SATELLITE GROUP FDR

Aidan Schaffer **Anjeli Estrada Alvarez** **Cady Brunecz** **George Maximos Zipitis** **Steven Kang** **Taylor Bayarerdene**

ABSTRACT

The objective of this project was to improve upon an existing design and manufactured tabletop Cassegrain telescope. Both the new and existing telescope are to be tested through simulations and experimentally, to correlate the results back to the simulation. The objectives for improvement are in detail in the requirements and specifications section but primarily include having a first vibrational mode greater than 120 Hz, being able to have a positive margin of safety with loads of 12G laterally and 18G axially and only covering 14% of the primary mirror surface area with the secondary and strut systems. Both telescopes will go through vibration hammer testing and simulations for correlating. The percent area covered was reduced from 35% to 21%. The initial model had a first vibrational mode of 30 Hz while the final model had a first mode of 55 Hz and a fourth mode of 127 Hz. While the struts vibrational mode was above 120 Hz, as the secondary subsystem got smaller to meet the area covered specification, this caused the vibrational frequencies of the mirror to decrease.

PROBLEM DEFINITION

Exploration of space requires high quality imaging from large distances. Optical satellites provide the ability to autonomously investigate planetary environments, including Earth's, monitor incoming meteoroids, and image relatively distant objects.

Space exploration is important to the entirety of the human population. The results of exploration yield advances in physics and engineering theory and application. Optical satellites also allow humans to monitor distant and remote parts of Earth.

The overall problem of creating high quality imaging of objects from large distances in space is critical for space exploration. By testing, analyzing and redesigning the tabletop Cassegrain telescope, the issues that arise in full-scale models such as the Hubble telescope will be better understood. We will be specifically working on the primary support, secondary mirror support and the alignment structure. By redesigning a small-scale satellite to meet our requirements and specifications, those concepts and design decisions can then be transferred to large-scale satellites. Based on our requirements, this will improve manufacturability, the strength to weight ratio, and increase the resonant frequency to ultimately provide better imaging of objects from large distances.

REQUIREMENTS, SPECIFICATIONS, DELIVERABLES

Our requirements include a setup for shaker/hammer, the device to be accessible for FARO for test/correlation, manual alignment of struts, manufacturable in house, and the optics will be properly kinematically constrained to the support structures to allow symmetric movement with uniform thermal growth.

Our specifications are: 10% improvement of strength-to-weight ratio, 10% reduction in total part count, first vibrational mode to be greater than 120 Hz when grounded at the Primary Mirror Support Structure (PMSS) interface and supporting all hosted hardware, the Secondary Mirror Support Structure (SMSS) shall interface to the Support and Alignment Structure (SAS) at three locations 120 degrees apart, the Primary Mirror Support Structure (PMSS) shall interface to the Support and Alignment Structure (SAS) at three locations 120 degrees apart, the SMSS and hosted hardware shall not obstruct more that 14% of the Primary Mirror clear aperture area, and the structure shall have positive margins of safety against yield and ultimate failure when exposed to a quasi-static load of 12 G laterally and 18 G axially simultaneously (lateral swept 15° increments.

Our deliverables are initial prototype, initial team presentation, schedule (work progress schedule and critical path management), preliminary design report, manufacturing readiness report, bill of materials, simulation of initial prototype, test and correlation of initial prototype, updated CAD design, simulation of updated model, updated prototype hardware, test and correlation of updated model, final team presentation, final design report, design day website, and design day poster.

CONCEPTS

Primary mirror subsystem team: The first design concept was to create a smaller, one piece, frame that still connects with the mirror/flexure in the same orientation. The main goal of this design was to increase strength to weight while also improving ease of assembly and reducing part count. The second design concept was to create a lightweight version of the baseline model. The goal of this design is to minimize the number of new parts requiring manufacture, while also increasing strength to weight.

Secondary mirror subsystem team: The first design concept, having the center mounting structure within the secondary mirror's vertical profile, was based on the baseline design. To meet the area obstruction specification, the sizing of all parts was reduced to reduce their projected areas. The second design concept, sketched in Appendix B 2.2, was having the secondary mirror support connect to the struts outside the primary mirror's vertical profile. While this design concept would be better to meet the area covered specification, it would severely impact the alignment of the struts. Therefore, having the structure within the secondary mirror's vertical profile was chosen.

