Harmonic Analysis of SPT4 300K-4K G10 Support Structure

SPT4

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April 25, 2022

Overview

This document presents a harmonic analysis of the 300K-4K G10 Support structure with simplified models of the seven optics tubes attached to the 4K plate. Something to note about this analysis is that it does not yet include the 1K plate and the associated attachment to the 4K plate as this is thought to be a lower order perturbation. The goal is to assess the stiffness of various G10 support structure geometries in order to gather evidence for the best design. This is **not** intended to be the final design decision, but rather to show the analysis that lead up to it. As a result, this write-up does not contain or reflect all the design constraints that the final design might abide by. It is also important to note that each design is considered purely for it's geometric stiffness, and thus all are chosen to have similar Area/length ratios so as to be evaluated on the same thermal ground.

Geometry:

The figure below shows a snapshot of the simplified model for the support structure and the optics tubes without the shells[1]. The 300K to 50K section of the support structure is seen to be significantly shorter than the 50K to 4K structure. This is due to the accomodation of the filtering optics that reside at 50K and need to go in front of the optics tubes, who's 4K lens has it's own distance constraint from the front of the 300K vacuum plate.

The optics tubes are simplified to reduce computation time and allow for better meshing. The optics tube is simply modeled as a cylinder with an attachment that interfaces with the 4K plate. The value of the mass as well as the center of mass is overridden to reflect the true mass distribution of the optics tube due to the lenses, baffling, and detector wafer. Details of this simplification can be found in following sections.

A sketch of the defining dimensions is shown below in figure 3.

[1]The shells are not included because they are difficult to mesh and are not thought to contribute much to the eigenmodes of the structure. This is because analytically speaking, the shells are connected to the same "nodes" as the various stages support structure, and thus, vibrate in parallel with the structure rather than serially.

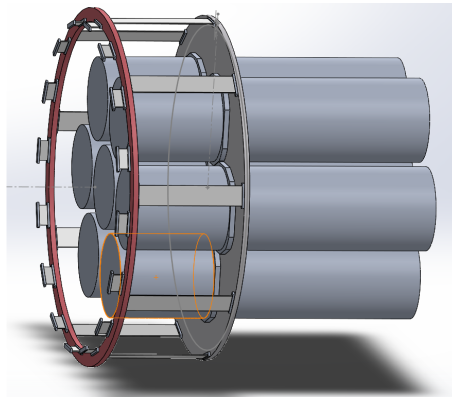

Figure 1: Here is a snapshot of the 300K-4K G10 Structure including the 50K interfacing ring, the 4K plate, and a simplified model for the optics tubes. No further geometry is considered for this analysis.

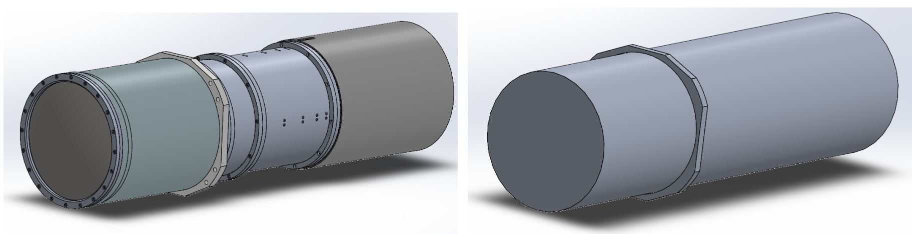

Figure 2: The left is the real optics tube and the right is the simplified one. The main difference for the simplified optics tube being the uniform diameter and overriden mass properties. The real optics tube's mass distribution including lenses, baffling, etc is found and transferred to the simplified optics tube.

Model Setup

This section details the setup of the FEA model. First the assembly is created, as snapshots shown in the previous section. Since Solidworks maps the mating relationships in assemblies to it's best guesses for simulation connectors, careful consideration had to be taken here so as not to introduce inaccurate connections between elements.

