Conceptual Design Notes for Primary Metering Structures for the SNAP Telescope

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Abstract

This document discusses conceptual design options for primary structures of the main telescope assembly for the SuperNova Acceleration Probe. Structural concepts for the secondary mirror support structures an the primary mirror baffle are discussed, analysed, and compared.

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1. Definitions

SNAP: SuperNova Acceleration Probe
CyE: Cyanate Ester
GFRP: Graphite Fiber Reinforced Plastic
CTE: Coefficient of (linear) Thermal Expansion
HST: Hubble Space Telescope
g: acceleration of gravity, $9.81 m/s^2$
FE(M): Finite Element (Model)
$\mu\varepsilon$: micro-strain, a strain of 1×10^{-6} m/m

2. Assumed Telescope Concept and Key Mechanical Parameters

This section summarizes the assumptions made for the purpose of this initial mechanical design study[3].

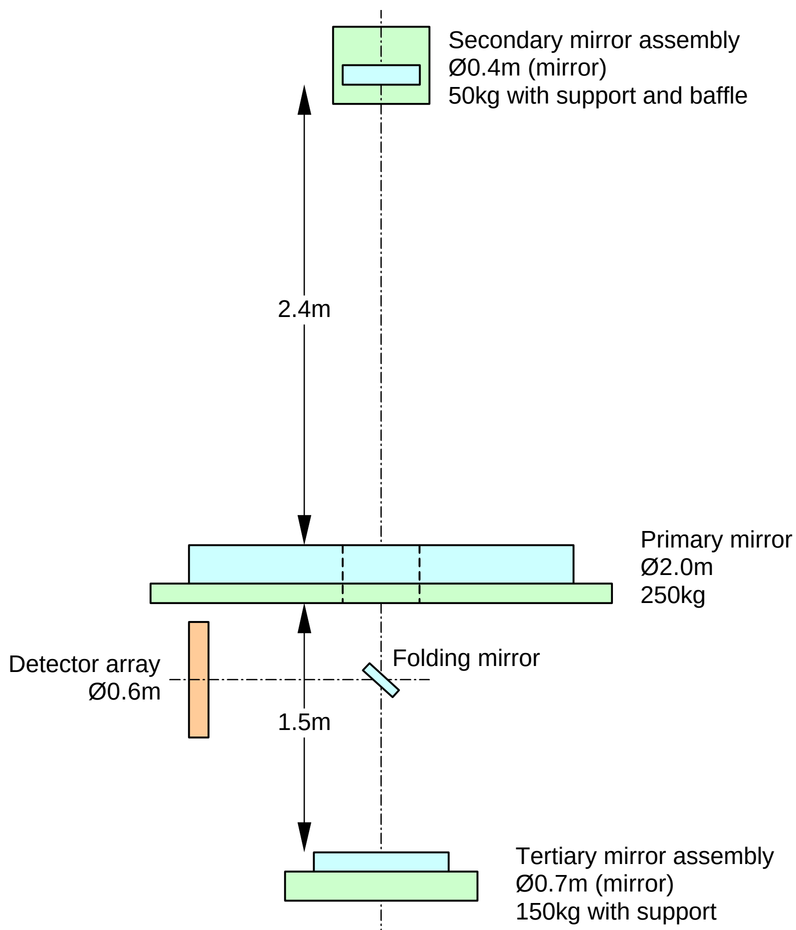

Figure 1: Key geometric and mass parameters of SNAP as assumed in this study.

The telescope is concept is shown in Figure 1. It is composed of five key elements: the primary, secondary, and tertiary focusing mirrors, the folding mirror, and the focal plane detector array. Figure 1 lists assumed masses for some of those elements. In particular, the mass of the secondary mirror assembly (including a support base, a baffle, any positioning actuators, etc.) is assumed to be 50 kg. For dynamic modeling purposes, those 50 kg are assumed evenly distributed in a cylindrical volume with a diameter of 0.5m and a height of 0.3m; the secondary mirror surface is assumed to be located 0.1m from the bottom of that volume, and 2.4m from the primary mirror surface.

The general mechanical support concept consists of a stiff base structure supporting the primary mirror and directly attached to the top of the spacecraft by some structure (likely a truss). The secondary and tertiary mirror assemblies and the focal plane detector array assembly are all supported by structures attached to the primary mirror strongback. This insures direct load and alignment paths from the primary mirror to the other key optical components of the telescope.