Struts subsystem team: The first attempt on the vibration analysis was to shorten the wires and use the same cross-sectional area for all sections of the struts. The second attempt was to use variable and hollow cross-sectional areas. The design concepts can be found in Appendix B 2.3. Both attempts met the frequency specifications, the first mode of the structure to be 120hz, but there is still room for optimization as the first mode of the structure is currently at 139Hz. The results for the second attempt can be found in Appendix A 1.4.

MECHANICAL ANALYSIS

In terms of material selection, 6061 aluminum was chosen for almost all of the components due to its relatively low cost and weight, high strength, and easy manufacturability. By making the entire structure, including the mirrors, out of one material, we can athermalize the design, meaning that thermal loads will not have an effect on the relative geometry of the optical path. However, the one component that is not made of aluminum, aside from fasteners, are the flexures. These are made from spring steel, due to its high strength and elasticity, meaning it can be subjected to large loads/deformations, and still retain its original geometry.

For the tolerance analysis, we will be calculating a worst-case scenario for the distance between the primary and secondary mirror strut connection points. The distance of 20.13 inches is a fixed requirement, and each strut must be able to precisely be at that length. With the differential screws, we can adjust with an effective 70 threads per inch (TPI), which gives us 0.014 inches of adjustment per revolution, which is enough resolution to hit the 20.13 inches requirement. The calculation for the effective TPI is shown in the equation below:

$$TPI_{eff} = \frac{1}{\frac{1}{20} - \frac{1}{28}}$$

Each strut is composed of several components, including the wire clamp (t), the wire, the jam nuts, the press fit parts (PFF/PFC, solid section (ZF/ZC), and tube section (AB). Each individual section has a tolerance, but we chose to prioritize the assembly tolerance because that is the length we must hit for the requirement. We have ten threads of adjustability on the fine and coarse side, giving us a total adjustment of 0.85 inches of adjustability. During manufacturing, we tripled this adjustability because we tapped through the press fit pieces, allowing the threaded rod to travel all the way into the tube. The worst-case scenario for a strut is calculated to be a maximum of 19.96 inches and a minimum of 19.624 inches, assuming during manufacturing we adhere to the individual part tolerance. The adjustability of 0.85 inches would allow us to extend to the 20.13 inches required distance, combined with the 0.014 inches per revolution resolution. The worst-case scenario calculations can be found in figure 1.1 under Appendix A.

For the fatigue analysis, the relevant material is Aluminum 6061 which does not have an endurance limit like steel. However, a fatigue strength of 14000 psi is given for a life of 5 x 10^8 cycles. Due to the cost associated with launching into space, it would be critical to ensure that the satellite can withstand many cycles. However, it will only experience the forces at launch once. Through FEA, the maximum stress experienced with an 18g axial load is ~1080 psi which is less than the fatigue strength. Therefore, fatigue should not be a foreseeable issue for this design.

For the torque calculation we will be analyzing the torque required for the ¼-20 bolt that joins the primary strut holder to the primary base. For this calculation, we are going to assume that this is a permanent connection, thus $F_i = 0.90F_p$. We will also be using a K value of 0.12 as we are using anti-seize for all of our bolted connections. All values for the properties of the bolt will be found from Shigley's 9th edition, Table 8-2, Table 8-9, and Table 8-15. The calculation is as follows:

$F_p = A_tS_p = 0.0318 \text{ in}^2 * 33 \text{ kpsi} = 1.0494 \text{ kips}$
$F_i = 0.90F_p = 0.90*1.0494 \text{ kips} = 0.94446 \text{ kips}$
$T = KF_id = 0.12*0.94446 \text{ kips} * 0.25 \text{ in} = 2.36 \text{ ft-lbs}$

The torque required for our ¼-20 bolted connections for the strut holders to the primary base is 2.36 ft-lbs.

A fundamental mechanical concern was ensuring the primary and secondary mirrors did not experience significant stress from thermal loads that would cause them to deform and thus alter their imaging capabilities. To address this, we used 3 flexures made of thin spring steel for mounting each mirror. These flexures are stiff in the plane tangent to the mirrors' side surfaces, which allows them to perform their primary function of mounting the mirrors to their support systems. However, they are only stiff in that plane, which means they are compliant to the mirrors' thermal expansion and mitigate the imposed stresses. Additionally, we mounted each set of flexures at 120° equidistant locations, which minimizes the mirrors' lateral translation and keeps them centered along the axis of the satellite.