0.1 Constraints:

The G10 tabs are simple in that rigid connections to the corresponding thermal stages are desired. This is achieved by applying coincident mating relationships between the faces of the G10 tab assembly and the corresponding thermal stage plate. Mounting holes in the

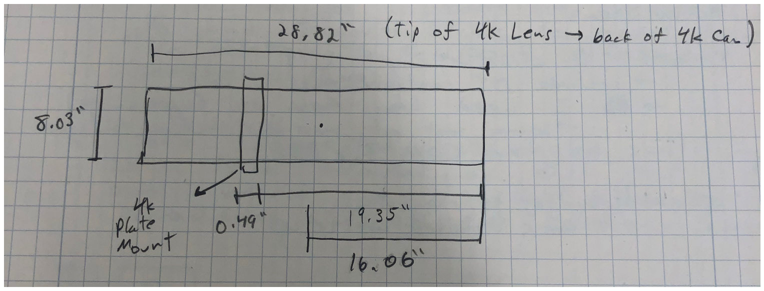

Figure 3: A simple sketch showing the overall dimensions of the optics tube and the location of the center of mass relative to the back plate.

stage plates and the G10 tab assembly are used to align the tabs to the proper orientation before being suppressed by the Solidworks Simulation engine to simplify the meshing.

The optics tubes are constrained by first aligning the tube to the corresponding mounting hole in the 4K plate. The only mating relationships that remain after alignment is the coincident mating relationship for the inner edge of the tube's mounting bracket and the outer edge of the corresponding mounting hole on the 4K plate. The translation and rotation of the optics tube relative to the 4K plate is constrained by applying a keyed pin connection to the outer curved cylindrical face of the optics tube and the inner curved face of the 4K plate mounting hole. The translation and rotation of the cylindrical body of the optics tube and it's own mounting bracket are constrained relative to each other in the same manner.

Similar constraints are applied to all the other designs that will be presented alongside this one. The main goal is to rigidly constrain the G10 members to the respective Temperature plates to assess the geometric stiffness with this boundary condition applied. In reality, these boundary conditions may vary from design to design and is an issue considered separate from this analysis.

0.2 Optics Tube Center of Mass:

The magnitude of the mass as well as the center of mass of the simplified optics tubes are overridden from the model values to be in line with reality. This is achieved by taking the Solidworks model for the real optics tubes and changing the non-aluminum materials to reflect the real material (i.e. lens material to silicon, baffling to Polyprolylene (PP Film in SW), etc). This gives a mass of 17.34lbs and a center of mass location of (0, 0, 6.84") relative to the origin of that assembly file. The absolute location is found before applying it

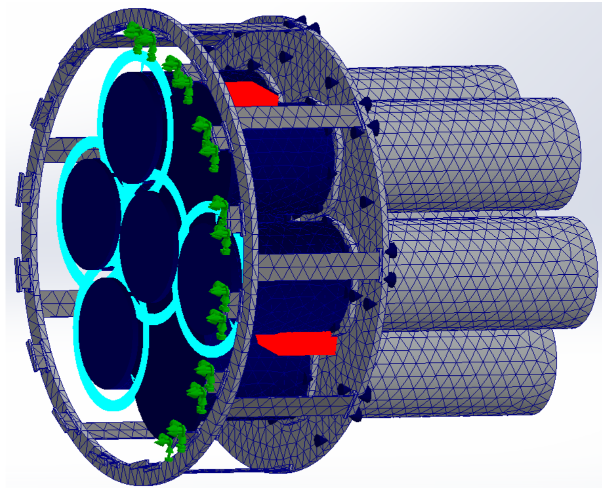

Figure 4: The simplified model meshed with a blended-curvature mesher for the interested reader.

to the simplified optics tube by comparing the relative locations of the parts relative to the respective world origins in each file. The result gives the location of the center of mass as shown in the sketch of the simplified optics tube dimensions in figure .

0.3 Boundary Conditions and External Loading:

The entire structure is then fixed at the 300K interface (the far left in figure 1), which is rigidly connected to the vacuum plate and eliminates any rigid body modes. Further, a 5kg distributed mass is placed on the far right end of each optics tube to model the attached detector modules.

0.4 Mesh:

There isn't much problem in meshing using the blended curvature-based mesher after suppressing all the mounting holes. An example of mesh is shown below

The FEA solver then calculates the first five modes, which takes about 30-40 minutes.

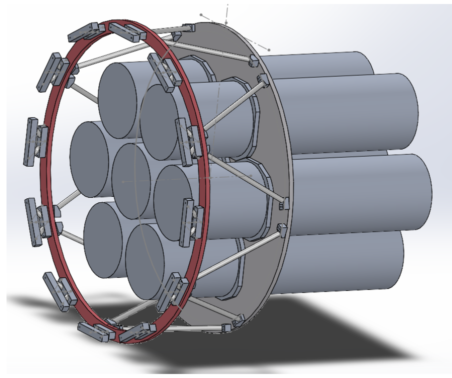

Figure 5: The Triangular Bipod support structure design.

