such that future Structures teams can refer to them. The main focus will still be on stiffness, employing modal analysis on ANSYS. It will be ensured that the redesign of the structure will meet the minimum frequency requirements while also determining if shear panels are necessary on the 6U faces as those have tighter constraints to fit with the dispenser.

If the purpose of reading this report is to get a gist for the analysis, sections 2.3.1-2.3.6 can be skipped as they go extensively into how everything subsystem was modeled which will only really be specific to Cislunar's current state and will quickly be outdated when compared to the other topics discussed here.

3. Background and Theory:

3.1. Face/Axis Naming Conventions:

For the remainder of this report, any references to the +/- X Faces/Rails, +/- Y Faces/Rails, and +/- Z Faces/Rails are in reference to the following convention (Figure 1):

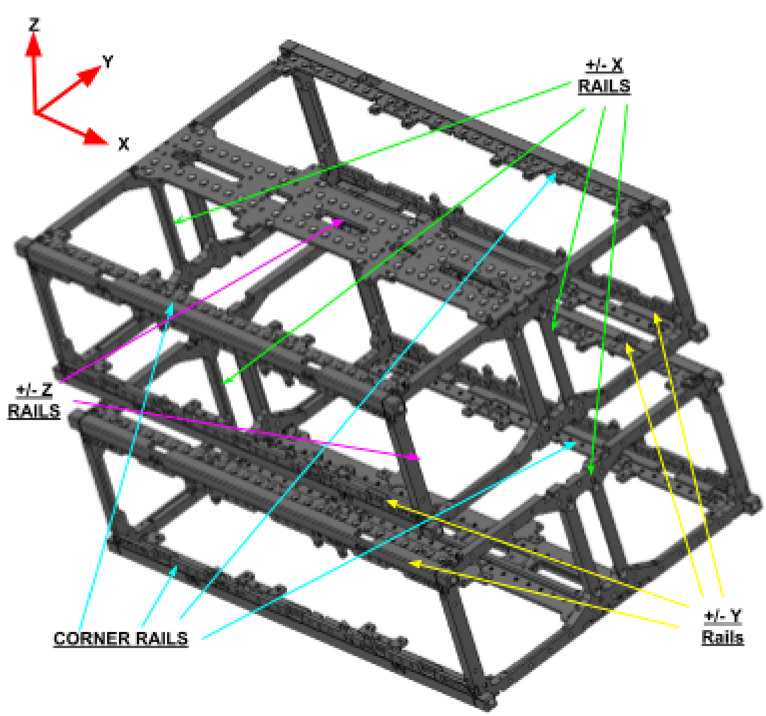


Figure 1: Cislunar Explorers coordinate convention

3.2. Preliminary Calculations:

The following three cases are derived from the 3 main design criteria for space vehicle structures[1]:

1) Strength: In this case, this will be the ability to survive the acceleration load encountered at launch as any propulsion firing/attitude adjustments will impart a minimal force on the entire structure and will be orders of magnitude smaller than the acceleration cases.
2) Stability: Also known as buckling, whether the entire structure has enough resistance to collapse when under the full compression load that will be faced from acceleration.
3) Stiffness: Also known as rigidity and is related to the natural frequency of the assembly, designers need to ensure that the natural frequency of the system is above the natural frequency the launch vehicle is designed for as well as any low-frequency transient vibrations at launch to avoid any resonance ((Launch vehicle providers have a frequency all payloads have to meet).

3.2.1. Strength (Acceleration/Load Factor):

To start the load cases for the Strength and Stability aspect, the total load factor experienced in the structure as determined by the launch vehicle was found. This number is the equivalent factor of gravity to multiply the mass due to the combined vibrations faced.

From the NASA GEVS (General Environmental Verification Standard)'s section on structural loads, the loads faced on the spacecraft during the launch are made up of a combination of loads: Steady-state, low-frequency transient, and higher-frequency vibroacoustic [3]. The combined loading during the launch is summarized in the following equation (Eq.1):

$$N_i = S_i \pm [(L_i)^2 + (R_i)^2]^{\frac{1}{2}} \qquad \text{(Eq.1)}$$

Where N_i is the combined load, S_i is the steady-state load factor, L_i is the low frequency dynamic (LFD) load factor, and R_i is the high-frequency random vibration (HFRV) load factor, with each of them being for the i-th axis (EX: N_x is referring to the combined load around the x-axis, one axis will be referred to the axial axis and the other two will be the lateral axis given the general symmetry of launch vehicles, this will generally be specified in the vehicle's user guide).

Looking closely at this equation, it implies the sum of the load factor around one axis is the sum of all the load factors, where the root sum square of LFD & HFRV load factor must be taken which is similar to taking the magnitude of the components of a vector (or vibration in this case). From NASA's STD-7001 vibroacoustic test criteria, " The relative timing of the transient and random vibration environments is unique for each launch vehicle, but simultaneous occurrence of maximum low frequency transient and maximum random vibration load is improbable. Therefore, a root sum square (RSS) approach is acceptable for combining the maximum low frequency and maximum random vibration loads for the liftoff flight event."[2]

Since the max steady state and dynamic/transient events occur at different times this may provide an overestimate of the max loads. However, due to this being a first pass at the analysis and a higher fidelity analysis being anticipated once subsystems are finalized, the load case used for this report will be coming from the two maximums. Additionally, when designing components, the components shall survive 1.6 times the loads calculated when analyzing to meet the standard F.O.S of 1.4 [2], with an additional 0.2 as an arbitrary choice to account for any potential inaccuracies arising from the analysis that doesn't produce translatable results.

The steady-state load factor and low-frequency dynamic load factors are typically found within a launch vehicle's user guide, with the low-frequency dynamic load often referred to/approximated as the transient load response. In reality, actual spacecraft loads and accelerations are a function of both the launch vehicle and payload structural dynamic properties. They can only be accurately determined via a coupled loads analysis, which combines the launch vehicle and payload and thus is beyond the scope of this report/the stage that Cislunar currently is at. As an alternative, some user payload guides provide a graph of the max combined loads between the steady-state and transient that can be experienced without having to do the coupled loads analysis. In these cases, the biggest load factor is taken if the components of it are not provided. Lastly, many user guides omit specific numbers of random vibrations due to those numbers varying widely based on mission specifications. Luckily a generalized number is given in the GEVS for payloads less than 22.4kg (50lbs) to be used for each mutually perpendicular axis for qualifications (acceptance is lower and should be used for physical testing as opposed to design). Since the max 12U mass is 24kg, this will be taken as being a close approximation and so 14.1g both in the axial and lateral directions will be used (Figure 2)[2].

Table 2.4-3
Generalized Random Vibration Test Levels
Components (ELV)
22.7-kg (50-lb) or less

Frequency (Hz)	ASD Level (g^2/Hz) Qualification	ASD Level (g^2/Hz) Acceptance
20	0.026	0.013
20-50	+6 dB/oct	+6 dB/oct
50-800	0.16	0.08
800-2000	-6 dB/oct	-6 dB/oct
2000	0.026	0.013
Overall	14.1 G_{rms}	10.0 G_{rms}

The acceleration spectral density level may be reduced for components weighing more than 22.7-kg (50 lb) according to:

	Weight in kg	Weight in lb	
dB reduction	= 10 log(W/22.7)	10 log(W/50)	
$ASD_{(50\text{-}800\ Hz)}$	= 0.16•(22.7/W)	0.16•(50/W)	for protoflight
$ASD_{(50\text{-}800\ Hz)}$	= 0.08•(22.7/W)	0.08•(50/W)	for acceptance

Where W = component weight.

The slopes shall be maintained at + and - 6dB/oct for components weighing up to 59-kg (130-lb). Above that weight, the slopes shall be adjusted to maintain an ASD level of 0.01 g^2/Hz at 20 and 2000 Hz.

For components weighing over 182-kg (400-lb), the test specification will be maintained at the level for 182-kg (400 pounds).

Figure 2: GEVs Generalized Random Vibration Test levels (Tabel 2.4-3)

A comprehensive list is tabulated in the table below based on the vehicle's respective user payload guide (All relevant figures for launch vehicles will be under *Appendix→ A1: Launch Vehicle Graphs*):

Table 1: Load Factor of Popular Launch Vehicles

Company	Launch Vehicle	Steady-state load factor	Low-Frequency dynamic (LFD) load factor	High-Frequency random vibration (HFRV) load factor	Combined Load factor
Northrop Grumman	Minotaur 1	X: 11 G Y: 0.5 G Z: 0.5 G	X: 4.73 G Y: 4.19 G Z: 4.19 G	X: 14.1G Y: 14.1 G Z: 14.1 G	$N_x = 11 \pm 14.87$ $N_y = 0.5 \pm 14.71$ $N_z = 0.5 \pm 14.71$
SpaceX	Falcon 9	X: 8.5 G Y: 2G Z: 2G	Combined w/ steady state	X: 14.1G Y: 14.1 G Z: 14.1 G	$N_x = 8.5 \pm 14.1$ $N_y = 2 \pm 14.1$ $N_z = 2 \pm 14.1$
Rocket Lab	Electron	X: 7.5G Y: 1 G Z: 1 G	Combined w/ steady state	X: 14.1G Y: 14.1 G Z: 14.1 G	$N_x = 7.5 \pm 14.1$ $N_y = 1 \pm 14.1$ $N_z = 1 \pm 14.1$
Firefly Aerospace	Alpha	X: 7G Y:0.5G Z:0.5G	Combined w/ steady state	X: 14.1G Y: 14.1G Z: 14.1G	$N_x = 7 \pm 14.1$ $N_y = 0.5 \pm 14.1$ $N_z = 0.5 \pm 14.1$
Relativity Space	Terran 1	X: 6.5 G Y: 0.75G Z: 0.75G	Combined w/ steady state	X: 14.1G Y: 14.1 G Z: 14.1 G	$N_x = 6.5 \pm 14.1$ $N_y = 0.75 \pm 14.1$ $N_z = 0.75 \pm 14.1$

While Minotaur 1 has the highest load factors (Table 1), it's an outlier such that the steady-state and LFD load factors were provided separately instead of being in a chart like the other launch vehicles. While this is no indication of the validity of the calculations that would've been made with Minotaur as the max load case, Falcon 9 will be selected due to the method of choosing the load factor based on a graph being more common among the other vehicles. The max load factors are calculated below (Eq.2):

$$Max\ Load\ Factor\ Magnitude = \sqrt{N_x^2 + N_y^2 + N_z^2} \quad \text{(Eq.2)}$$

$$Minotaur\ 1_{Max\ Load\ Factor\ Magnitude} = \sqrt{(11 + 14.87)^2 + (0.5 + 14.71)^2 + (0.5 + 14.71)^2} = 33.64$$

$$Falcon\ 9_{Max\ Load\ Factor\ Magnitude} = \sqrt{(8.5 + 14.1)^2 + (2 + 14.1)^2 + (2 + 14.1)^2} = 32.08$$

$$Percent\ Difference = \frac{|Value1 - Value2|}{(Value1 + Value2)/2} * 100\% = \frac{33.64 - 32.08}{(33.64 - 32.08)/2} * 100\% = 4.75\%$$

As shown by the calculations, this assumption is a good approximation as the values between the Falcon 9 and Minotaur load factors are less than 5% off. Additionally, looking at the equation for the combined load factor, the LFD load factor and the HFRV load factors components are within the root sum square portion, meaning that the HFRV component will dominate the entire term making the difference between a load factor with and without the LFD component negligible in this case. This is all to say that we can safely proceed with Falcon 9 as the max load case even though Minotaur's calculation is slightly bigger.

Thus, the Axial acceleration of 22.6g. and lateral acceleration of 16.1g will be applied to the ANSYS models.

3.2.2. Stability (Buckling)

While this will not be used in this report, having calculated the load factor, the force on the structure can be calculated to be used to ensure it would not buckle.
With the load factors calculated, the total applied forces can be found via the following (Eq.3) [4]:

$$F_{applied} = F_{satelite} + F_{springs} \quad \text{(Eq.3)}$$

Where $F_{satelite}$ is the weight of the satellite structure under the load factor and F_{spring} is the force of the springs/launching mechanism as determined by the launch vehicle providers. However, the force of the spring/launching mechanism isn't typically given, nor do all dispensers feature springs. Thus this will be ignored for now until further information on the dispenser is provided (once one has been provided and if it uses springs, this can be calculated by finding the spring constant and approximating displacement). The total force $F_{applied}$ will be distributed along the rails or X surface as a reaction force back onto the satellite (depending on which one seems more accurate once more research has been done on Buckling Analysis on ANSYS. The approximate max mass for a 12U will be used to simulate the final design, which is 24kg (Eq.4).

$$F_{satellite} = m_{satellite} \cdot Magnitude_{Load\ Factor} \cdot g = 24kg \cdot 32.08 \cdot 9.81\frac{m}{s^2} = 7552.9152N \quad \text{(Eq.4)}$$

3.2.3. Stiffness (Modal & Frequency)

This is the most important analysis at this stage and will be the focus given the nature of the redesigns within the scope of this report. During the launch, there exist three environmental factors that induce vibrations that translate throughout the structure and to the payload. From SMAD chapter 27, "These three physical transitions [axial compression of vehicle initial launch, engine thrust from chamber pressure, switch from pinned-free to free-free causing random compression/tension] work in combination to induce structural vibrations that include the bottom 50Hz – 60 Hz of the vibration spectrum. Their amplitude will be stable or transient, hopefully, but they'll be there and they have to be considered."[4]. This means that when designing a payload, it is extremely important to ensure the first frequency mode is above the transient frequency ranges that will be experienced at launch. Doing so will avoid large deformations in the payload structure that may prove to be fatal to the mission. Additionally, a CubeSat with a low natural frequency may interfere with the attitude control of the launch vehicle itself since the sensors may interpret the resonance of the CubeSat as being disturbances that need to be adjusted. Due to this, many launch vehicles provide a minimum natural frequency needed to be met.

This design constraint applies to everything, from the launch vehicle structure itself (Figure 3) to any small mechanisms on the satellite (Figure 4), but most importantly for Cislunar's case, the entire CubeSat bus structure.[1].

TABLE 18-9. Fundamental Frequencies for Spacecraft Design. The booster adapter and spacecraft structure should be designed for fundamental frequencies greater than or equal to those shown.

Launch System	Fundamental Frequency (Hz)	
	Axial	Lateral
Atlas II, IIA, IIAS	15	10
Ariane 4	*	10
Delta 6925/7925	35	15
Long March 2E	26	10
Pegasus, XL	18	18
Proton	30	15
Space Shuttle	13	13
Titan II	24	10

* 31 Hz for dual payloads, 18 Hz for single payloads.

Figure 3: Fundamental Frequencies of Common Launch Systems (Table 18-9)

Table 22-7. Key Performance Parameters for Mechanisms.

Requirement	Discussion
Performance	Specific description of what the mechanism needs to accomplish
Motion Profile	Position, velocity and/or acceleration requirements Can be simple point-to-point or complex programmable profiles
Accuracy, Resolution and Stability	Allowable errors. Usually stated in deg/arcsec for position
Mass and Power	Allowable assembly mass and peak/average power draw
Life	Operational, ground test and storage life
Reliability and Redundancy	Numerical probability of success for specified life
Temperature Range	Operational, non-op and survival temperatures
Vacuum Environment	All space devices require operation in hard vacuum of space Drives selection of materials for low outgassing
Stiffness	First resonant mode of assembly Must be high enough to avoid resonance with launch vehicle vibration environment Deployed frequency must be high enough not to impact ACS
Design Loads and Random Environment	Quasi-static acceleration loads, random vibration, and shock environments Safety factors used for design & derivation of qualification / acceptance test levels
Interface Definition	Mechanical and electrical interfaces and allowable envelope
Contamination Control	Proximity to sensitive space instruments levies specific requirements on cleaning and assembly controls

Table 22-7, Fig. 22-13, Eq. 22-15◄

Figure 4: Key Performance Parameters for Payload Mechanisms (Table 22-7)

While the natural frequency that the CubeSat's first frequency mode constraint is provided by the launch vehicle, a general number can be found in the graphic provided by NASA in the case a launch vehicle hasn't been selected (Figure 5)[5].