3. Assumed Mechanical and Other Design Requirements

3.1 Mechanical loading and stiffness

A Delta IV Medium launch vehicle has been assumed as the baseline for the SNAP mission[1]. Mechanical loading conditions and other mechanical design requirements for Delta IV payloads are defined in the *Delta IV Payload Planners Guide*[2].

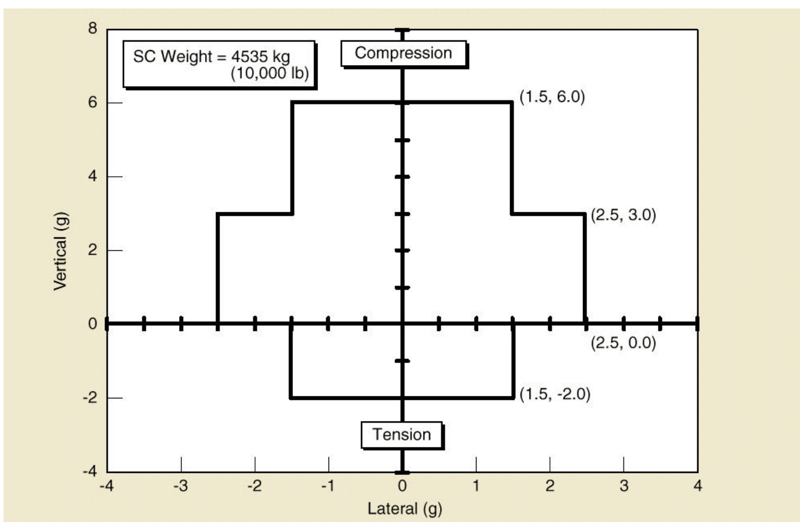

Figure 2: Design limit load factors for Delta IV medium launch vehicle[2].

For Initial conceptual studies, we reduce those conditions to a simple set of dynamic stiffness requirements and pseudo-static design limit load factors. Later in the program, stiffness requirements and load factors can be better defined based on coupled loads analysis of the entire launch vehicle / spacecraft assembly. However, before such analysis can be performed, the following conservative numbers can be used[2]:

- Fixed boundary (at the separation plane) fundamental frequencies of the entire spacecraft / instrument assembly must be greater than 27 Hz in the launch direction and 10 Hz in the transverse directions.
- All structural elements of the payload must have natural frequencies greater than 35 Hz.
- The design limit load factors for the spacecraft / instrument assembly are defined in Fig. 2. For this initial study, two distinct load cases were selected:
    1. accelerations of 1.5g transverse and 6g axial, simultaneously.
    2. accelerations of 2.5g transverse and 3g axial, simultaneously.

3.2 Thermo-elastic Dimensional Stability

Because SNAP is a high resolution telescope, its geometric and dimensional stability are critical to performance. Temperature variations in orbit and temperature difference between initial alignment and on-orbit conditions will tend to disturb alignement of optical components thorugh thermal expansion. At least 3 distinct approaches can be used (alone or in combinations) to minimize these effects:

- Minimize temperature variations and ground-space temperature differences through the use of heaters and active thermal control.
- Provide means of actively controlling the geometry of the telescope in orbit through mirror positioning and/or reshaping actuators.
- Design the telescope structures to minimize temperature-induced deformations through the use of near-zero CTE materials.

At the time of this study, a strategy had not been defined for SNAP. Operating temperature and temperature variation ranges were not defined. We made the assumption that all metering structures must be designed to near-zero CTE.

4. Designing Tubular Composite Struts for Delta IV Launch Conditions and Near-Zero CTE

4.1 Design for Stiffness

Long struts used in support trusses for space must be rigid enough to avoid low-frequency “violin” modes. As stated above, for a Delta IV mission, structural elements should be designed such that their fundamental frequency is above 35 Hz.

Since low mass, high stiffness and extreme dimenional stability are required, tubular struts made of graphite fiber reinforced composites are considered. For a thin-walled tubular strut, the fundamental frequency f_1 with pinned-pinned (p-p) and fixed-fixed (f-f) end conditions reduce to:

$$f_1^{\,p-p} = \frac{\pi}{2L^2}\sqrt{\frac{EI}{\rho A}} \approx \frac{\pi}{4L^2}\sqrt{\frac{E}{2\rho}}D, \qquad (1)$$

$$f_1^{f-f} = \frac{3.56}{L^2}\sqrt{\frac{EI}{\rho A}} \approx \frac{1.78}{L^2}\sqrt{\frac{E}{2\rho}}\,D. \qquad (2)$$

Eq. (1) and (2) show that the fundamental frequency is only a function of the length of the strut (*L*), the material (E/ρ), and the mean diameter (*D*) of the tube. It is not a function of the wall thickness (*t*) because both mass and bending stiffness vary linearly with *t*. Given a minimum frequency requirement and a material, these equations provide a minimum diameter requirement for a tubular strut.