1 Considered Designs and Reasons

This section details the designs analyzed for this write-up and the arguments to motivate the reasons for these design considerations.

The tabbed geometry that has been shown up to this point in figure 1 is considered for it's simplicity in assembly and previous use in CMB experiments such as Simon's Observatory and SPIDER. The strength of this design relies primarily on the large width of the tabs, which contributes cubic stiffness to the overall moment of inertia of the structure. The weakness is that, while cubic stiffness is great, the absorption of mechanical deflection in the G10 tabs is flexure, which is less stiff than absorbing deflection in compression/tension. This idea motivated the consideration of the triangular bipod member geometry in the next support structure.

The second consideration is a triangular bipod design in figure 5, which is motivated by an attempt to absorb the mechanical deflection in compression/tension of the G10 members rather than flexure. The number of bipod members and the thickness of the cylindrical struts are chosen to optimize the angle of the bipod so as to achieve this. The resultant angle of the bipods is 55 degrees relative to the axis of deflection.

The third design considered is a cylindrical rolled sheet of G10 in figure 6. Geometrically this should be the stiffest structure since it contains the most axisymmetric material, which then begs the question, is this mechanically feasible given the thermal constraints?

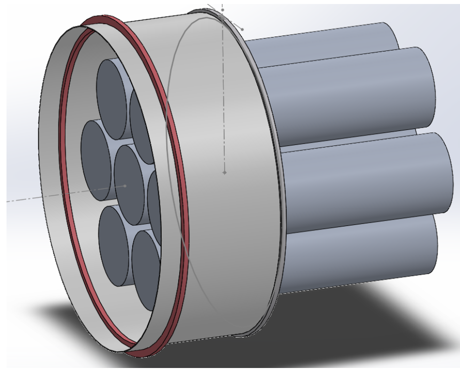

Figure 6: The cylindrical support structure design.

Resonant Frequencies

Natural Frequencies

Table 1: The table below summarizes the first five eigenvalues of the above analysis for the Tabbed Geometry in figure 1.

Eigenmode Number	Frequency (Hz)	Mode Shape
Mode 1	6.8	Flexure
Mode 2	6.86	Flexure
Mode 3	18.0	Torsion
Mode 4	48.8	Buckling
Mode 5	50	2nd Order Flexure

Table 2: The table below summarizes the first five eigenvalues of the above analysis for the Triangular Bipod Geometry in figure 5.

Eigenmode Number	Frequency (Hz)	Mode Shape
Mode 1	16.6	Flexure
Mode 2	16.6	Flexure
Mode 3	42.7	Buckling
Mode 4	46	Torsion
Mode 5	48	4K Plate Rotation

Table 3: The table below summarizes the first five eigenvalues of the above analysis for the Cylindrical Geometry in figure 6.

Eigenmode Number	Frequency (Hz)	Mode Shape
Mode 1	20.3	Flexure
Mode 2	20.3	Flexure
Mode 3	38.0	Buckling
Mode 4	49.9	4K Plate Rotation
Mode 5	50.3	4K Plate Deformation

Mode Shapes

1.1 Static Deflection

Another easy way of assessing stiffness is the magnitude of vertical deflection under inertial loading in the horizontal orientation. The only addition that differs from the previous model setup is that gravity is defined in the direction perpendicular to the optical axis. The results are summarized for all the support structure design.

Concluding Remarks

Given no broader context, the cylindrical support structure performs the best, followed by the triangular bipod, and the tabbed geometry. Any choice from here on a particular design