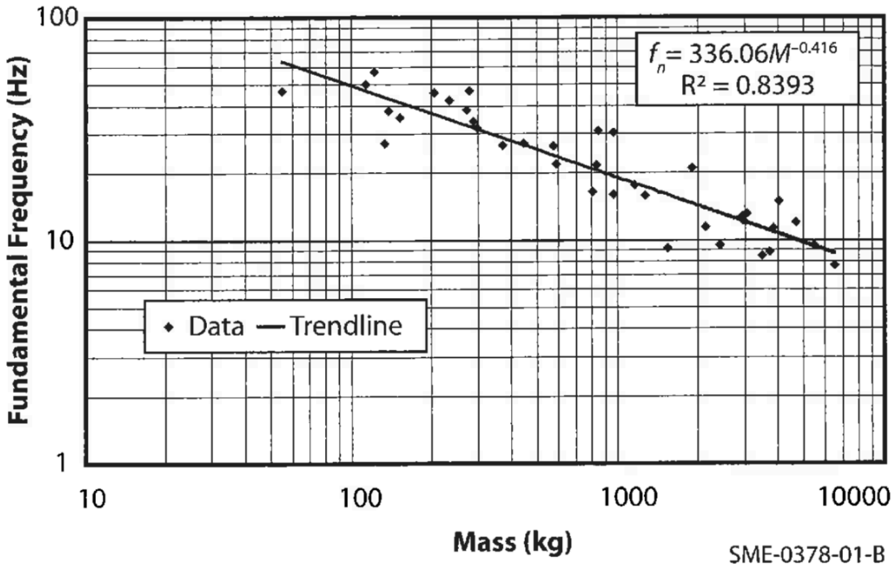


Fig. 22-4. Comparison of Observatory Launch Mass Versus Fundamental Frequency. (Graphic courtesy of NASA)

Figure 5: Mass vs Fundamental (Natural) Frequency at Launch

While the original graphic did provide a trendline, this linear relationship was extended to reach the mass range for a 12U Cubesat (~24kg). Since the graph is on a log-log scale, it is hard to accurately determine where on the horizontal axis corresponds to this 24kg, but it will be approximated as right in the middle of 20-30kg, which corresponds to a fundamental frequency of ~90Hz (Figure 6).

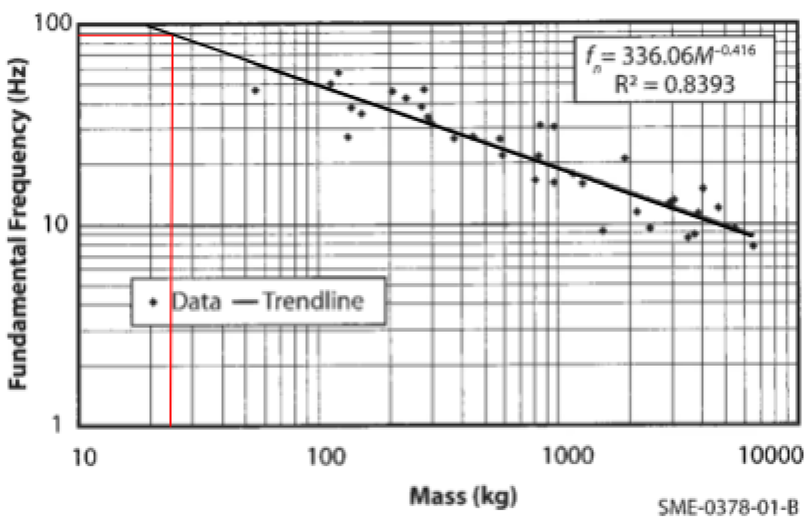


Fig. 22-4. Comparison of Observatory Launch Mass Versus Fundamental Frequency. (Graphic courtesy of NASA)

Figure 6: Mass vs Fundamental (Natural) Frequency at Launch (Interpolated)

With this number in hand, it means all modal analyses of the fully assembled Cubesat should result in a first frequency mode of greater than 80-90Hz to consider any structural redesign to be acceptable in a stiffness sense. In this report, a cantilevered (fixed at one end and free on the other) modal analysis of the entire structure will be employed due to the assumption that during launch, the bottom face would be treated as fixed to the bottom face of the dispenser. In reality, the analysis should be what is called a "free-free" analysis, where the model is free of any constraints (so not fixed at any points) as well as free of any forces. This is because after liftoff, while the entire launch vehicle is constantly experiencing forces from the engine thrust and surrounding pressures, the payload within its dispenser is free from these forces and isn't rigidly attached to the dispenser or else it wouldn't be able to deploy.

Since this condition isn't possible to simulate on ANSYS and not enough time was dedicated to finding a method, a cantilevered approach can be used as a replacement during design and analysis, with the knowledge that the true frequencies of each mode are approximately half of what it would be with a free-free analysis. This can be seen more clearly given the tabulated fundamental frequencies of a beam under different boundary conditions which will only be used as a guide rather than for any exact calculations (Figure 7):

Configuration	Fundamental Frequency
Cantilever (Fixed-Free)	$\frac{1}{2\pi}\left[\frac{3.5156}{L^2}\right]\sqrt{EI/\rho_L}$
Cantilever with end mass	$\frac{1}{2\pi}\sqrt{\frac{3EI}{(0.2235\, m_{beam} + m_{end})\, L^3}}$
Pinned-Pinned	$\frac{1}{2\pi}\left(\frac{\pi}{L}\right)^2\sqrt{EI/\rho_L}$
Free-Free & Fixed-Fixed	$\frac{1}{2\pi}\left[\frac{22.373}{L^2}\right]\sqrt{EI/\rho_L}$
Fixed-Pinned	$\frac{1}{2\pi}\left[\frac{15.418}{L^2}\right]\sqrt{EI/\rho_L}$

Figure 7: Beam Bending Fundamental Frequencies

Where L refers to the length of the beam, E is Young's modulus of the material of the beam, I is the moment of inertia of the beam's cross-section, and ρ_L is the mass per unit length[6].

When comparing two identical beams under the Cantilevered (Fixed-Free) and Free-Free boundary conditions, we obtain the following relation for the fundamental frequencies (Eq.5):

$$\frac{\omega_{n,Fixed-Free}}{\omega_{n,Free-Free}} = \frac{\frac{1}{2\pi}\left[\frac{3.5156}{L^2}\right]\sqrt{\frac{EI}{\rho_L}}}{\frac{1}{2\pi}\left[\frac{22.373}{L^2}\right]\sqrt{\frac{EI}{\rho_L}}} = \frac{3.5156}{22.373} = 0.1571 \qquad \text{(Eq.5)}$$

This, however, is overly optimistic but still shows that employing a Fixed-Free analysis that meets the minimum frequency (Figure 6) has some built-in safety factor. In conclusion, a

Free-Free analysis with the same beam would return a higher fundamental frequency but since a CubeSat's structure is much more complicated than a beam with uniform cross sections, the relationship is approximately the following instead (Eq.6):

$$\omega_{n,Fixed-Free} \approx 0.5\omega_{n,Free-Free} \quad \text{(Eq.6)}$$

3.3. Preliminary-ANSYS work:

The shear panels and rails are planning on being redesigned due to accommodations needed for the propulsion nozzle and ACS thrusters, as well as being potentially a more seamless method of combining two 6Us than designing mounts. Due to this, a modal analysis where the structure was cantilevered was conducted on one of the original 6U by Spacemind to help identify certain weak points with the current shear panels/rails (if any) and provide guidance towards how to best remedy them. This was performed on the 6U structure as opposed to a 12U form factor due to inserting a connection method between the 2 6U into the model potentially taking away from the rails and shear panels themselves and wouldn't be super helpful. The first three resonance modes are shown below (Figures 8-10):

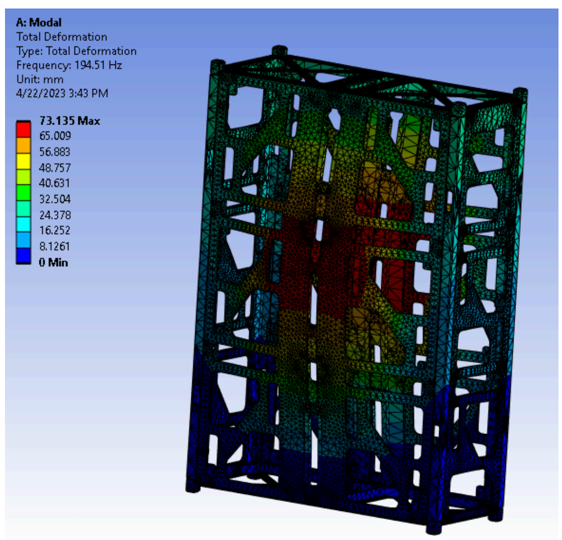


Figure 8: First frequency mode of Spacemind 6U Bus

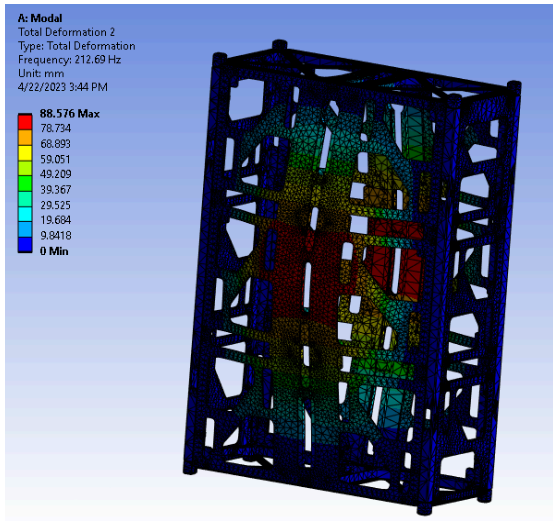


Figure 9: Second frequency mode of Spacemind 6U Bus

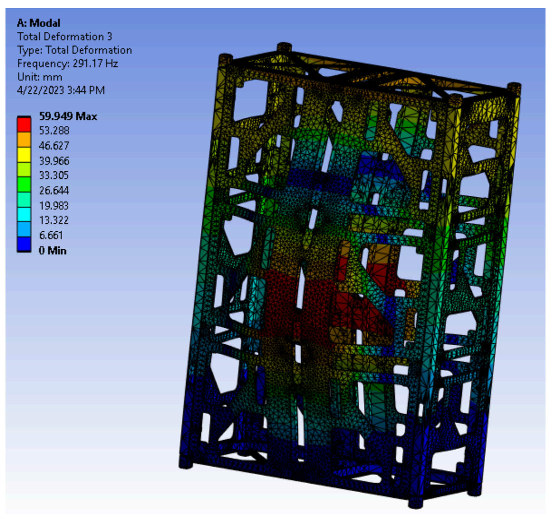


Figure 10: Third frequency mode of Spacemind 6U Bus

The three resonance modes all happened within 100 Hz of each other, starting at about 194.51Hz to 212.69Hz to 291.17Hz. Since this analysis was done by fixing one end of the satellite, this frequency should be about half what it is in a free-free analysis. The frequency isn't particularly useful as this wouldn't translate to the 12U. What is interesting is the displacements focusing around the center of the 6U face, indicating that the +/-Y Axis and +/- Z axis should be focused on as those will be the 6U faces on the 12U while the +/-Z axis should be relatively unaffected.

To get the most accurate results for the analyses, it would be ideal to get as many of the subsystem components into the model as possible. But since these analyses are being done before any finalizations, their approximate shapes and mass (according to data sheets/manufacturer information), as well as the relative placement will be estimated to ensure equal distributions for a center of mass relatively equal to the geometric center of the Cubesat, with the masses being assigned according to the most up to date mass budget at this time. The placements of each component will be specified by the respective subteams if necessary, such as ACS sensors/thrusters, but otherwise, they will be placed to best balance the mass.

The main purpose of adding these subsystems ahead of any finalization is due to the natural frequency inherently being related to both the stiffness and distribution of masses of the system. Looking at a multi-Degree of freedom vibrations model (Eq.7):

$$\lambda^2[M]\phi + [K]\phi = 0 \qquad \text{(Eq.7)}$$

Where λ^2 represents the eigenvalues found from solving the generalized eigenvalue problem (the natural frequency squared), $[M]$ is the mass matrix, $[K]$ is the stiffness matrix, and ϕ are the eigenvectors of the system.

From this equation, where the eigenvectors will not change for a given system, changes to the mass matrix must result in inverse changes to the eigenvalues (natural frequencies) assuming constant stiffness (matrix). The inverse is also true, where changes to the stiffness matrix without largely impacting the mass matrix will also result in inverse changes to the natural frequency. For a more simplified model, looking at the natural frequency for one degree of freedom also shows this relationship (Eq.8):

$$\omega_n = \sqrt{\frac{k}{m}} \qquad \text{(Eq.8)}$$

Once again, increasing the mass will result in a lower natural frequency, assuming that the increase in mass has a negligible impact on the stiffness. Rarely can you decouple the mass and stiffness of the system such that changes to one will not change the other. This concludes that adding subsystems and therefore increasing the components of the mass matrix and overall mass of the bus, is essential in obtaining an analysis with the worst-case natural frequency. Doing the same analysis without any subsystems in place would result in inflated natural frequency numbers and inaccurate frequency modes.

When making these mass models, the key thing is to get the center of gravity and mounting points as close to the original as possible. The center of gravity will inform the distribution of loads while the mounting will replicate the conditions for frequency modes (Table 7). While there are other properties that plays a role in attempting to match the first frequency of the object, many of these components are relatively small and thus naturally stiff so a high fidelity mass model is not necessary.

3.3.1. Spacemind's Rails Simplification:

When doing ANSYS work, it is typical to remove any minor features (such as small chamfers and fillets) since they would have little effect on the overall analysis. Additionally, small holes are also typically removed from the model. Both of these were done to save on computational costs when meshing and solving the model and thus provide a faster turnaround time for iterations. This will be done on the +/-X Axis rails as well as the corner rails, but not with the +/-Z rails nor the +/-Y rails by Spacemind as those will be redesigned.

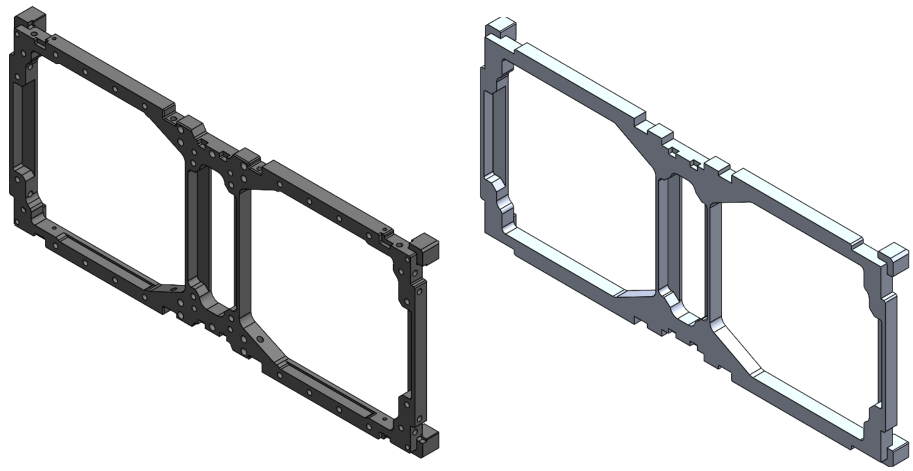


Figure 11: Spacemind +/- X rails vs +/-X Rails ANSYS

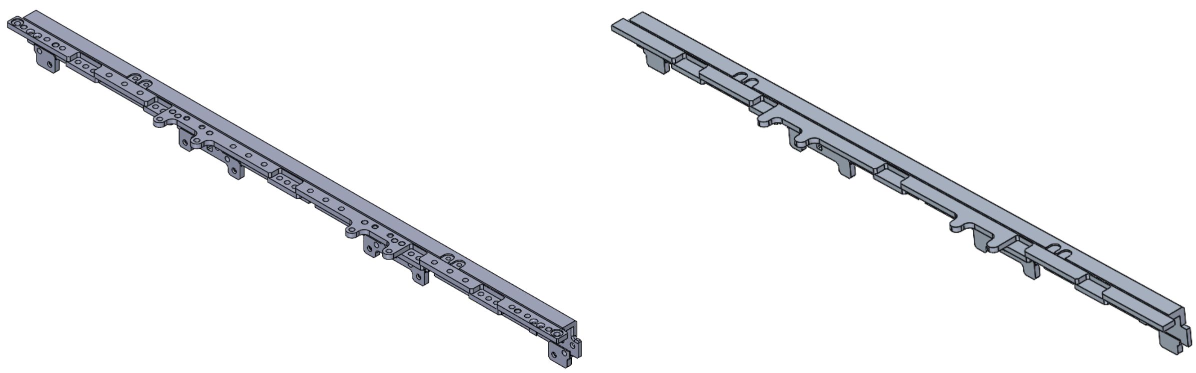


Figure 12: Spacemind Corner Rails vs Corner Rails ANSYS

NOTE: Removing mounting holes is reasonable since the mounting holes would have stainless steel bolts in them, which is stronger than the aluminum of the provided bus structure. This makes it such that for the aluminum around the mounting holes with bolts in them to deform, the steel would have to deform first. And rather than having to add a fixed constraint into every hole that would have a bolt in them, which is still not yet known which holes would be used for mounting, it is common to just remove them for an ANSYS model early on in the design process as the stronger steel that would be there is being replaced by the weaker aluminum.

3.3.2. Propulsion Simplifications:

The current propulsion system is modeled as two cylindrical propellant tanks, with pipes connecting the two to the combustion chamber/nozzle assembly. Due to the sizing and shape of the sub-system, current and future configurations will likely see the tanks on opposite ends of the 12U along the X-Axis. The combustion chamber, flame arrester, and nozzle will sit in the geometric center, and the nozzle flushed against the -Z axis.