One example of a finite element method used in the project is the frequency analysis of the entire assembly. The specification is for the frequency to exceed 120Hz. All the modes, except the first three, pass the specification. However, the first three modes are not expected to impact the imaging performance of the assembly. A detailed result of the analysis can be found on Appendix D.

MANUFACTURING

Build costs:

	Cost ($)
Hardware costs	808.66
Purchased shop time ($100/hr)	200
Team manufacturing time costs ($100/hr)	13,350

Development costs:

	Hours
Aidan	44
Anjeli	91
Cady	61
George	95
Steven	73
Taylor	17
Total Hours	381
Total Costs in $ ($100/hr)	38,100

Total build cost comes to $13,350, total cost in development is $38,100 which results in a total project cost of $51,450. The total BOM can be found in Appendix D, figure 5.1.

Struts:

Dimensions of the struts were chosen from the simulation model, where we referenced available tube diameters from McMaster to reduce manufacturing times. In the updated simulation design, the outer diameter of each of the three sections were different, and during manufacturing we decided to keep them all the same to reduce manufacturing hours. We updated the CAD model and reran the vibration analysis to make certain that we will still be able to meet the specifications. The simulation model is an ideal model, which did not include the extra lengths of the press fit parts, threaded rods, and jam nuts. During manufacturing we had to account for the discrepancy and turn the tube's length down with a lathe. Since the simulation model was ideal, the connections were modeled as ideal as well, which neglected the fact that threaded rods wobble when connecting sections together, so we used jam nuts to reduce the wobble, introducing the previously mentioned extra length. Initially the fine and coarse adjustments were allotted 10 threads each in the press fit part, which would give us a combined 0.857 inches of adjustment. Jim Alkins suggested during manufacturing that we tap through the entire press fit piece since it goes into a hollow tube. Tapping through the press fit allowed more than 2 inches of total adjustment, greatly reducing the tolerance stress on the assembly. We also prioritized the assembly length tolerance over the individual part length tolerance because we had the extra adjustment, whereas the total length of the strut is a fixed constraint because the primary mirror and the secondary mirror had to be a certain distance apart.

Primary:

The primary portion of the satellite is composed of the primary base, primary strut holders, strut bolts, and primary mirror. For all of these components, 6061 aluminum was used. The reason for this is because this is meant for aerospace use, which means we need to prioritize cost, weight, and strength. Aluminum is the optimum material across the board for this application. It is a low cost, lightweight, high strength material. From a thermal perspective, aluminum has a high coefficient of thermal expansion, however, since all of the parts are made from aluminum, they will all expand and contract at the same rate, leaving this to not be an issue.

For manufacturing, half of the parts were made on the HAAS CNC Mill/Lathe and the other half were made manually. The strut bolts and holders were made on a manual mill and lathe due to time constraints. There was a large backlog for the HAAS, so we made it a priority to only request parts that required CNC for the HAAS and to do the rest by hand. The strut bolts started off as a ⅜” hex rod that was then turned on the lathe with a taper down to the die size for 1/16 NPT. We then drilled 2.1mm down the center of the bolt and then cut the 1/16 NPT threads. The last step was to create two slits down the entire length of the threads with a 0.020” slitting saw. The strut bolt holders were created from 1”x2” bar stock and the angle were created using an angled vise. The mirror and base were created on the HAAS machine, however, the base has counterbored holes that were required to be drilled and tapped manually. All of the parts, including hardware, would amount to under $200, however, we did spend more than this as we had to recreate some parts due to human error.

Secondary:

The secondary subsystem is composed of the secondary mirror, flexures, flexure mount, strut holders, and central hex shaft. The CAD for these parts can be found in Appendix C 3.2.