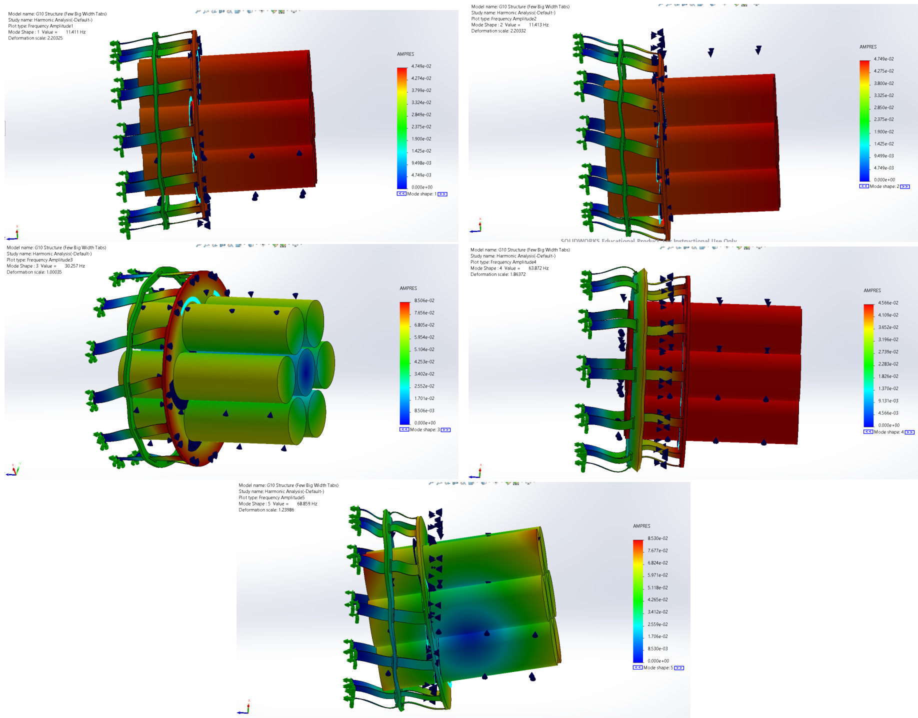

Figure 7: Mode Shapes indexed from left to right and top to bottom.

Table 4: The table below summarizes the static deflection of each of the designs under gravitational loading in the horizontal orientation.

Design	Max Deflection Magnitude (mm)
Tabs	4.7
Bipod	0.90
Cylinder	0.64

would entail considering the broader context of the desing. This would include conversations about thermal contraction stresses, feasibility of fabrication, and how well boundary conditions can be maintained in the glue joints.

From the current perspective, thermal stresses due to differential contraction would be best absorbed by a structure that is flimsy in the radial direction, which would be the tabbed geometry. Considering this, the rolled sheet might not absorb thermal contraction well, but there are some mitigations that can be put in place such as cutting holes in the sheet. These are debates that can be had.

The manufacturing of a rolled sheet this thin is not feasible (the thickness is 0.01”) the

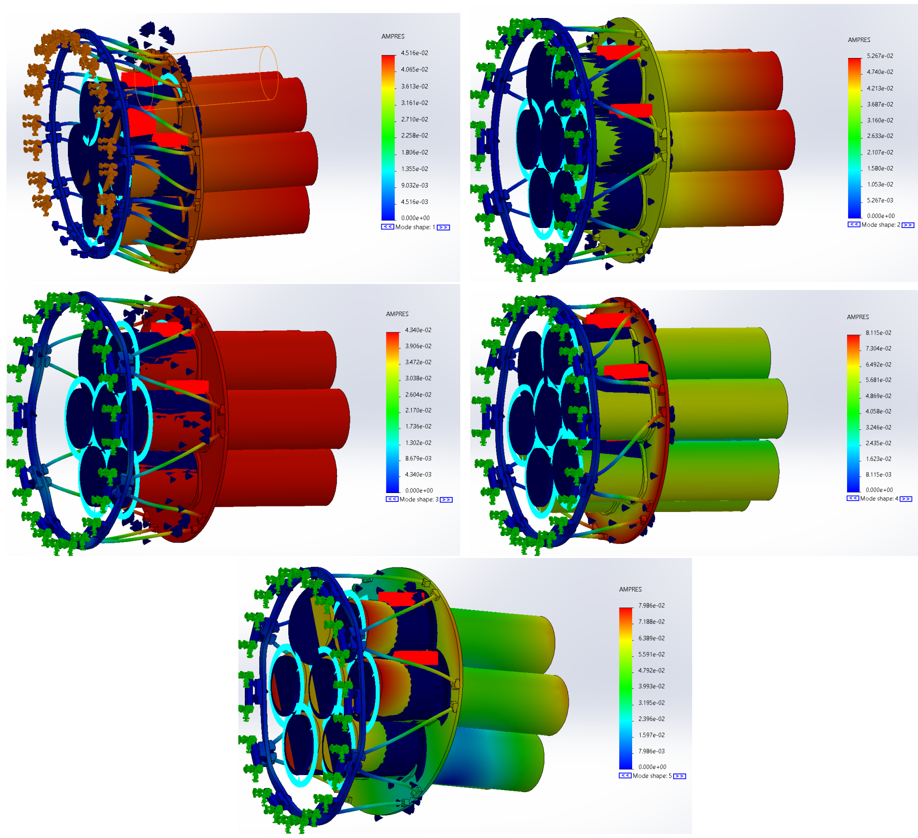

Figure 8: Mode Shapes indexed from left to right and top to bottom.