Initially, a wet mass of 5kg was provided but after doing some preliminary sizing of the propulsion tank using this number, it was discovered that the resulting propulsion tanks would take up more than half of the volume, and thus not possible for the scope of this mission (at this time, a new wet mass has not been provided). A MATLAB script was written to test a range of radiuses, thicknesses, and heights of cylindrical propellant tanks, based on two different materials (Titanium and Aluminum) with both a thin-walled and a thick-walled pressure vessel assumption, all at a given wet mass. Through this, it was determined that approximately 3kg is most likely the max that can be fitted to provide enough space for other subsystems without having to design irregularly shaped tanks. It was also found that a thin wall approximation is appropriate, which would be helpful for future iterations of the actual propellant tank design. More information on the MATLAB script can be found in *Appendix → A.2: Proptank MATLAB*.

At this wet mass, to best dedicate space towards the middle, a titanium tank with an inner radius of 90mm, height of 63.7mm, with a shell thickness of 1mm (0.059mm according to MATLAB but this seemed too optimistic so it was bumped up), would best fit the predicted max pressure of the inside tank while maintaining a factor of safety of 1.5. The resulting model on Solidworks returns 1890.85g, where 1500g was added onto the original mass of the tanks provided by SolidWorks to account for the mass of water (Figure 13). However, these tanks, while accounting for 8% ullage, did not account for any extra height needed to fit electrolyzers or the additional mass and will need to be updated once those are selected and finalized.

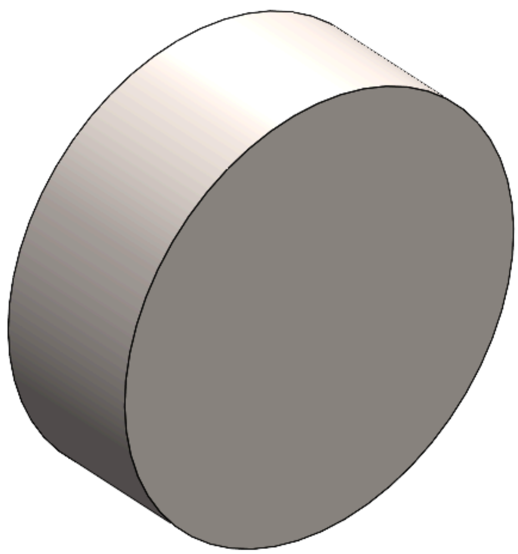


Figure 13: Propulsion Tank (Simplified)

The combustion chamber's dimensions were measured physically from the old Cislunar Explorer 6U, which were 68 x 75.5x54mm. According to the mass budget from the 6U, the total mass of the chamber was about 460g with the material being titanium. Due to this, the thickness of the chamber was varied while keeping the major dimensions mentioned until a mass of about 460g was achieved, resulting in a wall thickness of 4.7mm and a mass of 462.15g according to Solidworks. Holes were extruded on four different faces for connections of pipes from the propulsion tank to the chamber, as well as the check valve for the nozzle and flame arrestor, bringing the total mass down to 453.5g (Figure 14).

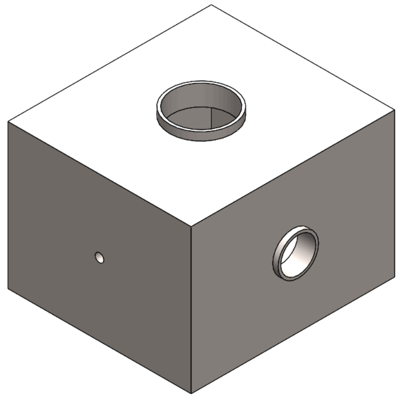


Figure 14: Combustion Chamber (Simplified)

The nozzle and check valve were both taken from the previous 6U's CAD, with their approximate geometry simplified but still retaining the material and major dimensions. The nozzle from the 6U was made from Ti-6Al and had a mass of 31.03g. The final simplified nozzle reached a mass of 27.31g (Figure 15). The check valve from the 6U was made from 316 stainless steel and had a mass of 160.91g The final simplified check valve had a mass of 145.51g (Figure 16).

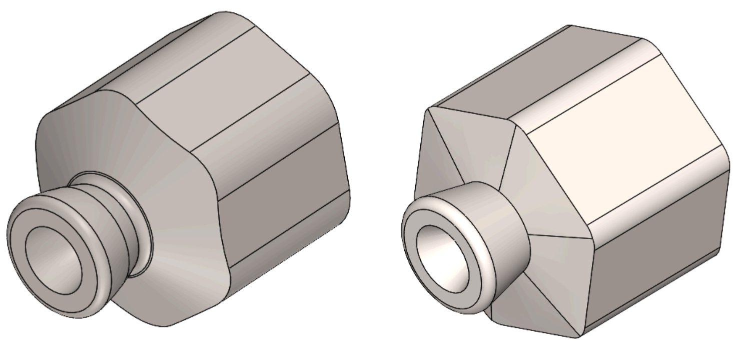


Figure 15: 6U Nozzle vs Propulsion Nozzle (Simplified)

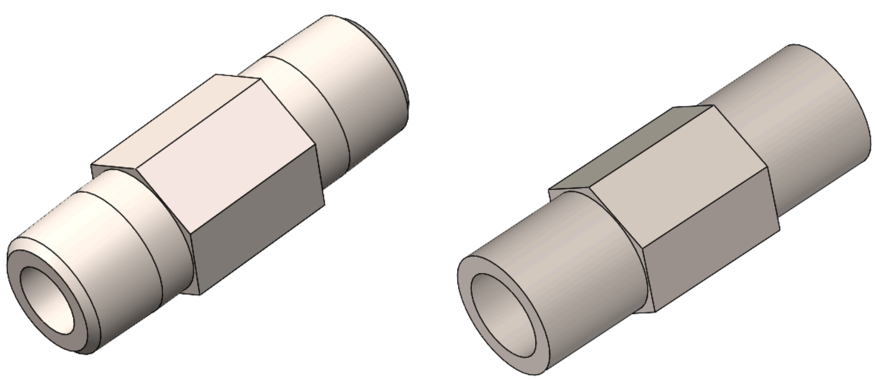


Figure 16: 6U Check Valve vs Propulsion Check Valve (Simplified)

The Flame arrestor's general dimensions are given on the WITT FLAME ARRESTOR FLASHBACK ARRESTOR 85-10 datasheet. The material is brass and the cylinder base follows the major dimensions, with the inside being subjected to the shell tool with various thicknesses until the overall part reaches a mass of around 15.31 oz (434.0312g). A wall thickness of 5.5mm results in a mass of 432.26g. From there, an elbow connector was added to the end to more accurately depict how it would attach to the combustion chamber. While the connector would add both masses and change the dimensions from the data sheet, the part before adding this should be accurate. The final mass with the elbow is 447.21g (Figure 17).

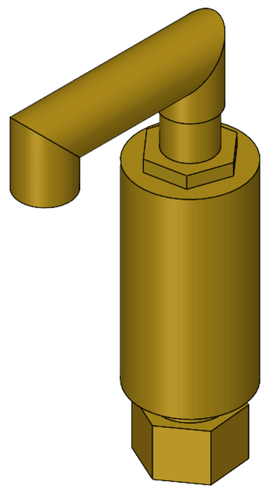


Figure 17: Flame Arrestor (Simplified)

The pipes that connect the chamber to the tanks are stainless steel tubes with an inner diameter of ⅛ inch which corresponds to a 0.028in wall thickness. Placing the combustion chamber equidistant from both tanks, the total length of piping needed is 53.3mm.

3.3.3. Flight Software/Avionics/Communication Simplifications:

While the communications and avionics as a whole add up to a nonnegligible amount of mass, estimated to be about less than 500g combined, each component within the respective systems is much less. Since these include mostly Raspberry Pis, cameras, multiplexers, radio boards, power amplifiers, antennas, linear noise amplifiers, and other PCBs, it doesn't make sense to model these for this report as trades are still being done. Additionally, randomly placing the components without some sort of electronics box or thought into their attachments will also result in an inaccurate modal analysis as those electronics will hit a resonance frequency much lower than the actual structure. Due to this, all these components will be ignored for now. For completion's sake, it will be mentioned that Flight software has no hardware and relies on the hardware from avionics.

3.3.4. Power Simplifications

The power subsystem will be the flight battery is chosen and solar panels. The most promising battery is the Nanopower BP8 by GOMspace, with a mass of 485g and dimensions of 94.8 x 95 x 42mm. Since the casing is most likely made out of some blend of ABS plastic, that will be the material used. The battery was originally planned on being modeled as a solid rectangular box of ABS and then shelled down to the correct mass, but since the starting mass was much less than 485g (386.22g), the mass was overridden instead to match the datasheet (Figure 18).

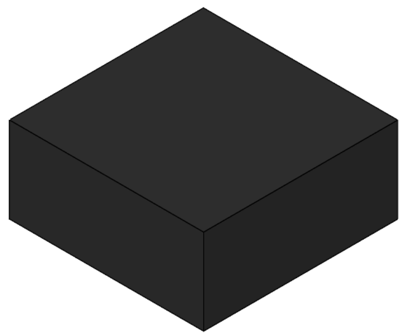


Figure 18: GOMspace Nanopower BP8 (Simplified)

The solar panels will be modeled as fiberglass since that is the preferred material for high-quality PCBs, matching the surface dimensions of the respective shear panel that it sits on with a thickness of 1.5mm as given from the previous 6U version of Cislunar. However, while this was modeled and put onto the assembly, it was decided that they should be suppressed and not included in the actual modal analysis as they provide minimal stiffness and shouldn't be expected to affect the results. Additionally, the design of the actual panels will not be a solid block of fiberglass, so for ease of iteration/computational time, they will be ignored.

3.3.5. ACS Simplifications

ACS will be made up of 6 thrusters, with the most promising option being the 100mN HPGP by Bradford ECAPS, weighing about 0.04 kg each. These thrusters come with a propellant tank but since at this time, a mass for the propellant cannot be provided and thus the tank cannot be properly sized, they will have to be ignored and revisited. The thrusters will be modeled as a circular base assumed to be the size of a quarter with a protruding nozzle. NOTE: The material could not be found and was assumed to be aluminum 6061. ACS will most likely end up making their thrusters and these should be updated accordingly.

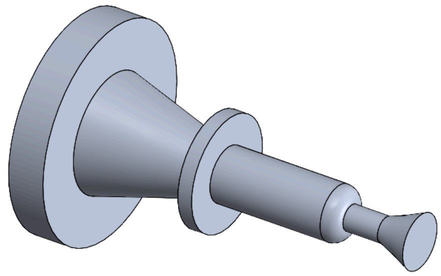


Figure 19: Bradford 100mN HPGP Thruster (Simplified)

In addition to the thrusters, 3 sensors in each category necessary will be included. The spinning sun sensor by Redwire is 109 g with a dimension of 66 x 35 x 25mm, while the associated electronics have a max mass of 725g, and dimensions of 51 x 81 x 89mm. Both items will be modeled as aluminum and shelled to the thickness necessary to reach the masses shown on the datasheet (Figure 20). The electronics box was shelled with a thickness of 12mm to result in a mass of 722g while the sensor was shelled with a thickness of 6mm to result in a mass of 107g (Figure 20). This sensor must be placed on the + Z side (opposite propulsion nozzle).

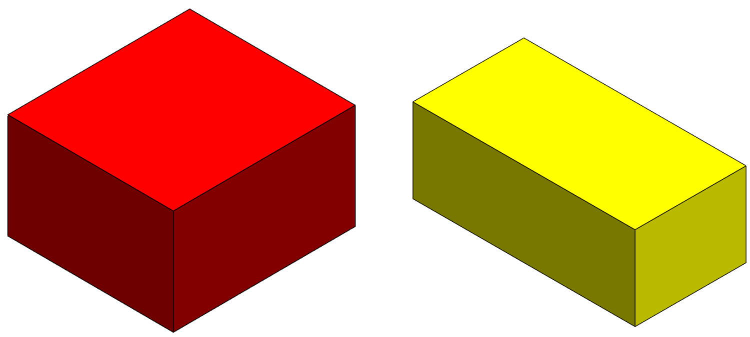


Figure 20: Redwire Spinning Sun Sensor Electronics (Left) and Sensor (Right) (Simplified)

The Horizon Crossing Indicator (HCI) sensor of choice will be the Mini Digital HCI by SERVO, which weighs 50g with a dimension of 1 x 1.375 x 1.15 in (25.4 x 34.925 x 31.75 mm). The HCI was modeled as aluminum 6061-T6 and had a mass of 25.59g before overwriting to the correct mass (Figure 21). This sensor must be placed on one of the +/-Y sides.

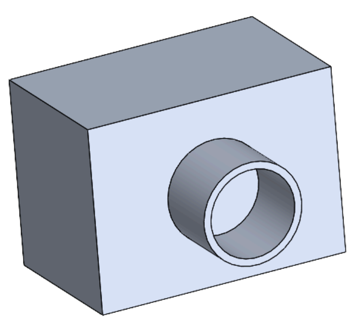


Figure 21:SERVO Mini Digital Horizon Crossing Indicator (HCI)

The GPS by swift Navigation weighs 26g and is about 48 x 71 x 12.4mm. Since this is mostly a PCB, it will be modeled as fiberglass and just overwritten to the mass needed (Figure 22).

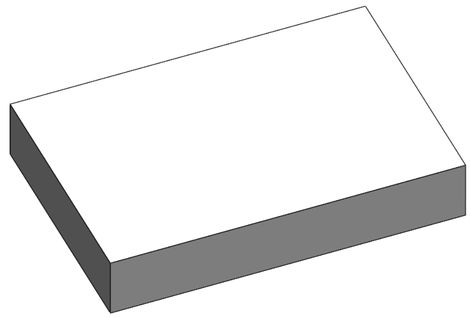


Figure 22: Swift Navigation GPS

3.3.6. DISCLAIMER

Admittedly, it's not certain if the methods used to model these systems is the best one. The method of shelling the systems that are boxes stems from the idea that since, certainly, it isn’t a solid block of the material, shelling it to the thickness that returns the mass given from the source should give a somewhat more accurate mass distribution/structural properties. This is because, with things like electronic boxes, it's only the outer enclosure that matters in structural analysis as they should be the first to break and are designed to protect what's inside. The simplifications also did not consider the extra masses of mounting components (fasteners, epoxy, etc) nor harnessing, which once again is just not possible to accurately depict at this stage. Lastly, these subsystems may not end up being accurate in terms of the material as many of the data sheets do not provide this information and a guess was made based on similar components sharing the material type or just by considering the mass, utility, loads faced, to make a final educated guess. In the future, these models should be updated as we obtain more information.

4. Shear Panel/Rail Redesigns:

4.1. Requirements

Some requirements are specified by Cubesat development standards (CDS) by Cal Poly, which provides some generalized guidelines that are often referenced by NASA[7]. As we do not currently have a launch provider/dispenser, the generalized requirements that are fit to meet a variety of missions will be referenced. Once one has been selected, these requirements would potentially change to fit the specifications of the provider.
NOTE: These requirements should be renumbered according to the current naming convention

Table 2: Requirements

Number	Requirement	Rationale	Parent	Child
STR-1	The main structures shall meet a minimum safety factor of 1.6 for analysis	Ensure that the redesigned rails and shear panels will still meet the minimum requirements while still providing some leeway when conducting simulated environment testing	Peck's Lecture	
STR-2	The Cubesat shall have a minimum of ~80-90 Hz for the first frequency mode when conducting a cantilevered (fixed-free) analysis	This is the minimum value needed according to Fig 6. Even though the minimum actual minimum is ~50Hz for a free-free analysis as provided by Professor Peck, a simpler analysis could be done where the Cubesat is cantilevered from one end and the resulting natural frequency will be half of what it would've been if it was free on both ends. Thus, this requirement has a built-in safety factor of at least 2 if met. Discussed more in *Background and Theory→Preliminary Calculations → Stiffness (Modal & Frequency)* Section.		
STR-3	The main structures shall be made from aluminum alloy	This requirement is derived from the Cubesat development standards (CDS). Aluminum 7075, 6061, 6082, 5005, and/or 5052 are used for the main structures (shear panel/rails) and are widely accepted across launch providers/dispensers; other materials will be allowed on a vehicle-to-vehicle basis. Since neither has been selected, this will be the default.	CDS-2.2.12.1	
STR-4	Rails on +/-Y and +/-Z Axis rails shall have a minimum of 2 (3 if propulsion nozzle side) openings	This requirement is such that the designed rails will accommodate all the thrusters needed for the respected faces, whether it's ACS or prop.		
STR-5	Rails on +/- Y shall be designed such that it combines the two 6Us	This requirement is derived from the decision to proceed with using the two 6U provided by Spacemind. Corner mounts were investigated but not chosen; rails will provide a better method of combination.		
STR-6	The structure shall survive random vibration testing	This requirement is from the CDS and depends on the launch vehicle. While this wouldn't apply to this report, it will be noted for future reference once a launch vehicle is selected	CDS-3.3.1	

4.2. +/- X Shear panels

Looking at the analysis done on the 6U back in section 3.3, the +/-X sides do not seem to face much deformation in the modal analysis at resonance and the stiffness mostly comes from the 6U faces (+/-Y & +/- Z).Thus these shear panels would likely not do much to affect the first couple of frequency modes. This implies that dedicating time to increasing the stiffness of these panels may have little impact overall. The main focus would be to withstand acceleration loads along with the force of the springs from the dispenser, in addition to combining the two 6Us. This resulted in a simple design, where most of the material has been removed, leaving only what's necessary for bolt holes for mounting onto the rails of the CubeSat (not shown in the figure). The final design has a mass of 47.62g and is made from aluminum 6061 (Figure 23):

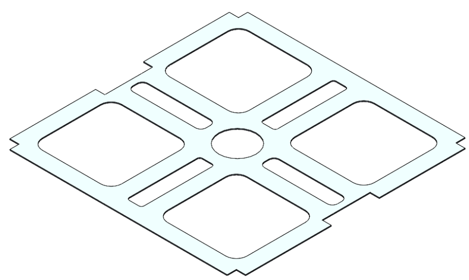


Figure 23: +/- X Shear Panels (Redesign)

4.3. +/- Y Rails

Since the +/- Y axis requires the redesign of both the rails and shear panel, the rails will be designed first as that is what provides the majority of the stiffness with the panels only enhancing it. Additionally, while it isn't ideal to redesign components of a bus that was already designed, it was decided that the rails will combine the 2 separate 6Us into one. There was a trade study done on methods of connections during the Fall of 2022, where 4 corner mounts were designed to achieve this task. However, after the discussion of solar panels on the +/- Y face came up, a redesigned rail could allow for one cohesive face to mount solar cells rather than two 3U sides plus an unwanted gap in between the two 6U, providing more power overall. This method would accomplish the task of combining the 6Us more elegantly and cohesively without leaving a gap in between.