The most difficult component to manufacture on the secondary subsystem is the strut holders; these are the parts that connect to the struts. These parts need to have a threaded hole that is angled along two different axes. These two angles allow the hole to be normal to the strut bolt and to align with the strut mounting position on the primary's base. Initially, the part was designed with a compound angle which is not able to be manufactured on a four-axis machine; a figure of this design can be found in Appendix C 3.1. Therefore, the design was changed to have six separate parts that connect to each strut. The first angle will be set by rotating the parts and setting the angle with pins, while the second angle will be machined to the part itself. A figure of the new design is in Appendix C 3.2. The second angle for the new design can be machined on the fourth axis of the Haas.

The flexures were outsourced to Bill Mildenberger as they had to be milled out of spring steel and despite the cost of $200, this was the most efficient choice due to the machining complexity and accuracy required.

The rest of the parts were made either on the Prototrak or on a manual mill/lathe. The flats on the secondary optic to mount the flexures were done by using a dividing head and a mill. The hex shaft (pinholes and fastener holes on each face) could have been done manually but was done on the Prototrak since the operations had to be repeated on multiple faces. A hole along the axis of the hex shaft was milled out on a manual lathe and tapped for later fastening to the flexure mount. The outline of the flexure mount and hexagonal indent were done on the Prototrak due to the angled milling operations and the flexure holes were done on a manual mill; a figure of the flexure mount part is in Appendix C 3.3. Additionally, 1/8" holes were drilled on the corners of the hexagonal indent to drill out the radius on the corner left by a 1/8" end mill. This would allow for the vertices of the hex shaft to fit into this hexagonal indent.

Discussion:

If the system were to be scaled to 1000 systems, some changes we could make would include narrowing down on the number of unique parts that we have. For the secondary strut holders have two different designs, but if this were to be scaled, we could redesign the secondary strut holders to be the same for all 6. Another change we could make is to create CNC code for all of our parts to automate the process and limit the amount of manual work required. This would speed up the process to create one assembled satellite. The last change we could make is to come up with an alignment device that doesn't require as much time to hound in on the right amount. Currently we use a differential screw, which requires a lot of hands-on work to align the structure properly

TEST PLAN AND RESULTS

Although multiple specifications were defined, only two of them will be tested. The first test evaluates the response of the sub-systems against the frequency specification, while the second test evaluates if the structure can be used for imaging.

Initially, the team was divided into 3 sub-teams: Primary, Secondary and Alignment, each team worked individually to improve the initial design provided by Professor Muir. Up until Gate C, the Alignment sub-team using simulations was able to pass the frequency specification, ensuring that the whole structure has the first mode higher than 120Hz. However, after Gate C, when all the updated designs were integrated in the complete assembly, the frequency requirement was satisfied for mode 4 to10. Although the first 3 modes didn't pass the specification, the team along with the sponsor decided to proceed with the fabrication. This decision was based on simulation results, which showed that the first 3 modes wouldn't affect the imaging performance.

The frequency specification will be assessed through physical testing on two components, the fabricated primary base, as a hammer test, and the strut assembly as fixed-fixed boundary conditions at the two sides of the wires. Both tests were simulated for correlation methods. These simulations identified the high displacement and exciters' locations, which informed the placement of the accelerometers in the physical test. As shown in Appendix-D both Primary base and Strut assembly pass their respective frequency specifications. The Primary base has all modes greater than 600Hz, while Struts assembly greater than 120Hz.

The simulated fundamental frequency of the simplified strut model is 130Hz, while the test of the fabricated strut model shows a frequency of 106Hz. In the simplified strut model screws were not included and the strut bolts were fully threaded. The fabricated model has a pre-set adjustment on both the coarse and the fine adjustment sides; however, the tested model did not incorporate that adjustment. The additional weight, imperfect connection caused by the screws and differences in total strut assembly length, contributed to the lower fundamental frequency observed in the test.

Both the primary mirror and primary base have relatively low differences between their simulated and test results. The simulated frequencies were 798Hz and 608Hz, while the tested frequencies were 736Hz and 617Hz, respectively. These differences are likely due to differences in material properties from the simulated 6061-Aluminum and the actual material used. Additionally, there are likely minor differences in thickness and geometry, as the simulations used an idealized model.