4.2 High Stiffness, Near-Zero CTE Material

Graphite fiber reinforced composites offer high stiffness and low CTE. However, to maximize stiffness, tubular struts are built with mostly longitudinal fiber orientations. This leads to slightly negative longitudinal CTE, typically around -1×10^{-6}/ºC. For a 2.4m long secondary mirror metering structure, and assuming an on-orbit temperature around 100ºK, this would cause length changes from assembly at room temperature (295ºK) of the order of 470µm.

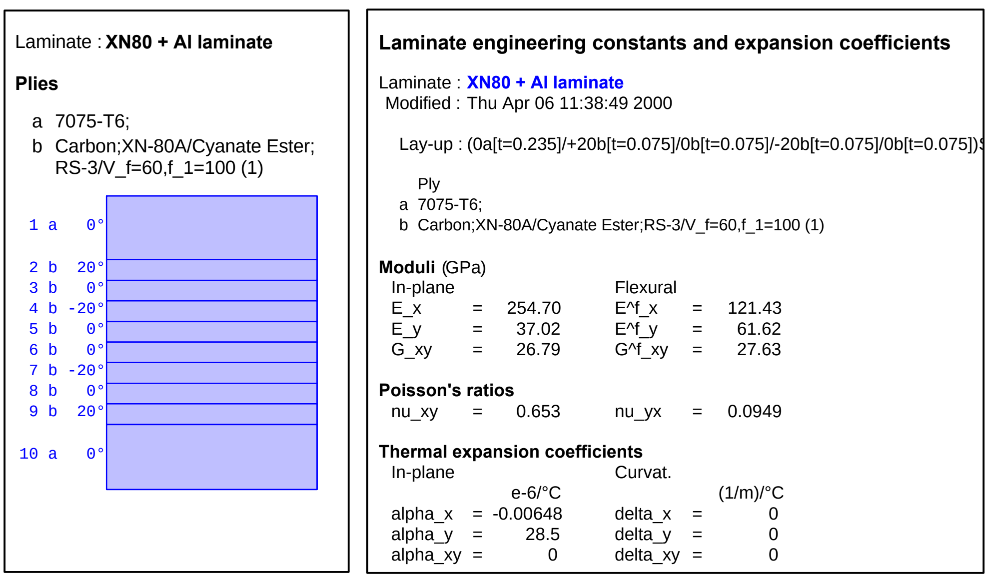

Figure 3: Example of graphite fiber/resin/aluminum composite with near-zero CTE; the 8 inner layers are XN80A/CyE 75µm prepreg, and the outer layers are 235µm aluminum each.

One approach to making near-zero CTE, high stiffness composite tubes is to include in the composite layup some amount of positive CTE material to balance the negative CTE of the fibers. An example is illustrated in Figure 3. It is a 1.07mm thick laminate made from 8 layers of Cyanate-ester resin/XN80 graphite fiber composite at shallow angles to the tube axis (0 and 20 degrees), and 2 layers of aluminum foil. The total thickness of the aluminum layers is tuned to achieve 0 CTE of the composite, as shown in the figure.

It should be noted that very high modulus fibers such as XN80 have very negative CTE (-1.6×10^{-6}/ºC) and therefore require a significant amount of positive CTE material to compensate. However, the effective modulus of the resulting laminate is quite high (255 GPa). In practice, the large amount of aluminum required in this example would likely have to be distributed through the thickness as several thin layers to keep temperature induced interlaminar shear stresses reasonable.