While the main design of the +/- Y rails has to conform to the dimensions of the existing bus due to the untouched +/- X axis rails and corner rails, the design that goes in between the gap of the 2 6U has much more leeway. At first, an attempt at developing a method to compare rail designs before integrating them into the entire assembly was made based on the moment of inertia matrix on Solidworks. However, the results were determined to be inconclusive and were not pursued any further as the matrix displayed on Solidworks refers to the mass moment of inertia rather than the area moment of inertia. Instead, a design will just have to be made and analyzed within the greater assembly and iterated from there. The experimental process is documented in Appendix A.4.

To summarize the process, the idea was based on this big assumption that choosing the design with the largest frequency for the first mode would mean it is the stiffest design, and thus once integrated into the entire assembly, should provide the most increase in stiffness out of the other designs. While this is quite a big jump to be made, the alternative is testing each iteration within the entire assembly and running a modal for each, which was just not possible given the timeline of this report. Thus, rather than just arbitrarily designing a rail based on pure intuition, this method will be used as it provides some form of metric to compare between designs. And if it turns out to be arbitrary, the original method wouldn't have been much better off.

The experiment was formed based on the natural frequency equations of beams subjected to different end conditions (Figure 7). Since the top and bottom ends of the rails will be fastened down to the +/- X rails, it can be treated as a pinned-pinned beam. From this, the corresponding equation shows that the frequency has an inverse relationship with the length of the beam as well as the mass per unit length while having a proportional one with Young's modulus and the moment of inertia. Between designs, the material and length will stay the same due to the previously mentioned requirements. Thus, only the mass per unit length and moment of inertia could be changed. However, when quickly iterating between Version 1 Designs, the mass between the rails was purposely kept relatively the same (+/-2g of each other), meaning any changes to the moment of inertia matrix from Solidworks should correspond with the predicted relationship with the natural frequency (E, L and now ρ_L are constant). Unfortunately, this was not as simple as it was hypothesized, as although one design would see each component of the principle inertia matrix increase, either simultaneously or with a significant increase to one component, the natural frequency wouldn't result in the expected outcome.

However, the entire process wasn't completely useless as it was seen that reducing the gap between the CubeSat by thickening that surface was the most effective in increasing the resonance frequency of the first mode. Additionally, it was seen that the first design had the most potential in terms of increasing the stiffness as it had the highest frequency for the first mode with a value of 765.17 Hz for the first mode before the discovery of decreasing the gap size. Due to this, this design will be the main one that will be iterated upon (Figure 24).

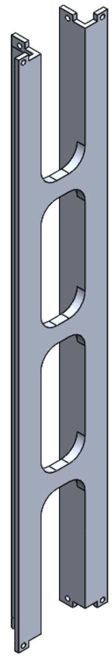


Figure 24: +/- Y Rails Ver1a

After selecting this design, only two more variations of it were analyzed, both of which featured thicker walls to close the gap as discussed, with the only other difference being the second variation had more of the pattern vertically (Figure 25).

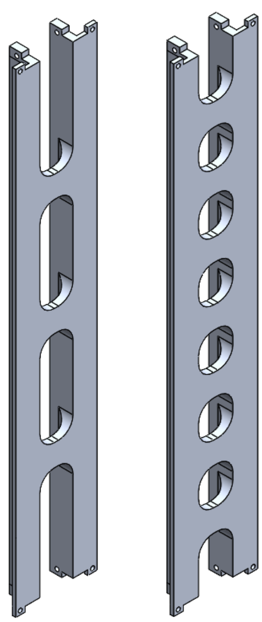


Figure 25: +/- Y Rail Ver 2a (Left) & Ver2b (Right)

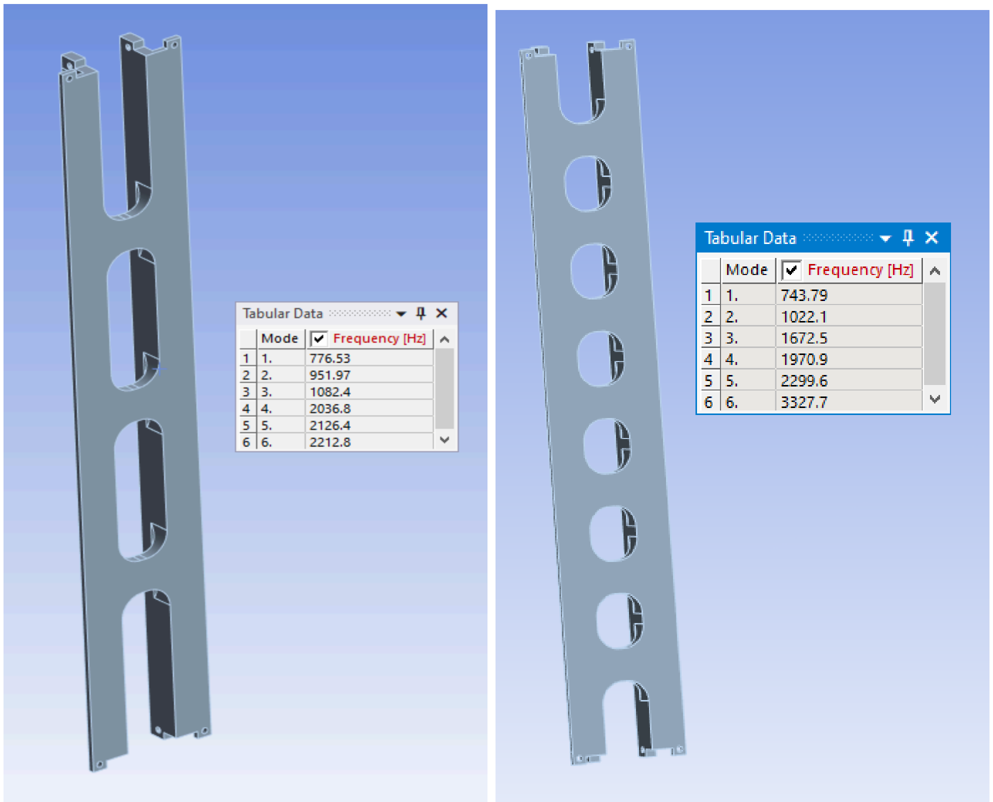


Figure 26: +/- Y Rail Ver2a Modal (Left) & Ver2b Modal (Right)

From these last few analyses, Ver2a had a slightly higher frequency at the first mode at 776.53 Hz vs Ver2b at 743.79 Hz, making Ver2a the design to progress with (Figure 26). While it seems counterintuitive that Ver2b had such a drastic drop in frequency (even lower than Ver1a), it seems like the added mass ended up overpowering the increase in the moment of inertia instead, resulting in a net decrease in the frequency. As a final lesson from all this, it must ensure that any amount of mass added to the beam is intentional in increasing the moment of inertia more than mass. This means any additional masses should be as far away from the center of mass as possible given the integral form of the moment of inertia. This also gives a possible reason why Ver2B had a lower frequency as all the extra mass increased the mass per length of the rails a lot more than it did the moment of inertia. A final design was made where additional masses were added to the ends of the rail to validate this conclusion (Figure 27):

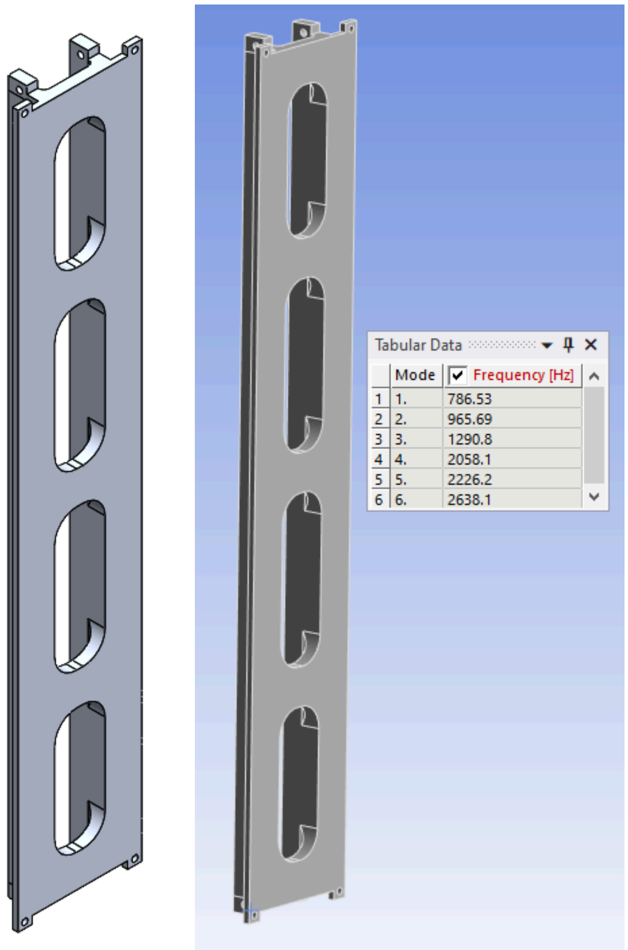


Figure 27: +/- Y Axis Rail Ver3 & Modal Analysis

This +-Y Axis Rail Ver3 achieved a frequency of 786.53 Hz for the first mode, the largest out of every iteration. Again, this value while way surpassing the 80-90 Hz minimum set at the beginning, will not directly translate to the overall structure. This metric was just used to compare between rail designs and while the highest first frequency mode between them should correlate with it being the stiffest, this cannot be validated without a full system analysis. But since this design used a combination of intuition with the placement of additional mass according to the theory, as well as achieving the largest frequency, this will be the final design for now. In the future, there should be efforts into looking to thicken the walls some more and further close the gap if the mass budget allows. However, it should be kept in mind that increasing thickness too much could have an inverse effect due to previous findings as well as not providing enough space needed to fit an appropriate torx key for fastening the bolts down. This design is far from optimized and is open to even more redesigns in the future as Cislunar becomes more defined.

4.4. +/- Z Rails

With Cislunar's demonstration of water electrolysis propulsion, the Cubesat bus provided by Spacemind doesn't accommodate a propulsion system, requiring a hole to be drilled through one of the +/-Z axis rails to allow for prop firing, making it necessary to redesign at least one of the rails. Additionally, the provided rail also provides additional features for mounting the Stack (Figure below) which will not be used, providing the middle rails additional unutilized mass.

Unlike the +/-Y Axis rails that required a design from scratch, there were original rails that can be analyzed to provide a baseline for the +/-Z axis. After splitting the faces that will be in contact with the +/- X Axis rails in order to add fixed support to them to treat the rail as a pinned-pinned beam, it was seen that any future design should attempt to at least surpass a first mode frequency of 207.52 Hz to be considered an improvement for stiffness purposes (Figure 28).

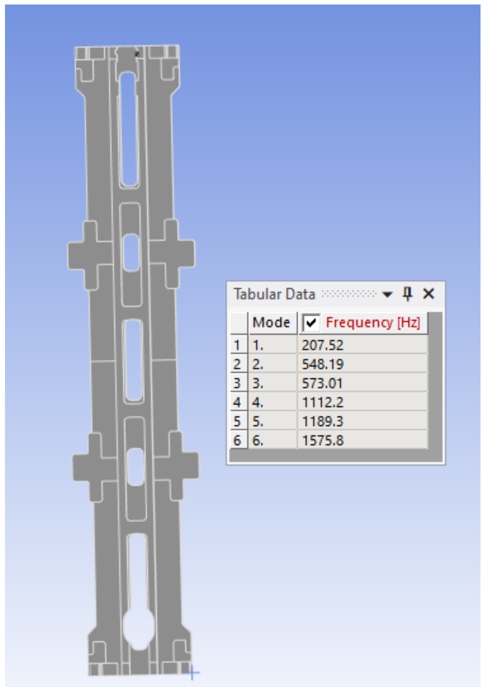


Figure 28: Spacemind +/- Z Rails

There will be two configurations of any design for the +-Z axis rails; one with a hole in the center to accommodate the propulsion nozzle and one without. Aside from the hole, both configurations will be identical, and thus, only the one with the hole will be analyzed due to the expectation that the configuration would perform worse than if it didn't have a hole in the center. The initial design started off with a fairly similar shape to the original, with the ends of the rails being the exact same in order to accommodate the +-X axis rails that will not be redesigned and a majority of the mounting features for the stacks being removed. From there, a hole was placed into the center to meet the requirement of fitting the propulsion nozzle. From the previous equation, we need to ensure the beam weighs as little as possible while also increasing the moment of inertia, which can be achieved by concentrating mass towards the ends. Following this philosophy, the following design was conceived (Figure 29):

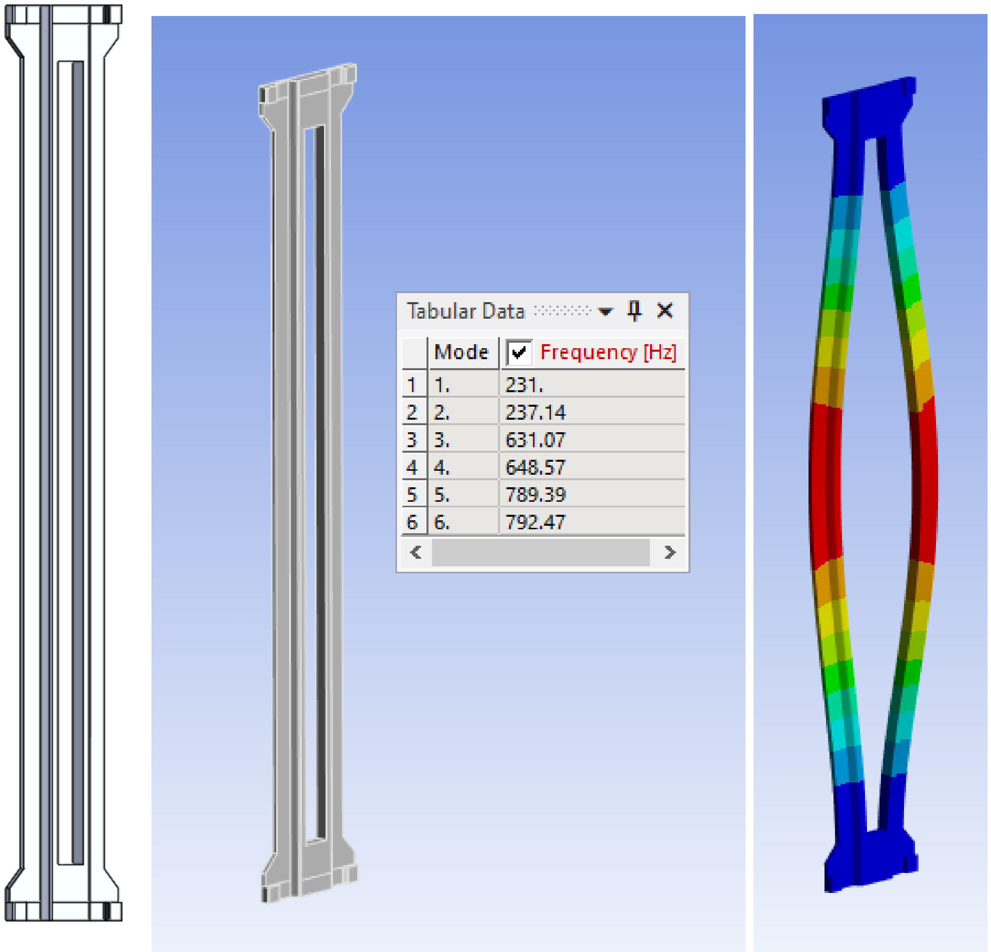


Figure 29: +/- Z Rails Ver1 (Left), Modal Analysis (Middle), Second Frequency mode (Right)
NOTE: This design did not feature a hole for the propulsion nozzle as the opening is more than enough to allow for one.