Once the assembly of the fabricated structure is completed, imaging testing will be conducted. Although no simulations were performed for this requirement, the area covered was calculated using projected curve from a top view in the CAD software. The area covered represents the percentage of the primary mirror covered by the Struts and Secondary mirror support. The specification for the area covered was to be less than 14%. However, the team in agreement with the sponsor decided to proceed with fabrication of the structure that has an area covered as 21%. Even though the specification was not met, there was a significant improvement in the area covered as well as weight reduction. In terms of imaging, the slight increase in the area covered will introduce slightly more shadowing on the resulting images.

INTELLECTUAL PROPERTY

This design is not patentable as it is not novel. Each of the design decisions were based on methods that already exist, and the overall idea of a Cassegrain optical satellite is not new. Many companies are working in this space, most notably Raytheon Company with the most related patents.

One existing relevant patent was filed by the US Department of Energy in 1998 under Roderick A. Hyde with patent number US6219185B1. This patent, similar to our design, features a primary optic for collecting light into the system and a secondary optic for transmitting light into the imaging system. However, instead of struts or some other rigid alignment support system, the optics are in separate, free-floating spacecraft. The primary intercepts incoming light over its entire aperture while the secondary "sweeps" across the primary's focal surface, converting the light into images.

Another existing relevant patent was filed by an individual, Youngwan Choi, in 2021 with patent number US11668915B2. This design is also like ours, with a primary mirror and a secondary mirror attached rigidly. However, the secondary optic is convex, instead of concave, and there are numerous other mirrors and lenses that reflect and collect the incoming light for conversion into images.

These examples show how different aspects of the premises of our design are longstanding and cannot be newly patented.

SOCIETAL AND ENVIRONMENTAL IMPLICATIONS

The impact of a project like ours, in terms of developing an optical satellite, has no direct impact on public health or safety. Nearly the entirety of the satellite is made of aluminum, which can be toxic to the environment during the mining and refining processes. However, it can also be recycled nearly in perpetuity while requiring far less energy than is required in the initial manufacturing process. Therefore, to minimize the environmental effects of building an aluminum satellite, we can use recycled aluminum.

Additionally, launching the satellite carries the pollution associated with rocket launches, like the potentially harmful byproducts of burning rocket fuel. The effects of individual launches are negligible on a global scale, but with the rising number of launches in recent times and the acceleration of climate change, it has become a problem that requires addressing. One possible avenue would be decreasing the number of harmful byproducts released by either using fuel that requires less while still providing enough thrust to clear the atmosphere or using fuel that burns cleaner.

Providing improvements for an optical satellite is beneficial to society on a large timescale. By developing techniques to manufacture new satellites that provide superior imaging, require fewer parts, are cheaper to make, and/or are stronger, we can enable more and better imaging in space that can further society's scientific goals of space exploration. These goals include researching materials not found on Earth, observing distant planets and objects, and even looking for signs of life.

RECOMMENDATIONS FOR FUTURE WORK

In terms of future work, the most important task would be additional simulation, testing and correlation. Unfortunately, we did not have the time nor resources to complete a full thermal simulation, test, and correlation study, which would be something that could be continued with future work. This would allow us to test the optic displacement specification under the given thermal loads. Ideally, we would like to perform some environmental testing on the system as well. This would involve applying significant vibration/temperature cycles, to simulate the conditions experienced during launch and orbit. This would, however, run the risk of being a destructive test, so the value of such a test would have to be deeply considered.

Additionally, the geometry and materials used in the design could very likely be further optimized with more time. For example, the primary base went through a multitude of changes and redesigns, but the final design was still a relatively simple geometry, as it was manufactured by us. With a larger manufacturing budget, a much more complex yet efficient design could very likely be created. In terms of materials, aluminum was chosen for all components mostly because of its price and relative strength, but with more research and resources, alternative materials could potentially be selected, taking into account their thermal properties.