most that has been done given the fibrous material is about 50 mils thick. This is a strong argument against this design, which is unfortunate given the stiffness, but also the triangular bipod is not much worse.

The triangular bipod perhaps has the best ability to maintain the rigid boundary conditions in the glue joints. The glue hole can be tapped to increase the glued area, and a mechanical screw can be put in the foot of the anchor point to provide a mechanical connection between the G10 member and the anchor mount. The same cannot be said for the tabbed geometry, which has historically had issues with tabs pulling out.

Another conversation that will need to be had extends this analysis to ask what resonances actually matter, and whether it is imperative to avoid those in stiffening the structure or not. The two main issues being flexure of the optics tubes relative to the 300K vacuum plate and microphonic heating of the focal plane. The inertial forces of the telescope accelerating and moving could drive the flexure modes to oscillate the optics relative to the 300K plate. The question being how stiff does this structure need to be to mitigate this affect and also attempt to provide a requirement for what acceptable levels of deflection may be.

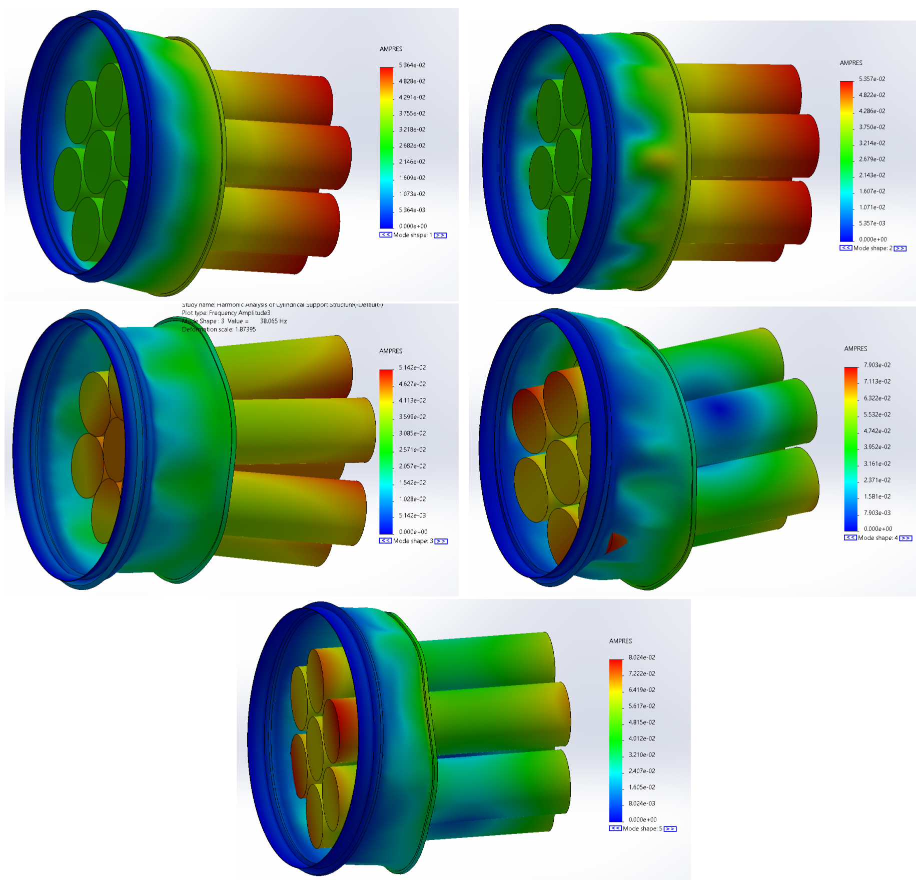

Figure 9: Mode Shapes indexed from left to right and top to bottom.