Compared to the +/-Y Axis, this initial design is off to a great start, with a 23.48 Hz increase from the original rails provided (231 Hz). It seems like the assumption that focusing on increasing the moment of inertia along the long axis (varies depending on the plane of the sketch) while minimizing the mass added to the other axis is correct. This provided higher natural frequency since the placement of the mass should increase the moment of inertia more than the amount of mass added. But while removing the entire middle section and dedicating the removed masses to the ends successfully reproduced a higher natural frequency that matched intuition and theory from Figure 7, the resulting first and second frequency modes are extremely close together, with the second frequency mode showing bendings of the two sections in opposite directions. This isn't super ideal since it means there is a greater chance of hitting a resonance frequency within the span of only 7Hz (Figure 29).

Thus, in an effort to further separate the two modes, some added supports were employed in the center of the rail, forcing the addition of a circular hole to still accommodate the nozzle.

Additionally, filets were added to the design which is for stress concentration purposes (Figure 30).

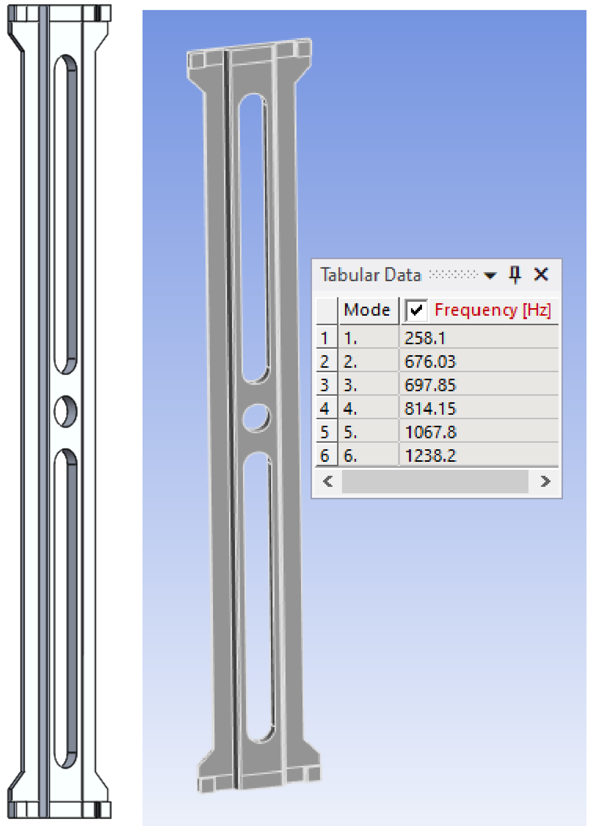


Figure 30: +/- Z Rails Ver2 (Left), Modal Analysis (Middle)

Surprisingly, there has been another large increase in the natural frequency up to 258.1 Hz, which might indicate filets help with vibrational purposes as the addition of the support in the center shouldn't have increased the moment of inertia by much. The filet sort of makes sense as reducing the amount of stress near the end conditions would result in higher vibrations to cause the same amount of deformation/vibration behavior. Additionally, the second mode was successfully pushed up by ~400Hz but now is close to the 3rd mode (676Hz vs 697Hz). While this isn't a large concern due to how big the frequency is, if time permits some work will be done at the end to try to remedy this. But this isn't a priority like it was with the first version as 200Hz is much more likely than 600Hz.

Due to how much of an improvement was seen, this will be the final version for now, with efforts focused on getting an overall ANSYS model completed to validate these choices and a final decision made on these rails for the semester.

5. ANSYS Analysis

Following the conclusions of both sections on Simplified Systems, as well as Shear Panel/Rail Redesigns, a final assembly was made based on previously mentioned required placements of certain components, where the rest were placed to best balance the mass (Figure 31):

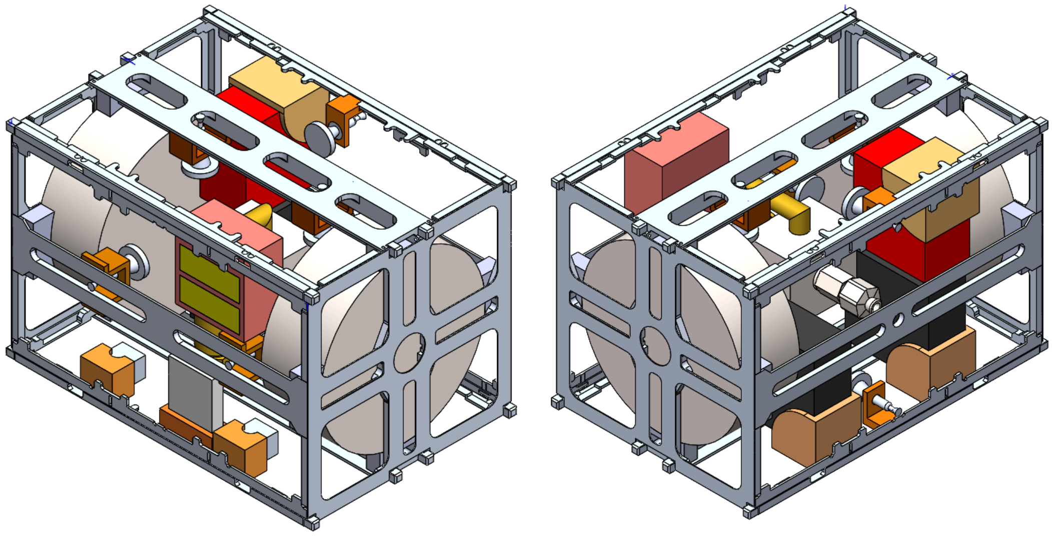


Figure 31: Fully Assembled 12U w/ Simplified Systems

Each item was assigned material from the following list according to the specific ones mentioned in the Simplified Subsystems section, where all the mounts are assigned ABS (Figure 32).

Aluminum alloy, wrought, 6061, T6
Brass, C37700
Titanium alloy, Ti-6Al-4V, annealed
Plastic, ABS (high-impact)
PCB laminate, Epoxy/Glass fiber, FR-4
316 Stainless Steel

Figure 32: Material Selection for ANSYS model

NOTE: All mounts that were not discussed previously were arbitrarily made from ABS (except for the prop tanks which were aluminum) and mounted as it is assumed the majority of them will be 3-d printed to fit the necessary angles/surfaces of the pre-existing corner rails. Not enough time remained to properly think about the mounting of each component due to the only viable options being mounting onto the rails. This indicates the need for some sort of brace that runs throughout the structure both for structural support and as a form of mounting.

5.1. ANSYS Model: Fixed-Free analysis

Recall from the Stiffness section that while the analysis we would like to employ is a free-free analysis, the best we can do for now is a fixed-free one. Thus, the entire bottom face of the structure was applied a fixed constraint, with every single body in this model being assigned a “Body Sizing” of element size 1e-2m (Figure 33).

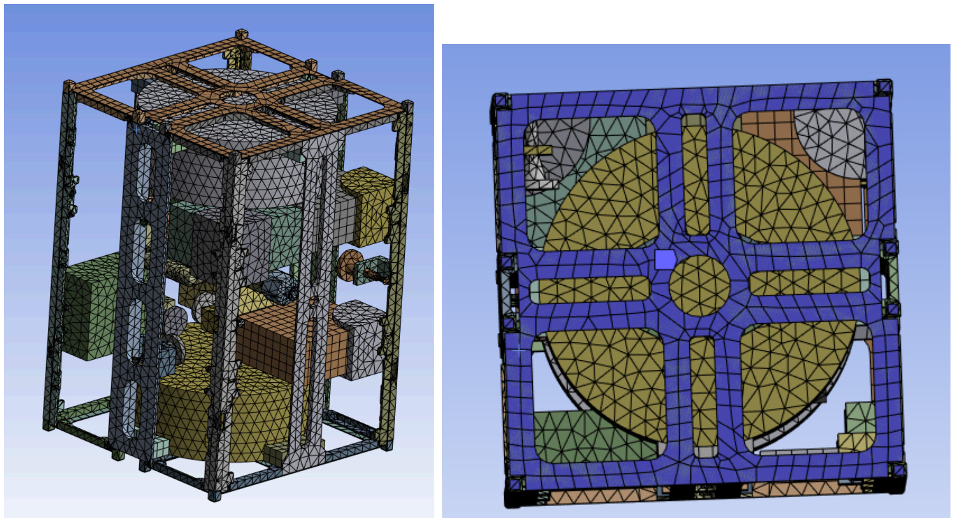


Figure 33: Mesh Sizing of 1e-2m & Fixed Constraints for Modal

From solving the Modal Analysis, it can be seen that the first 6 frequency modes ended up having a frequency of less than 90Hz, with the 6th barely being over (Figure 34):

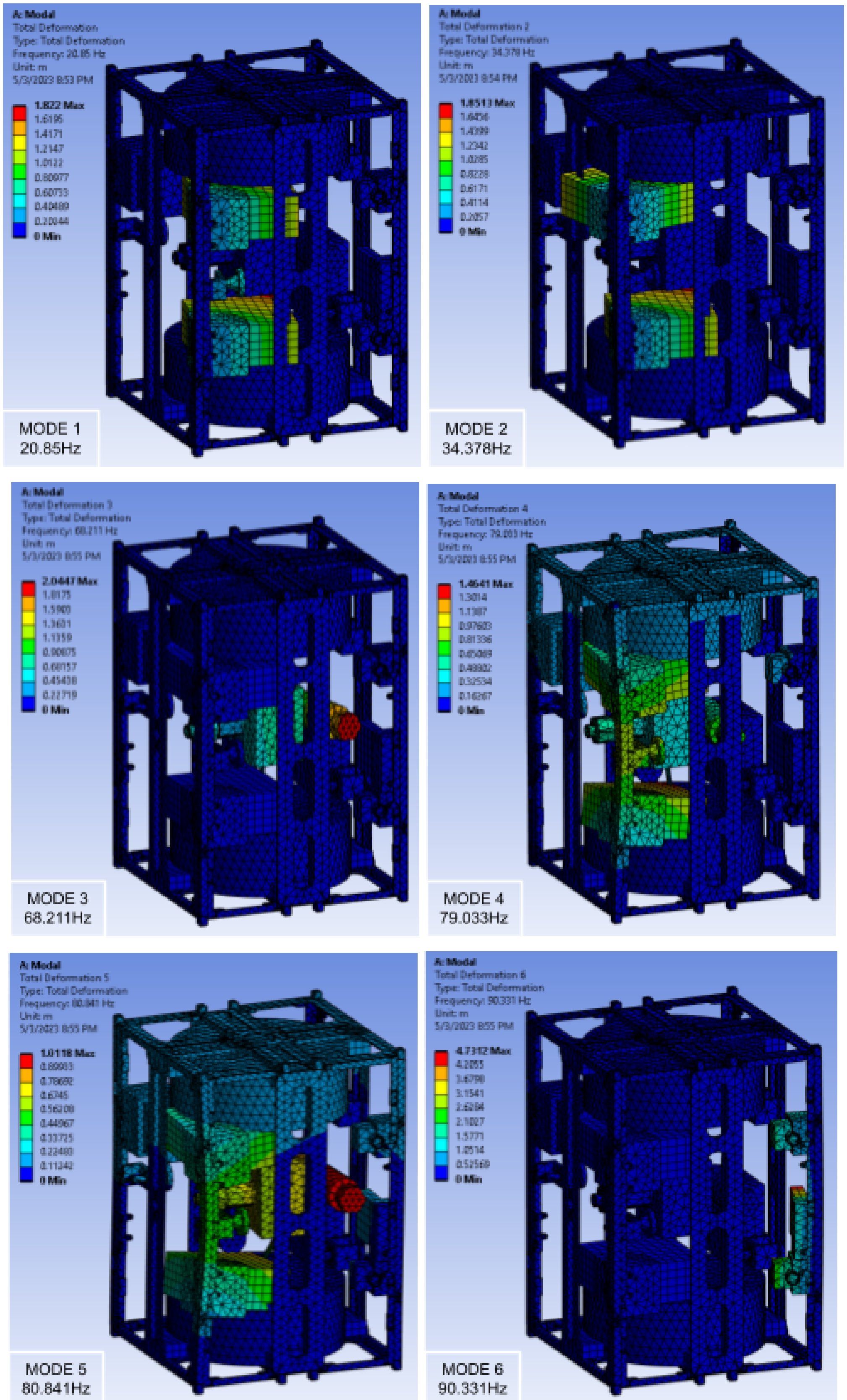


Figure 34: Frequency Modes 1-6

While the first 20 modes were solved, only the first 6 are focused on since they did not meet the requirement of having a frequency of higher than 80-90Hz. From these analyses, it doesn't seem like the rails are what's causing the structure to fail this requirement, but rather all the mounted components from other subsystems. Modes 1,2,3 and 6 seem to be purely an issue of mounting as only subsystem components are showing deformation while the rest of the rail structures show little to none. There was a large oversight in mounting the majority of the sensors/mounts in a cantilevered form rather than attempting to fix as much of its surface to something else as possible, which makes sense that it resulted in the system having frequency modes much lower than expected. Referring back to both Figure 7 as well as Eq.4, a fixed-fixed condition when compared to a cantilevered one sees almost a 7x increase in the natural frequency of the beam. Thus, by cantilevering the majority of the components (fixed on one end and free on the other), the natural frequency of them is decreased significantly. In the future, more work has to be done to ensure that components are mounted as flat against something as possible and that if they have portions that are free hanging, they should be properly supported.

Aside from the modes induced by poor mounting conditions, modes 4 and 5 also raised concerns as these modes show deformation of the rails as well. It is harder to tell whether it's the components reaching resonance or the structure itself, or a combination of both. But regardless, this modal analysis points to the larger of the system being undersupported throughout. On the original Spacemind CAD, the 6U had "stacks" throughout the bus (Figure 35). It was first thought that these stacks were used purely for the division of volume within the 6Us. But upon further inspection along with the modal analysis, it seems that they help maintain the structural integrity a lot more than expected, which is what allowed Spacemind to get away with having such a lightweight bus in the first place.

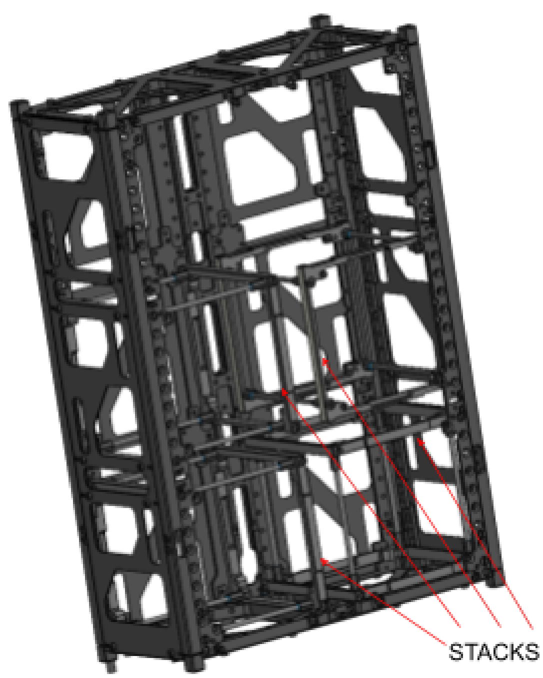


Figure 35: Spacemind Stacks

Unfortunately by combining two individual 6Us as well as featuring large propulsion tanks, this means that we cannot reuse the stacks seamlessly and would require some level of redesign. Given that the stacks are extremely rigid in the way they divide up the volume in the center of the CubeSat, which is essential to allow the fitment of all necessary subsystems, a bulkhead design is proposed instead (Figure 36).

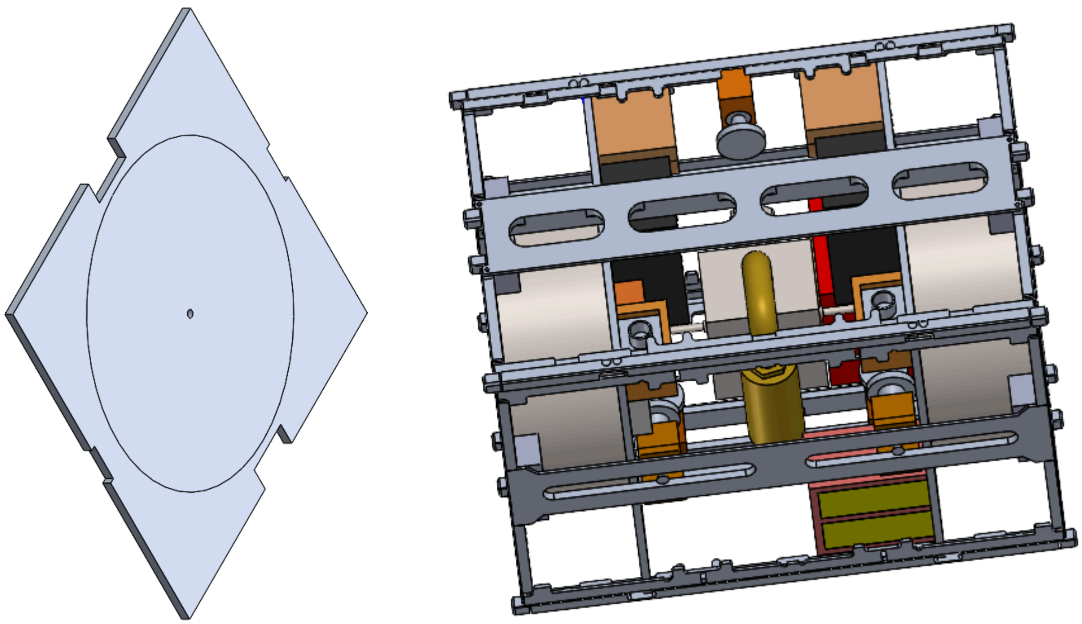


Figure 36: Bulkhead Concept (Left) and Assembly (Right)

This concept was integrated into the overall assembly, where the subsystems are no longer mounted onto the corner rails (except for thrusters) but now onto the bulkhead directly. The assembly seemingly has all the mounts for each subsystem bonded to the bulkhead, which shouldn't be too far off as it is expected for each mount to have some sort of bracket connecting the mount to the bulkhead and constraining it; they are just not yet designed. The flame arrestor was also updated as the previous one had extremely sharp features. The assembly seen in Figure 36 will be the one used for the final modal analysis (Figure 37).