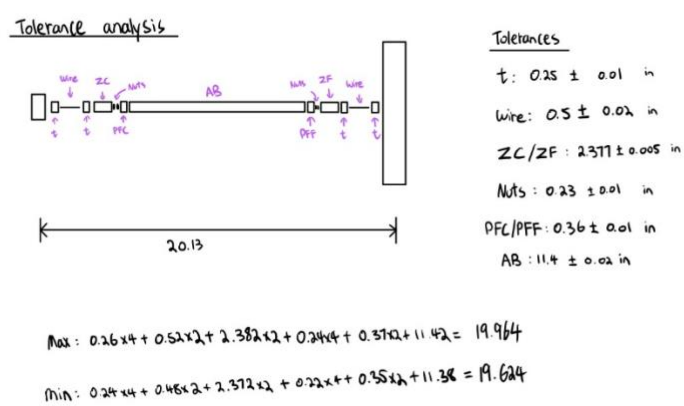
Finally, we would be very interested in taking real images and optimizing the imaging process for our design. This would involve firstly setting up a controlled lab environment, and then moving on to a harsher, real-life environment. Once the imaging process has been solidified, software could very likely be created that allows us to take multiple images and stitch them together, to make up for the areas blocked by the struts and secondary mirror structure. This would allow us to image an entire object, regardless of the primary mirror obstructed area.

ACKNOWLEDGMENTS

We would like to thank our sponsor, Professor Chris Muir, for all of his help throughout this project. Additionally, we would like to thank Chris Pratt, Jim Alkins, Alex Prideaux, Sam Kriegsman, Bill Mildenberger, Sheli Hernandez, Mike Pomerantz, and Rob Bauer for their contributions to this project.

APPENDIX A

1.1: worst-case tolerance analysis of strut



1.2: Primary subsystem Pugh matrix

Design:	Baseline	Aidan	Cady
Strength to Weight	0	+	+
# of Unique Parts	0	+	0
Ease of Assembly	0	+	0
# of New Parts	0	-	0
Total	0	2	1

1.3: Secondary subsystem Pugh matrix

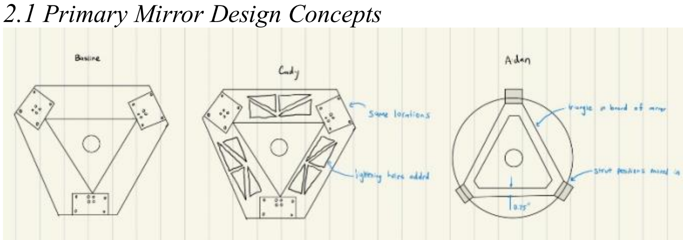
Factor:	Baseline	Within primary profile	Out of primary profile
Area Obstruction	0	+	+
Vibratio n mode	0	0	-
Axial & lateral strengths	0	0	0
Seconda ry Mirror Adjustments	0	0	-
Total	0	1	-1

1.4: Struts subsystem Pugh matrix

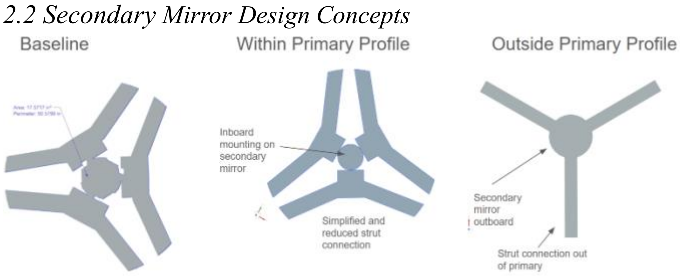
Factor:	Ba seline	Short er wire, same CS	Short er wire, variable CS	Sphe rical joints, variable CS
Connec tion	0	0	0	-
Weight	0	0	+	+
Ease of manufacture	0	0	-	0
Cost	0	0	+	-
Stabilit y	0	+	+	+
Total	0	1	2	0