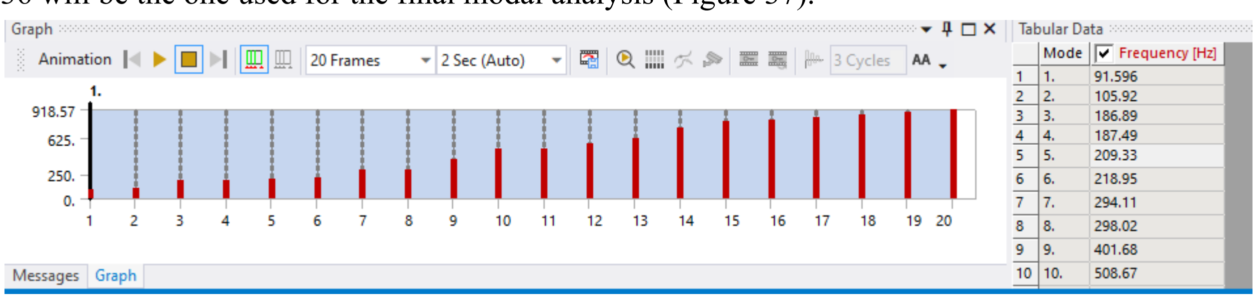


Figure 37: Modal Analysis with Bulkhead concept results

It seems like the bulkhead concept worked, as the first frequency mode is above 90 Hz and is caused by the flame arrestor being cantilevered, which can easily be pushed further by adding supports to the flame arrestor such that one end isn't floating. However, there was a concerning warning message (Figure 38).

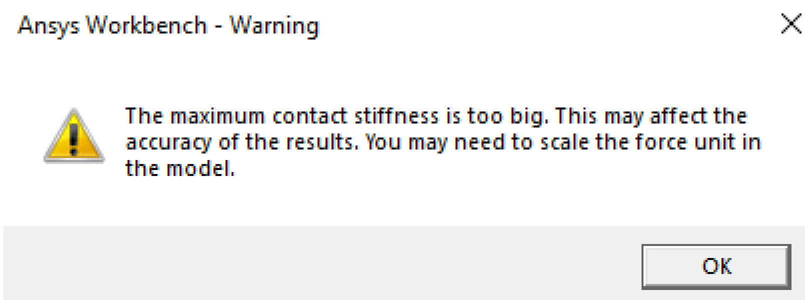


Figure 38: Contact Stiffness Warning Message

This contact stiffness warning most likely came from using the auto-defined contact regions between components, which unfortunately doesn't take into account what specific part of the face is one component in contact with. Instead, this assumes the component is in contact with the entire surface of for example the bulkhead it is mounted to which is not physically possible. One solution would be to go into Spaceclaim on ANSYS and manually split each individual contact surface into the faces that are actually in contact with the component, which is extremely time-consuming and tedious. Instead, "Share Topology" can be used which essentially assumes only the portion of the faces touching another component is bonded. This method wasn't used at first since there were issues with getting a proper mesh working. While the exact details of what goes on once this is toggled are unknown, it splits the faces where needed, for example, the face of the bulkhead is now split into individual faces where the components are touching it (Figure 39):

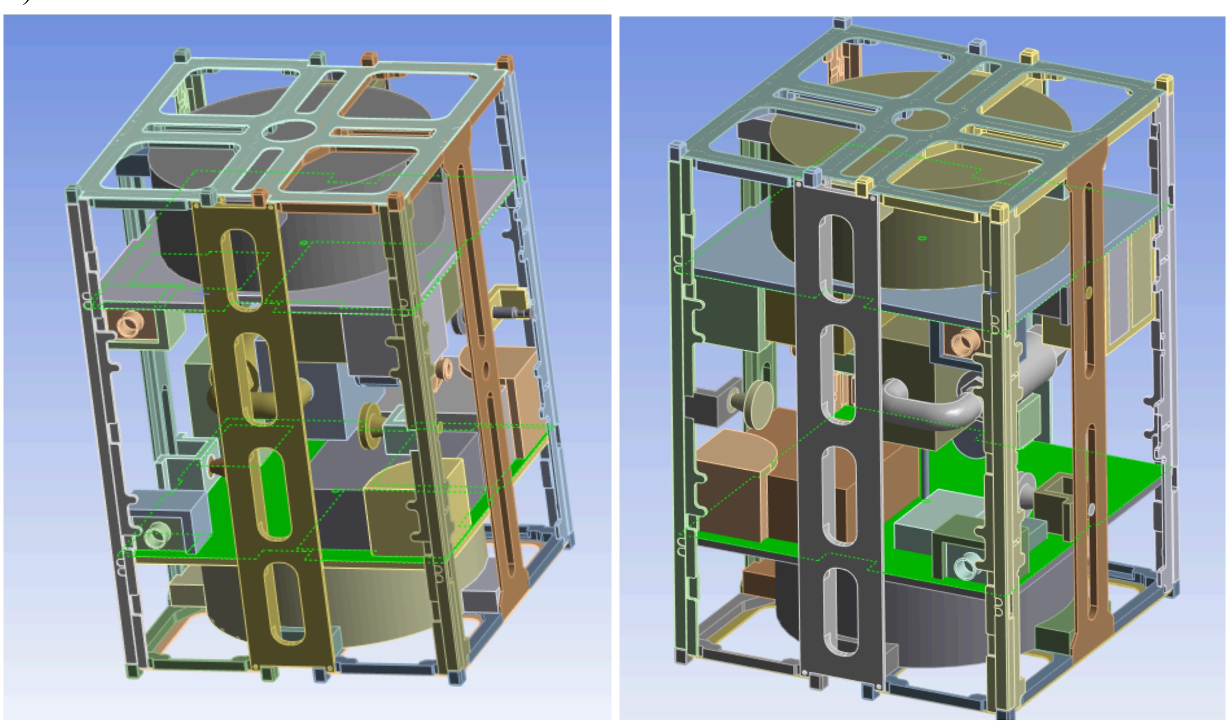
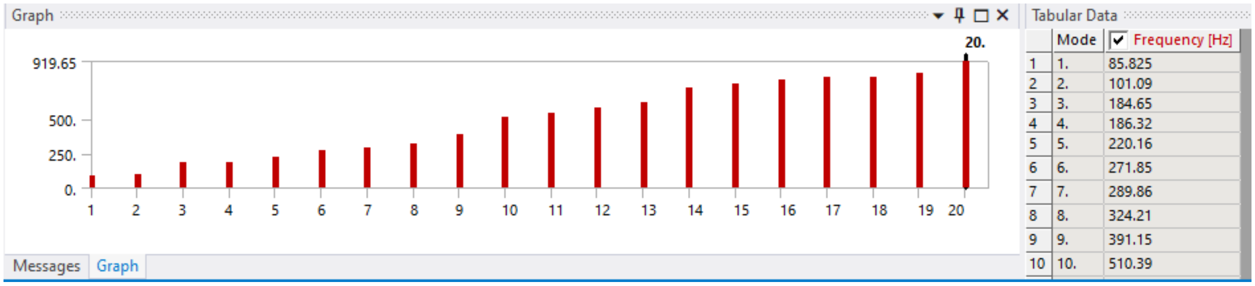


Figure 39: System with Share Topology (Left) and without (Right)



	Mode	Frequency [Hz]
1	1.	85.825
2	2.	101.09
3	3.	184.65
4	4.	186.32
5	5.	220.16
6	6.	271.85
7	7.	289.86
8	8.	324.21
9	9.	391.15
10	10.	510.39

Figure 40: Modal Analysis with Bulkhead Concept (Shared Topology)

This time around, the shared topology was able to result in a proper mesh but it isn't known how. At first, the mesh still failed and after running it a couple of times without really changing any settings, it resulted in a working mesh. Every component except the +/- X shear panels had a body sizing of 1e-02m while the +/-X panels had 5e-3m. This is due to a warning of "One body has been found to have only 1 element in at least 2 directions" about the panels, which can be solved by decreasing the mesh size. Share Topology should be looked into in the future to try to reproduce the same results as this method does save a lot more time than the alternative (unless someone is feeling extra motivated to split every single face to compare results). The analysis shows that the first mode is slightly under the ~90Hz requirement but inspecting it closer, the first two modes are only due to the flame arrestor (Figure 41):

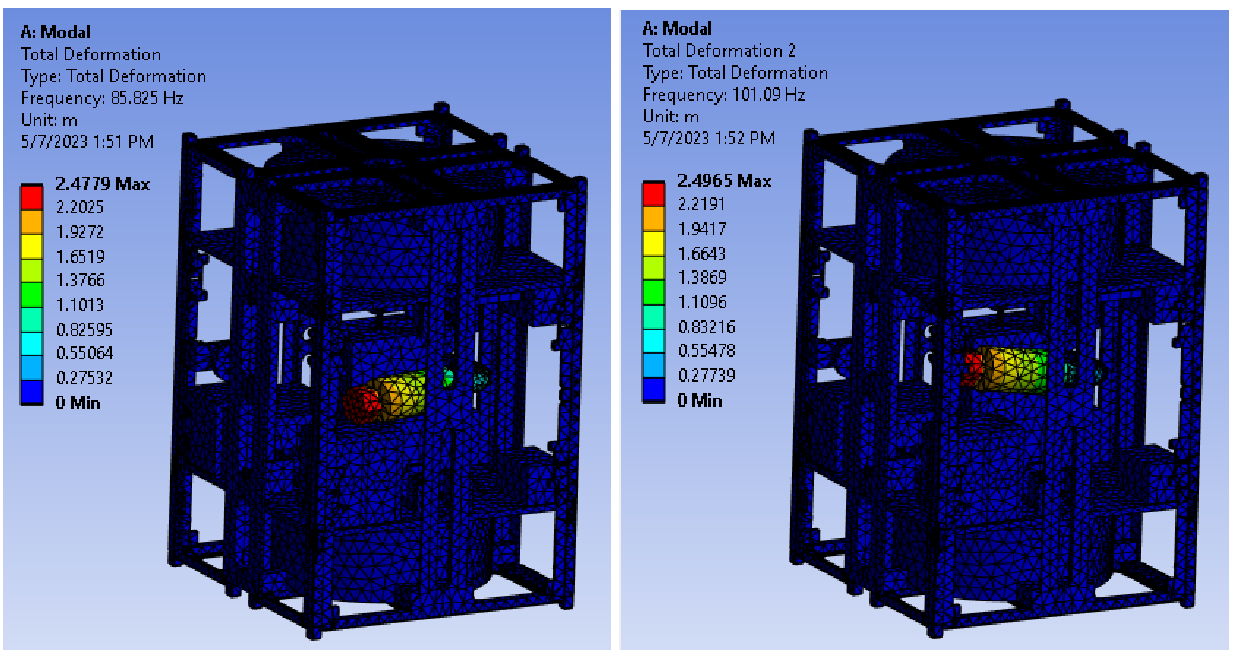


Figure 41: Frequency Modes 1 and 2 of Bulkhead Assembly

Once again, this indicates that the bulkhead method will be necessary moving forward as both a way to add stiffness to the overall structure as well as allowing for better mounting of components such that those do not have frequency modes close to the launch environment. Although the assembly still didn't technically meet the frequency requirement, it can easily be remedied by coming up with proper supports for the flame arrestor.

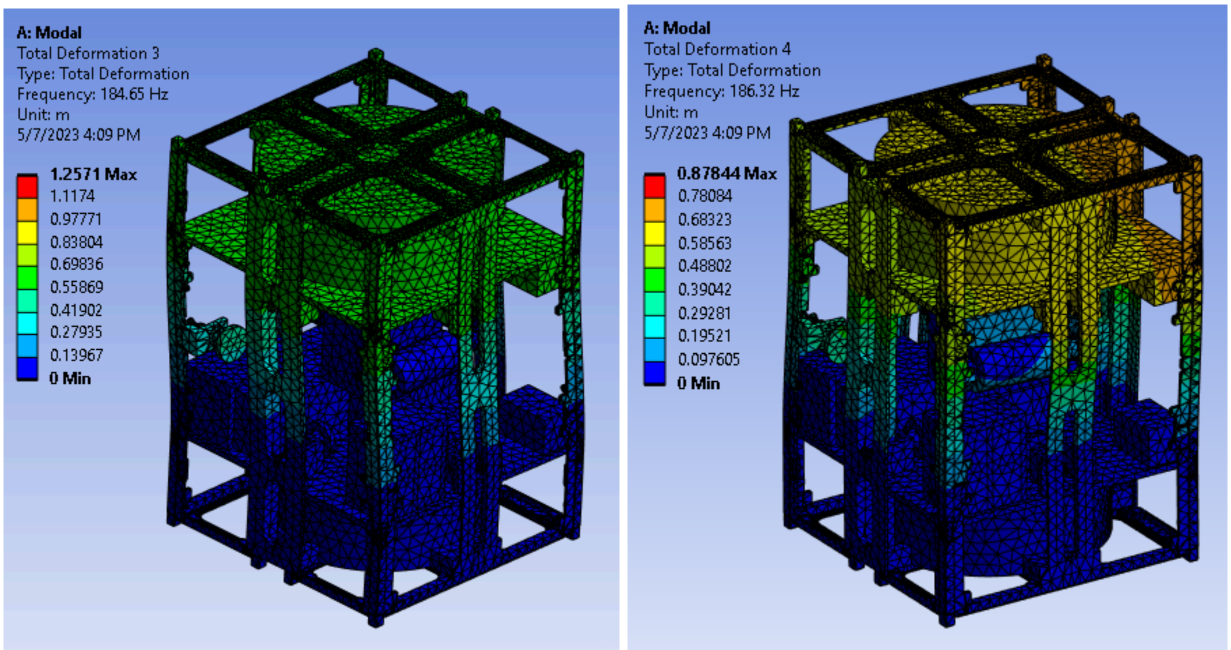


Figure 42: Frequency Modes 3 and 4 of Bulkhead Assembly

While modes 3 and 4 are well above the minimum frequency needed, methods to further increase the stiffness according to these will be discussed (Figure 42). These two modes are unique as they aren't caused by a single glaring issue such as the thrusters hitting resonance or the propulsion assembly in the center hitting resonance which have relatively simple solutions. These two see the vibrations of the entire top portion of the structure, which is closer to what was expected as this reflects the first mode of a cantilever beam (the end faces the most displacement). While the following design wasn't implemented as it isn't necessary to increase the stiffness any further, additional supports in the form of poles/columns can run throughout the bus to provide additional support. For example, four poles running through the length of the CubeSat at each corner can help resist this twisting motion and thus make the entire structure stiffer. But more effort should be put into mounting the thrusters and combustion chamber/flame arrestor first before tackling this.

5.2. ANSYS Model: Acceleration/ Buckling

Although the report went into calculating two potential load cases that could be used for Acceleration and Buckling, they aren't as much of a design driver as the stiffness of the structure. A preliminary acceleration load case was done but it didn't provide much useful information as the majority of the structure saw little to no deformation. However, this analysis should still be done at the very end when all subsystems are completed before verification and testing to ensure that some form of mounting for electronics or thrusters doesn't snap off due to the acceleration/applied load faced. But structurally, the bus should be fine and not yield under these cases.

For future reference, I anticipate three different acceleration cases to have to be done:

Case 1: Axial Acceleration (22.6g)

- Fixed on + or - X face (direction of acceleration has to change accordingly), frictionless on sides, this simulates the Cubesat being pushed against the bottom of the dispenser while not being constrained on the other surfaces

Case 2/3: Lateral acceleration on +/-Y and +/-Z (16.1g)

- Fixed on the one side (either Y or Z), frictionless on the other side. This simulates the Cubesat being pushed against one of the sides of the dispenser but can still slide along the slides of the dispenser.
- Repeat with the other side

As for Buckling, this was the load case with the least amount of work and research done and thus should be re-verified. It is likely the calculations done would have to be rechecked in the future once a dispenser is chosen but also probably not the most necessary and would be extra. But with there being a lot more structural redesign necessary, this could potentially be a concern with the majority of the weights being dedicated towards the ends of the CubeSsat. If one other loading has to be looked into, I recommend it to be stability/buckling with acceleration being the last focus.

6. Future work for I&T

There needs to be extensive work in updating the simplified models of each subsystem, as well as the overall assembly, started in this report to ensure they are all up to date. At the time, nothing was finalized in terms of components so it is highly unlikely that the orientation of the components in this report will reflect the final. As components start to be decided, mounts also have to be considered and designed according to the findings of this report (i.e. avoid cantilevering components). Almost every mount made was from ABS and assumed to be 3-d printed (except propulsion tank mounts) to avoid spending too much time designing realistic/optimized mounts so there is lots of work to be done here in the future.

Additionally, according to ACS, the thrusters need to be placed as far from the center as possible in order to get the most out of them for maneuvering and reduce propellant costs per firing. This was not possible on both the Z and Y-facing thrusters due to the propulsion tanks being placed on the ends. If it is still necessary and the team cannot afford the extra costs from the inability to increase the moment arms of these thrusters, an idea would be to design some sort of stands/holders to elevate the propulsion tanks to fit the thrusters underneath. This was explored but then quickly abandoned due to not being able to raise them high enough to also fit other components underneath the propulsion tanks. If the team really focuses on sizing components down, especially the battery and spinning sun sensor, this idea can be re-explored in the future (pending minimal interference with the water's ability to be in contact with the electrolyzers by moving the propulsion tanks closer to the center).

Mounting holes need to also be kept in mind and placed onto the shear panels/rails as these were not modeled to provide more flexibility to decide how to mount components. For example, the mounting holes of the rails/panels can ensure the solar panels are mounted such that they don't interfere with the solar cells' placement. This can be done later down the line but some holes will have fixed placements (such as mounting them onto the existing rails by Spacemind). Additionally, the rails should be optimized and changed accordingly to fit whatever the latest structural redesign requires. These were designed without the idea of adding a bulkhead and prior to having a complete list of items that will be on the Cubesat, making it essential to constantly revise these as more information is obtained. It could very well be that they are overdesigned (which I speculate is the case) given how thin the original rails were and the original impression that the rails will be what provides the majority of the stiffness, which shouldn't be the case if more of the internal structure is designed properly.

The bulkhead design presented in this report is extremely simplistic and is only meant as a concept for future members to build off of. As there isn't enough time left to properly optimize it towards the Cubesat, one of the big immediate tasks would be to design a proper bulkhead with some more formal engineering processes. Some design pointers would be to look into doing a geometric pattern that both reduces mass, increases strength, and allows for wires to cross the bulkhead if necessary. An example of this is provided by the Violet Satellite Project, which features an iso-grid design for a wall panel (Figure 42) [8].

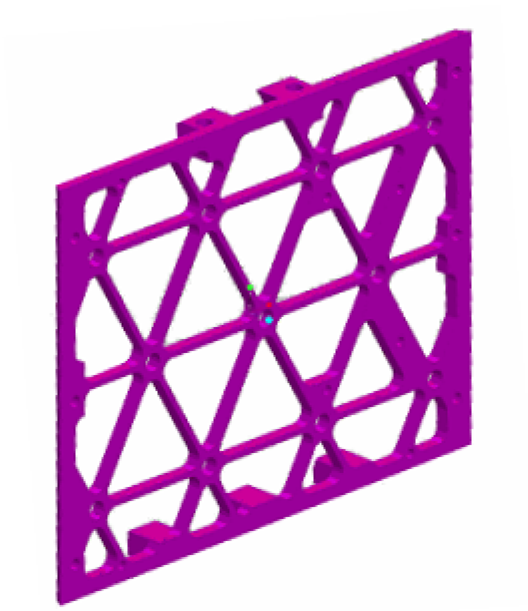


Figure 42: Cornell University Violet Satellite Project Iso-grid Panel

I suspect a large bulk of the time will be towards optimization since the current orientation of the bulkheads, where they sit on top of the propulsion tank, reduces the amount of already limited volume in the center towards other subsystems. There will be integration issues as many of the components will run into the combustion tank once the propulsion tanks have been updated with the electrolyzers. I also predict that as the bulkhead starts to get thinner in an effort to find the minimum thickness that can still meet all necessary requirements, there will need to be additional poles/columns to help further support and increase the stiffness of the overall structure (unless stronger material than Aluminum is considered). The Violet satellite didn't have these because they are a 1U CubeSat and are naturally stiff, to begin with (natural frequency is inversely proportional to length). The bulkhead placements should be experimented with as they don't have to sit horizontally and parallel with the propulsion tanks, that is just what was easiest to integrate into the assembly at the time. It could be that the solution would be to have one large stack assembly that goes in the center of the CubeSat which would help stiffen the structure and another independent mounting structure could be designed such that components wouldn't have to be cantilevered. Or if the stack idea has some glaring integration issues, four poles/columns can be placed at each of the four corners that run along the length of the CubeSat. Either way, there is lots of work to be done structurally with this aside from the rails, which should be kept in mind and updated to fit whatever the latest design choices are.

Lastly, more work should be done to look into the Spacemind 6U structures and analyze what they have done and reverse engineer each major design choice. This should've been done as part of the Background and Theory work and is a big oversight on my part considering the discovery of the structural purpose of the stacks only after a full assembly analysis. While there shouldn't be any more big gaps in understanding the Spacemind design, it would be helpful for future structural members to gain both a better understanding as well as intuition behind why certain choices were made and how to replicate them in our 12U structure.

7. Conclusion:

While it's hard to say conclusively that the rail redesign was a success without a final analysis further down the road, they should have met the requirements mentioned in this report. The safety factor mentioned was met if the frequency requirement was met, which makes it somewhat redundant but mentioned for completion. The rails were stiff enough if made from aluminum 6061, which is recommended by the CDS for the main structure as the overall assembly surpassed the ~90Hz requirement (not including the first frequency mode as that is due to the flame arrestor). They should each have enough openings to fit the required nozzles/thrusters on the respective faces that need them. Lastly, each surface retains the mounting holes of the Spacemind rails, with the redesigned Y and Z rails allowing for more mounting options for the solar panels if needed. If for whatever reason these designs are not chosen to be moved forward with, much of the work done in this report will provide a foundation for the new designs so regardless, the work done this semester will be helpful to Cislunar Explorers moving forward.

8. Reflection:

My senior design project evolved over the course of this semester. I initially started off wanting to explore my passion for GNC with the ACS subteam, but it was clear that my lack of knowledge in the field would make it hard to have a successful senior design report. It was thanks to the Chief Engineer, Emily Matteson, who first raised this concern to me, that I switched over to the topic of redesigning the shear panels. At this point, I had missed out on more than a month of work and would've missed out on more had I not switched topics when I did. There was definitely an initial struggle of quickly getting back on track with having to spend weeks on preliminary literary reviews before I could start any concrete designs. It was about this time when I designed the first iterations of each shear panel that the idea of redesigning rails came up. I had offered to take on the task since the shear panels interact directly with the rails and rather than having to hear about changes being done to the rails that would force the panels to change, I decided to design them in tandem with one another. It also made sense to do this as the rails would provide more stiffness than the panels so the redesign may end up making the panels obsolete. This put me on track with my finalized topic of "Structural Redesign and Analysis of 12U Bus". This topic is a lot more encompassing and realistically, my scope was not limited to just the rails. I ended up suggesting a preliminary bulkhead design along with many other mini suggestions that could be pursued in the future which came from the modal analysis I had done.

Regrettably, I didn't get to do as much optimization work as I had wanted with both the rails and bulkhead. I had tried to create many of the foundations for what I hoped to be many future structural analyses. This included recreating each simplified rail by copying the dimensions one by one as well as each subsystem to the best of my ability according to datasheets which was much more time-consuming than I had originally expected. There were also multiple issues that came up during the semester, from the propulsion tank sizing to placements of the ACS thrusters, that I had to work through and just make a decision to move forward with.

I intended for my work and assumptions to be detailed and documented in a way such that they can be easily understood, replicated, and built off of. Despite Cislunar no longer being a project in the future, I hope that my work here can still be used as a reference if the 12U bus by Spacemind is used for a future flight experiment by SSDS. Or more generally, as a good background to have before getting into structural designs of any CubeSat design.

9. Acknowledgements

I would like to thank the following people for their help with my report this semester:

- Dr. Mason Peck for meeting with me throughout the semester and providing me with pointers at integral points of this semester. This report would not have been what it is without his review of my work which led me to focus on modal analysis, as well as the addition of the bulkhead.

- Professor Mark Campbell for his help with providing the source on the natural frequency of a beam under different boundary conditions, which served as the basis for many of my assumptions. Also for answering any quick questions I had about my project after classes.
- Emily Matteson for putting me on track to work on this senior design topic and leading the team as Chief Engineer
- Nidhi Sonwalkar and the rest of the I&T subteam for working alongside me throughout the semester.

10. Sources:

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[4] H.R.Chiranjeeve, K.Kalaichelvan, A.Rajadurai "Design and vibration analysis of a 2U-CUBESAT structure using AA-6061 ..." : https://www.iosrjournals.org/iosr-jmce/papers/NCCAMABS/Volume-1/9.pdf
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[8]Kevin Meissner, Cornell University University Nanosat-6 Program "Structural Design", https://docs.google.com/document/d/0B_PXgnSe4vMRYU1lVFp0OEYyX2c/edit?resourcekey=0-P7MW-p-9K6L3C3t3ENAZSw

11. APPENDIX:

A.1: Launch Vehicles Payload User guide

Northrop Grumman Minotaur 1 Payload User guide

https://www.northropgrumman.com/wp-content/uploads/Minotuar-User-Guide-1.pdf

Rocketlab Electron Payload User Guide

https://www.rocketlabusa.com/assets/Uploads/Rocket-Lab-Launch-Payload-Users-Guide-6.5.pdf
Relativity space Terran 1 Payload User guide
https://static1.squarespace.com/static/59a8fb50d2b8575fad311abb/t/5f68d35cda98575604906432/1600705386517/Relativity+Terran+1+Payload+User%27s+Guide_Rev2.0.pdf
Firefly Aerospace Alpha Payload User guide
https://www.mach5lowdown.com/wp-content/uploads/PUG/Firefly-Alpha-Payload-Users-Guide-20160609093040.pdf
SpaceX Falcon 9 Payload User Guide
https://www.spacex.com/media/falcon-users-guide-2021-09.pdf

A.2: Proptank MATLAB:

```
%Constants
%DONT TOUCH THESE
%material properties from GrantaEduPack
%Ti-6Al-4V
yield_ti= 786.002 *10^6; %Pa, converteed from 114 ksi
%Alumunium 6061 T6
yield_al=240 *10^6; %Pa, converted from 34.8 ksi

density_water= 1000; %kg/m^3

max_pressure=344738; %Pa, converted from 50psi, max pressure in tank
```

Figure A.2.1: Proptank MATLAB Constants

This section is pretty straight forward as they are all just defining constants. The yield values of both the aluminum and titanium (the two main materials being considered) are found on GrantaEduPack and had to be converted from units of KSI to Pa to work in SI units throughout the code. The density of water is temperature dependent but it will be assumed to be 1000 kg/m^3. If there is a better value for the density of water at ground level, this can be replaced but it shouldn't vary too much. The max pressure faced in the propellant tanks was provided by the current propulsion lead, Tyler Walker.

```
%Variables
%CHANGE THESE
fos=1.5; %factor of safety for tank
volume_fos=1.08; %need 8 percent more volume (8% needs to be empty)

radius_inside= [45 60 70 80 90 100]*10^-3; % m, tests out different radius

mass_water=1.5; %kg, for one tank
```

Figure A.2.2: Proptank MATLAB Variables

There are only four variables in this code: factor of safety (F.O.S.) for stress, the F.O.S. for volume, also known as ullage, the different radiuses, and the mass of the water. The F.O.S. for stress was chosen pretty arbitrarily but can be higher (it's seen later that the thickness for Titanium is extremely low so the F.O.S. can be ramped up significantly). The ullage is just given by the I&T lead, Nidhi Sonwalker. The radius is given in a vector form in order to test out multiple options for a given mass and see which mass returns a column of results that would be a viable geometry. The mass is only the mass for one tank, so doubling this gives the actual amount of water. In the future when the mass will start to be optimized, there can be a while loop integrated into this code that iterates through a step size for the mass of the water and see how high it can go while still providing a viable geometry that will fit within the CubeSat, but this largely depends on the state of other subsystems and whether they have been finalized.

```
%Calculations, don't touch these formulas unless theres a mistake

max_stress_ti=yield_ti/fos; %Pa, sets the max stress for both material when calculating thickness
max_stress_al=yield_al/fos; %Pa, allows us to find max geometric properties rather than stress
volume_water= (mass_water/density_water)*volume_fos; %volume of water, m^3

height=(volume_water)./(pi*radius_inside.^2); %calculated from volume equation for a cylinder V=pi*r^2*h


%thin shell assumption: stress=p*r/t

thickness_ti_thin=(max_pressure*radius_inside/max_stress_ti); %thickness for ti

thickness_al_thin=(max_pressure*radius_inside/max_stress_al); %thickness for al

%thick cylinder : stress= P(r_o^2+r_i^2)/(r_o^2-r_i^2), solve for r_o
%greatest principle stress in theta direction at the interior
%https://courses.washington.edu/me354a/Thick%20Walled%20Cylinders.pdf

% outer radius for titanium (m)
r_o_ti=sqrt((radius_inside.^2*(1+max_stress_ti/max_pressure))/(max_stress_ti/max_pressure-1));

thickness_ti_thick=(r_o_ti-radius_inside);

% outer radius for aluminum (m)
r_o_al=sqrt((radius_inside.^2*(1+max_stress_al/max_pressure))/(max_stress_al/max_pressure-1));

thickness_al_thick=(r_o_al-radius_inside);
```

Figure A.2.3: Proptank MATLAB Calculations

The max stress for both materials was calculated according to the yield stress of the respective material and factor of safety (Eq.A.2.1). This will set the max stress that can be experienced at any point of the propellant tanks to meet the desired factor of safety.

$$F.O.S = \frac{Material\,Yield\,Stress}{Actual\,Stress} \qquad \text{(Eq.A.2.1)}$$

The volume of water is simply found by taking the definition of density and solving for volume and multiplying it by the extra percentage of ullage needed (Eq.A.2.2):

$$Density = \frac{Mass}{Volume} \qquad \text{(Eq.A.2.2)}$$

The height of the tank (prior to accounting for thickness) is found by using the volume equation for a cylinder (Eq.A.2.3):

$$Volume_{Cylinder} = \pi r^2 h \qquad \text{(Eq.A.2.3)}$$

Under a thin shell assumption, the thickness of a cylindrical pressure vessel is pretty straightforward to calculate if the pressure inside and radius is known. Since the max stress happens along the hoop stress direction and so knowing the max stress, pressure, and radius, a thickness can be calculated (Eq.A.2.4):

$$\sigma_{hoop} = \frac{P_{max} r}{t} \quad \text{(Eq.A.2.4)}$$

The thick pressure vessel assumption was also provided, where the outer radius can be found assuming no external pressure from the following equation (where max stress occurs at the inside radius) (Eq.A.2.5). The thickness then can be found by finding the difference between the outer and inner radiuses.

$$\sigma_{hoop} = P_i [\frac{r_o^2 + r_i^2}{r_0^2 - r_i^2}] \quad \text{(Eq.A.2.5)}$$

```
%results
% each row is: radius, height, thickness ti (thin), thickness ti
%(thick), thickness al (thin), thickness al (thick)
%each column is the corresponding values for each radius at a given mass

final_output=[radius_inside; height; thickness_ti_thin;thickness_ti_thick;...
    thickness_al_thin;thickness_al_thick]*1000 %multiply by 1000 to get mm
```

Figure A.2.4: Proptank MATLAB results

This last section just tabulates the results all in one matrix so they can be read off of easier.