APPENDIX B

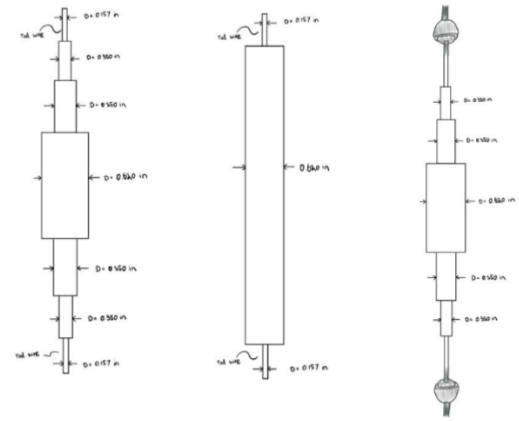
2.1 Primary Mirror Design Concepts



2.2 Secondary Mirror Design Concepts

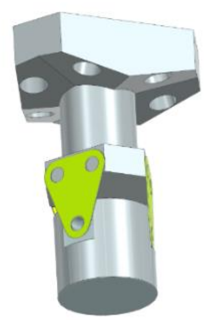


2.3 Struts Design Concepts

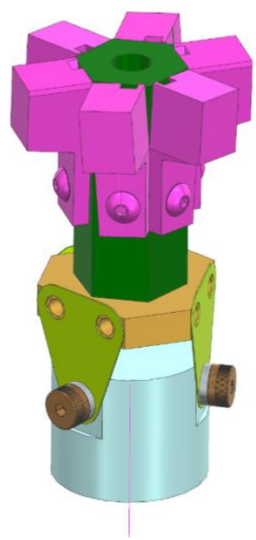


APPENDIX C

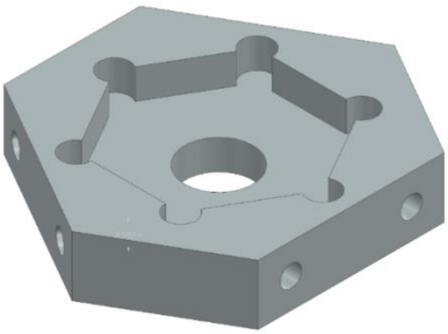
3.1: Initial compound angle secondary design



3.2: New design with single angled plane and rotation by pins



3.3: Flexure mount CAD

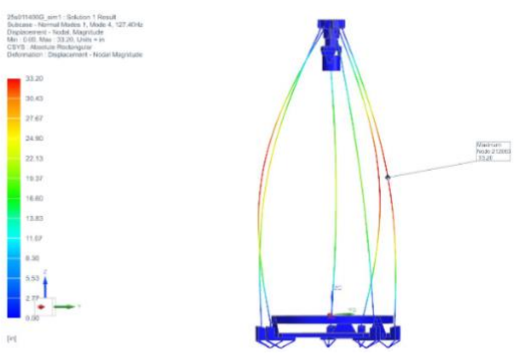


Appendix can contain figures and tables as well as other relevant or supporting information that is not included in the body of the report. If they are contained in the appendix, they should be referenced somewhere in the main body of the report.

- An assembly drawing and BOM can be included in the appendix to show the system.

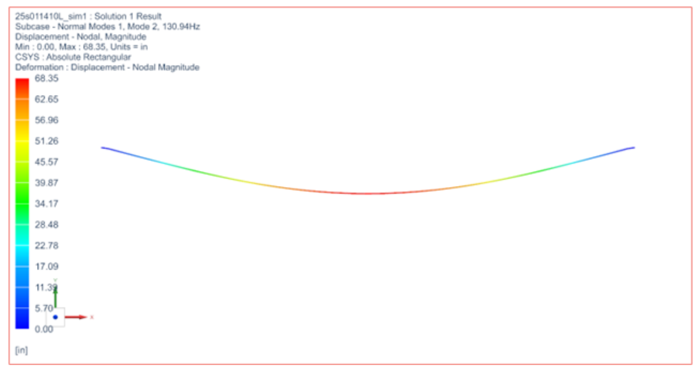
APPENDIX D

4.1 Frequency Analysis of entire Assembly shows that modes 4 to 10 exceed 120Hz.

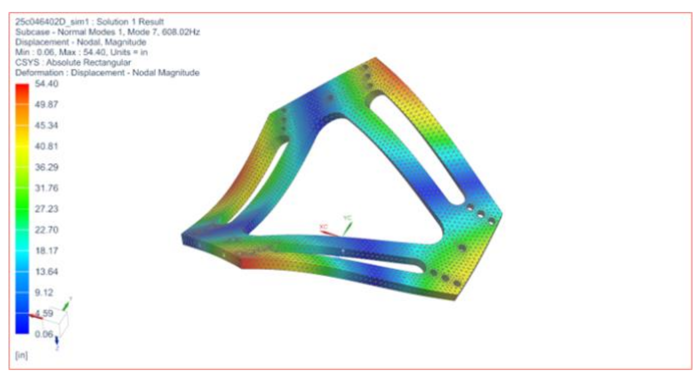


4.2 Correlation Model for Struts assembly indicating the location of exciter (Red arrow) and the sensor locations (Green arrows).