A.3: Simplified Systems Data Sheets

Mass budget (12U)

https://cornell.app.box.com/file/922390737139?s=if58xxn3c18e63xycuk3g8qgu2i7x58j

GOMSpace P8 data sheet

https://gomspace.com/UserFiles/Subsystems/datasheet/gs-ds-nanopower-bp8pdf_(DS_-_1034928_-_1_-_2)_-_1-(1).PDF

GPS Sensor

https://cornell.app.box.com/file/1173903960616?s=t9zleizna3oz3uwfn0hk4uf8ev1kna0i

Steel tube data sheets

https://cornell.app.box.com/file/814280828839

Flame arrestor data sheets

https://cornell.app.box.com/file/814273677439?s=44i14g2rkvyemllx1nmz7mo2zg1ggg97

RedWire Spinning Sun sensor

https://cornell.app.box.com/file/1160682988853

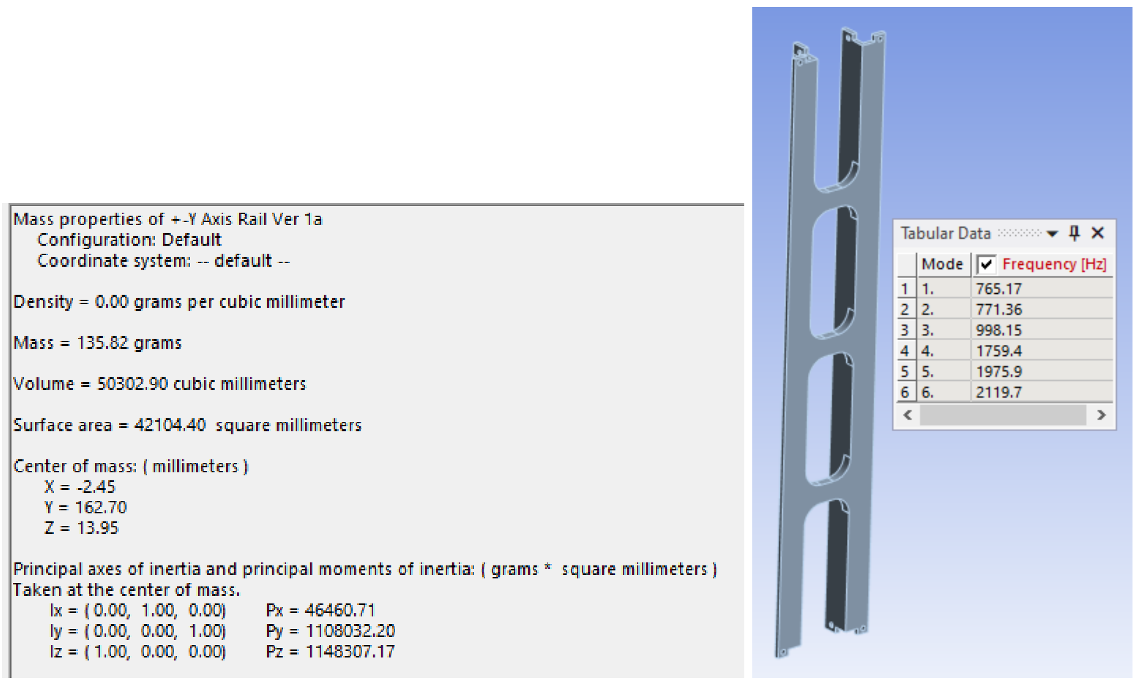
Servo HCI Sensor

https://cornell.app.box.com/file/1160691473614

Bradford Thrusters

https://cornell.app.box.com/file/1156365118253

A.4: Area moment of Inertia VS Mass moment of Inertia


Mass properties of +-Y Axis Rail Ver 1a
Configuration: Default
Coordinate system: -- default --

Density = 0.00 grams per cubic millimeter

Mass = 135.82 grams

Volume = 50302.90 cubic millimeters

Surface area = 42104.40 square millimeters

Center of mass: (millimeters)
X = -2.45
Y = 162.70
Z = 13.95

Principal axes of inertia and principal moments of inertia: (grams * square millimeters)
Taken at the center of mass.
Ix = (0.00, 1.00, 0.00) Px = 46460.71
Iy = (0.00, 0.00, 1.00) Py = 1108032.20
Iz = (1.00, 0.00, 0.00) Pz = 1148307.17

Tabular Data

	Mode	Frequency [Hz]
1	1.	765.17
2	2.	771.36
3	3.	998.15
4	4.	1759.4
5	5.	1975.9
6	6.	2119.7

Figure A.4.1: +/-Y Rails 1a Mass properties (Left), Modal Analysis (Right)

Mass properties of +-Y Axis Rail Ver 1b
Configuration: Default
Coordinate system: -- default --

Density = 0.00 grams per cubic millimeter

Mass = 140.50 grams

Volume = 52037.40 cubic millimeters

Surface area = 44393.50 square millimeters

Center of mass: (millimeters)
X = -2.37
Y = 162.70
Z = 13.95

Principal axes of inertia and principal moments of inertia: (grams * square millimeters)
Taken at the center of mass.
Ix = (0.00, 1.00, 0.00) Px = 46713.62
Iy = (0.00, 0.00, 1.00) Py = 1090675.40
Iz = (1.00, 0.00, 0.00) Pz = 1131129.42

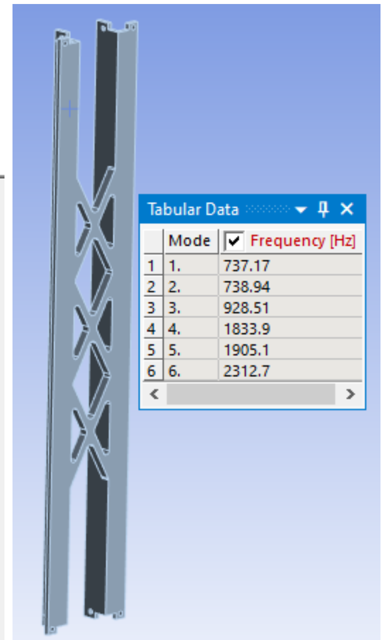


Figure A.4.2:+/-Y Rails 1b Mass properties (Left), Modal Analysis (Right)

Mass properties of +-Y Axis Rail Ver 1c
Configuration: Default
Coordinate system: -- default --

Density = 0.00 grams per cubic millimeter

Mass = 140.49 grams

Volume = 52034.77 cubic millimeters

Surface area = 44395.07 square millimeters

Center of mass: (millimeters)
X = -2.37
Y = 170.51
Z = 13.95

Principal axes of inertia and principal moments of inertia: (grams * square millimeters)
Taken at the center of mass.
Ix = (0.00, 1.00, 0.00) Px = 46706.29
Iy = (0.00, 0.00, 1.00) Py = 1143986.18
Iz = (1.00, 0.00, 0.00) Pz = 1184444.85

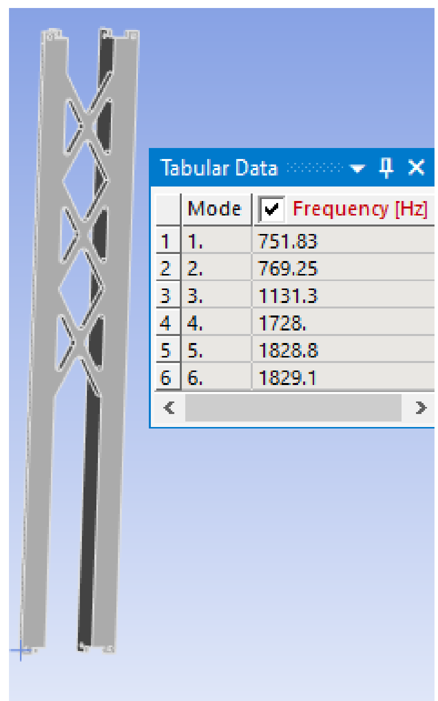


Figure A.4.3:+/-Y Rails 1c Mass properties (Left), Modal Analysis (Right)

Mass properties of +-Y Axis Rail Ver 1d
Configuration: Default
Coordinate system: -- default --

Density = 0.00 grams per cubic millimeter

Mass = 140.49 grams

Volume = 52034.77 cubic millimeters

Surface area = 47276.87 square millimeters

Center of mass: (millimeters)
X = -3.14
Y = 170.51
Z = 13.95

Principal axes of inertia and principal moments of inertia: (grams * square millimeters)
Taken at the center of mass.
Ix = (0.00, 1.00, 0.00) Px = 47597.73
Iy = (0.00, 0.00, 1.00) Py = 1144878.08
Iz = (1.00, 0.00, 0.00) Pz = 1184445.30

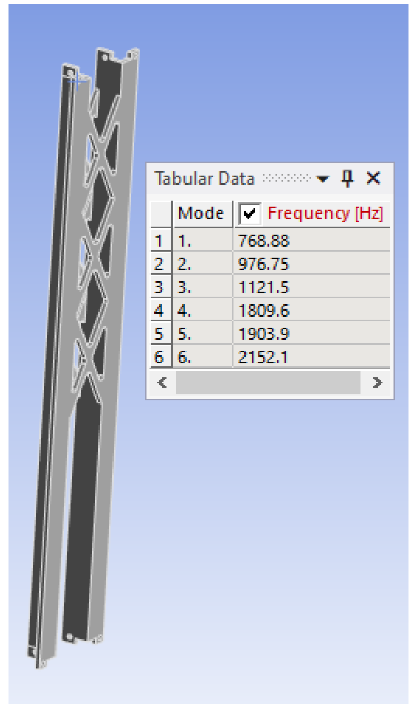


Figure A.4.4:+/-Y Rails 1d Mass properties (Left), Modal Analysis (Right)

Mass properties of +-Y Axis Rail Ver 1e
Configuration: Default
Coordinate system: -- default --

Density = 0.00 grams per cubic millimeter

Mass = 138.56 grams

Volume = 51317.04 cubic millimeters

Surface area = 45169.38 square millimeters

Center of mass: (millimeters)
X = -2.58
Y = 162.70
Z = 13.95

Principal axes of inertia and principal moments of inertia: (grams * square millimeters)
Taken at the center of mass.
Ix = (0.00, 1.00, 0.00) Px = 46360.63
Iy = (0.00, 0.00, 1.00) Py = 1222853.96
Iz = (1.00, 0.00, 0.00) Pz = 1263056.35

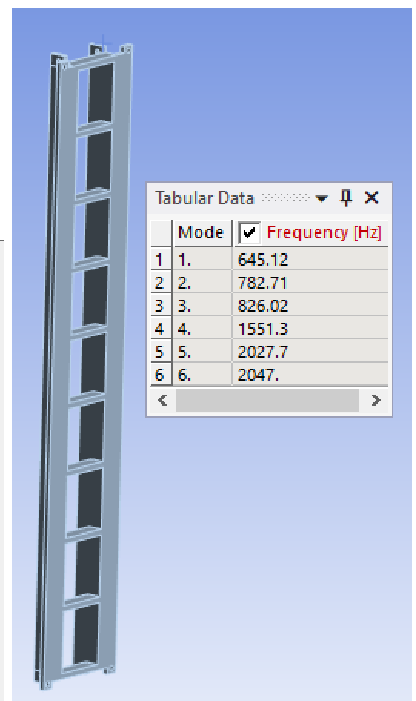


Figure A.4.5:+/-Y Rails 1e Mass properties (Left), Modal Analysis (Right)

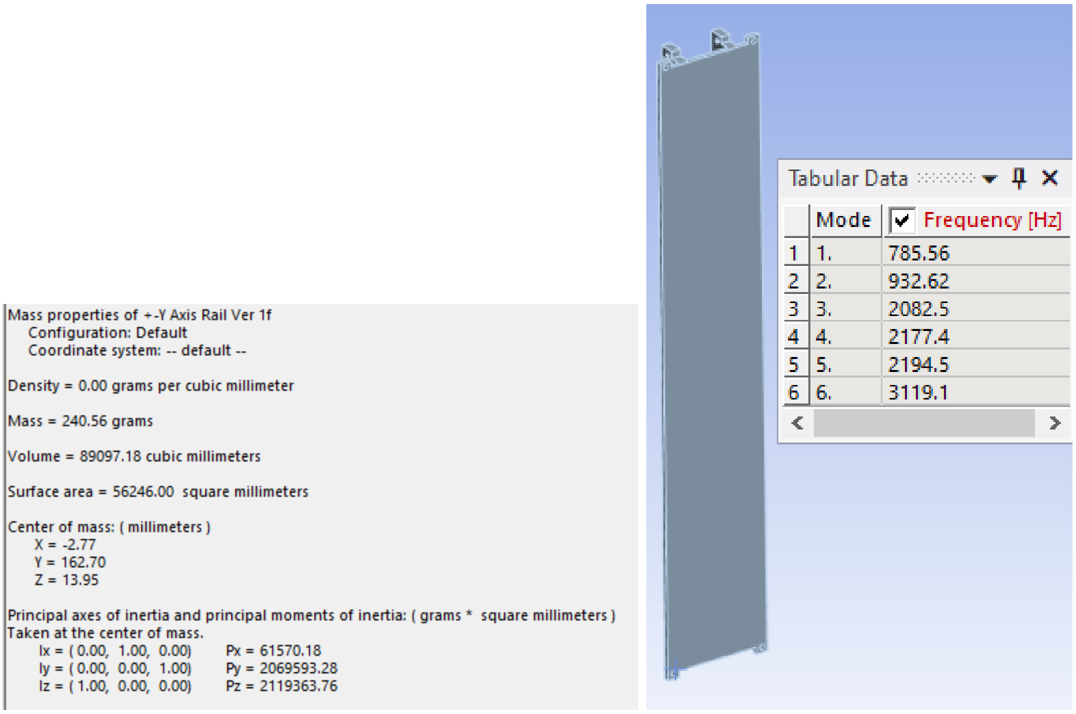


Figure A.4.6:+/-Y Rails 1f Mass properties (Left), Modal Analysis (Right)

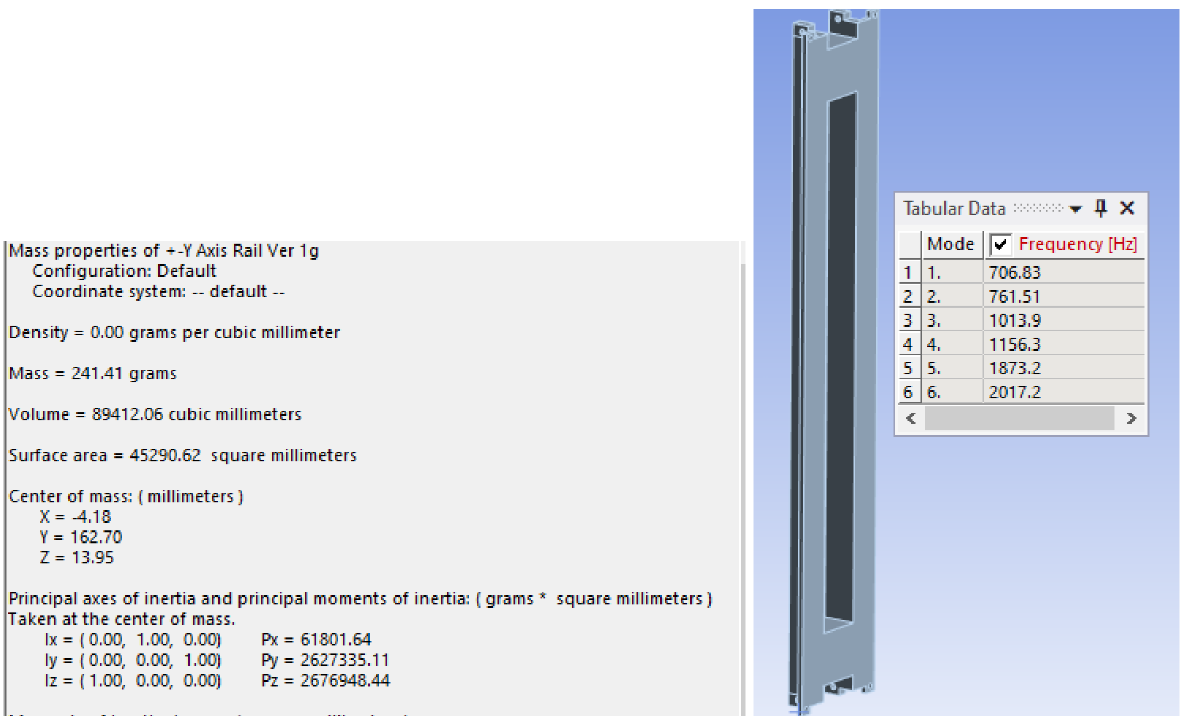


Figure A.4.7:+/-Y Rails 1g Mass properties (Left), Modal Analysis (Right)

This “experiment” was based on idea that by keeping mass relatively constant, as well as maintaining the same material/length due to requirements, the independent component of the inertia matrix shown on Solidworks could be tuned and adjusted one by one until the oen affecting natural frequency can be found. But as stated before, while the moment of inertia is related to the stiffness of the beam, the mass moment of inertia (what's on Solidworks) is different from the area moment of inertia (geometric). There isn't much distinction between the two in passing as they are both moments of inertia but referring to one or the other depending on context. This was not realized until after this experiment.

The intention behind running a mini experiment like this was such that if successful, there would be a conclusive way to compare structural designs even without modal analysis, which would speed up the design process significantly and reduce the amount of analysis needed. It also only took one Swanson session to get all these designs and modal analysis ran so while it was mostly useless, it wasn't done in compromise with something with higher priority.

This process also helped built lots of intuition behind rail designs in terms of stiffness. The last two figures (A.4.6 and A.4.7) were when the thickness of the rails was played with as well as testing out conclusions drawn from the natural frequency equations of a beam. For example, while placing masses at the ends like A.4.7 should have increased the moment of inertia, it resulted in a lower frequency, indicating that too much mass was dedicated to the other axis's moment of inertia as it is axis dependent. There were multiple designs done that can each be analyzed and theorized why the natural frequency either increased or decreased, which could be then applied to future designs. For example, A.4.4 had the highest frequency (prior to increasing thickness) but was not pursued due to manufacturability issues. So while overall not helpful in terms of finding a faster method for the iterative design of rails, reading and looking over the designs can help determine what will work and what won't in terms of patterns for the main structure of a CubeSat as it went through quite a few different designs.