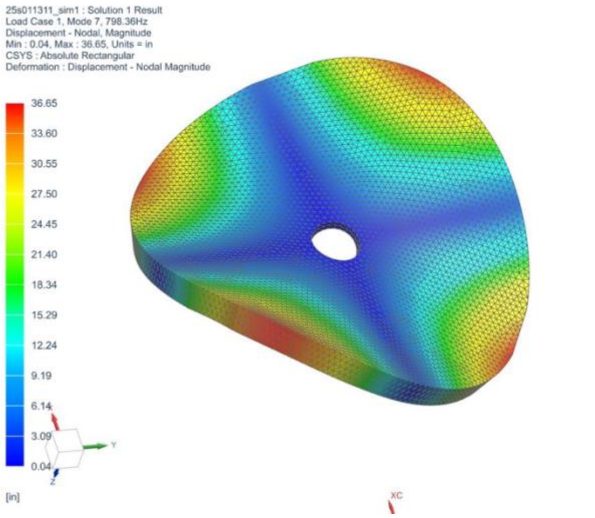
4.3 Correlation Model for Struts assembly indicating the mode to be greater than 120Hz.



4.4 Correlation Model for Primary base indicating the mode to be greater than 600Hz.

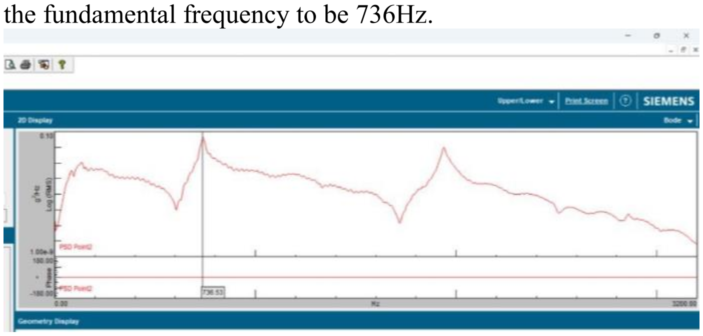


4.5 Correlation model for Primary Mirror indicating the mode to be greater than 700Hz.

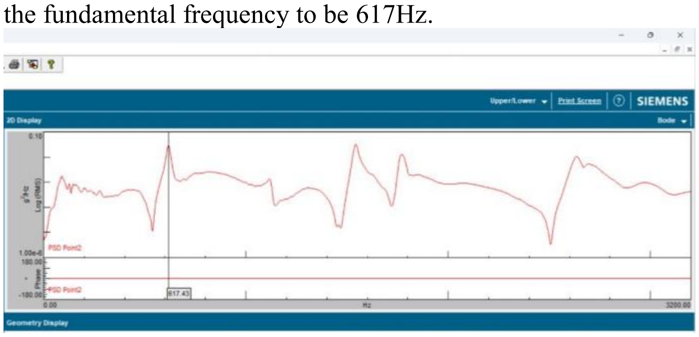


4.6 Actual vibration test results for struts assembly indicating the fundamental frequency to be 106Hz.

4.7 Actual vibration test results for the Primary Mirror indicating the fundamental frequency to be 736Hz.



4.8 Actual vibration test results for the Primary Base indicating the fundamental frequency to be 617Hz.



APPENDIX D

Part #	Part Name	Quantity
25s011301G	AB - 1	6
25s011305G	ZF - 5	6
25s011303G	PFF - 3	6
25s011302G	PFC - 2	6
25s011304G	ZC - 4	6
25c024902A	Strut Mounting Part Left	3
25c024901A	Strut Mounting Part Right	3
98381A504	Dowel Pins	12
25c012525A	Hex shaft	1
25s012301A	Secondary Flexures	3
25S011310A	Secondary Mirror	1
25c046402A	Primary Base	1
25s016317A	Strut Holder	6
25s011311B	Primary Mirror	1
25s011312B	Primary Flexure	3
91259A465	Alloy Steel Shoulder Screw	9
94773A768	18-8 Stainless Shims	9
91255A523	Button Head Hex Screw	6
98381A5003	Dowel Pin	12

Figure 5.1: Total Assembly BOM