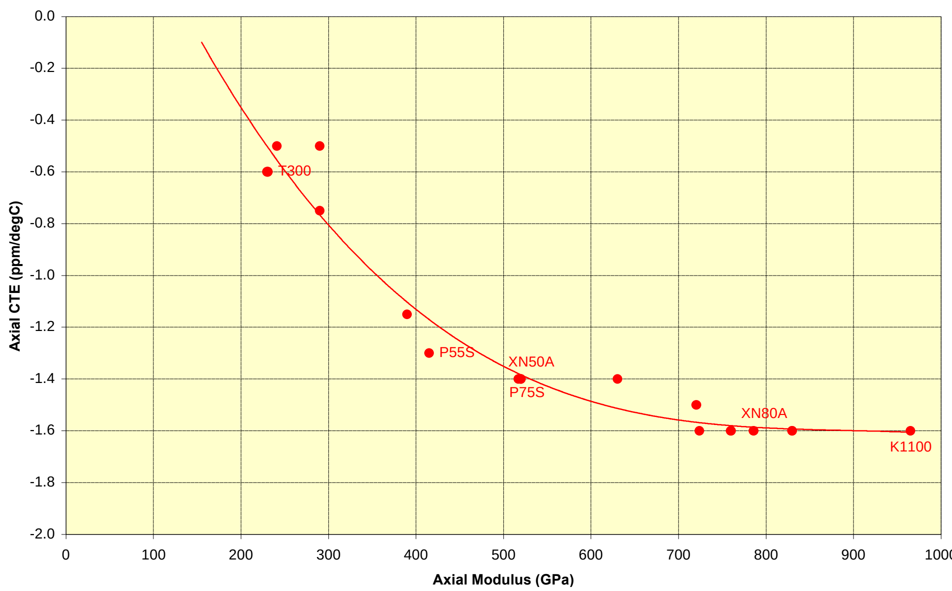

Figure 4: Modulus-CTE correlation in high modulus graphite fibers.

Near zero CTE could also be achieved with lower modulus, less negative CTE fibers (Fig. 4) together with smaller relative amounts of positive CTE material. The resulting composite modulus would however be lower (around 190 GPa using fibers like XN50, P75, or M55J for example).

The rest of this study assumes that a zero-CTE fiber/resin/aluminum composite layup is available to make tubular struts and that the modulus of that composite in the axial direction of the strut is 250 GPa. The density of that composite was assumed to be 1800 kg/m^3.

End fittings would be required to connect the various struts into trusses and frames; those fittings could be made of Invar or Titanium, depending on the severety of the dimensional stability and mass requirements.

5. Conceptual Designs for Secondary Mirror Support Structure

5.1 Hubble-like Cylindrical Metering Truss/Frame

This concept is essentially identical to that used for the HST metering truss. It consists of a two-bay (HST used 3 bays) cyclindrical truss structure, with 16 struts per bay, circular stiffening rings (3) and a 4-bar "spider" structure attaching the secondary mirror assembly to the upper ring. Figure 5 illustrates that concept, with members sized for this application.

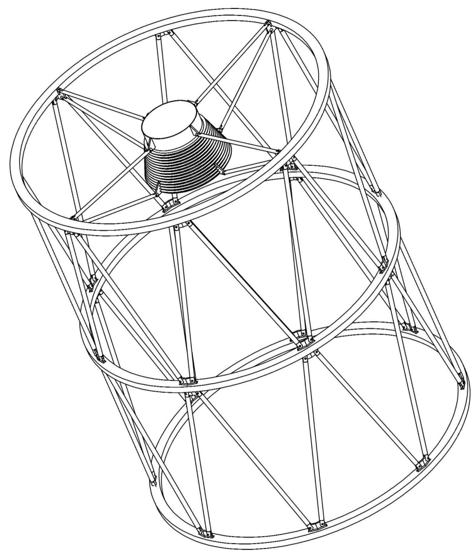

Figure 5: Hubble-like cylindrical metering structure concept, sized for SNAP.

For initial sizing, the components of that structure were grouped into 3 sets of cross-sections: all 32 bay members, the 3 rings, and the 8 "spider" beams. For simplicity in initial sizing, all 3 cross sections were arbitrarily given a 1mm wall thickness and assumed circular (in practice, the stiffening rings would likely not have a circular cross section).

	material	Density [kg/m^3]	Modulus (axial) [Gpa]	Wall [mm]	Diameter [mm]	
					Inner	Outer
Truss members	Al/GF/resin Composite	1800	250	1	30	32
Rings	Al/GF/resin Composite	1800	250	1	68	70
Spider Beams	Al/GF/resin Composite	1800	250	1	22	24

Table 1. initial sizing of Hubble style cylindrical truss/frame.

Minimum diameter requirements were calculated for fixed-fixed end conditions (the spider support structure is not a truss : it relies on bending for stability). In this case, because the individual struts are relatively short, those minimum diameters are smaller than practical (few mm). Initial diameters were then selected based on intuition and a FEM of the structure was constructed to evaluate natural frequencies. Using this model, the 3 diameters were then tuned to minimize mass while achieving a 35 Hz minimum fundamental frequency. The resulting cross sections are shown in Table 1.

Figure 6 shows the first few natural modes of the optimized structure; the first 3 modes are between 37.2 and 37.3 Hz.

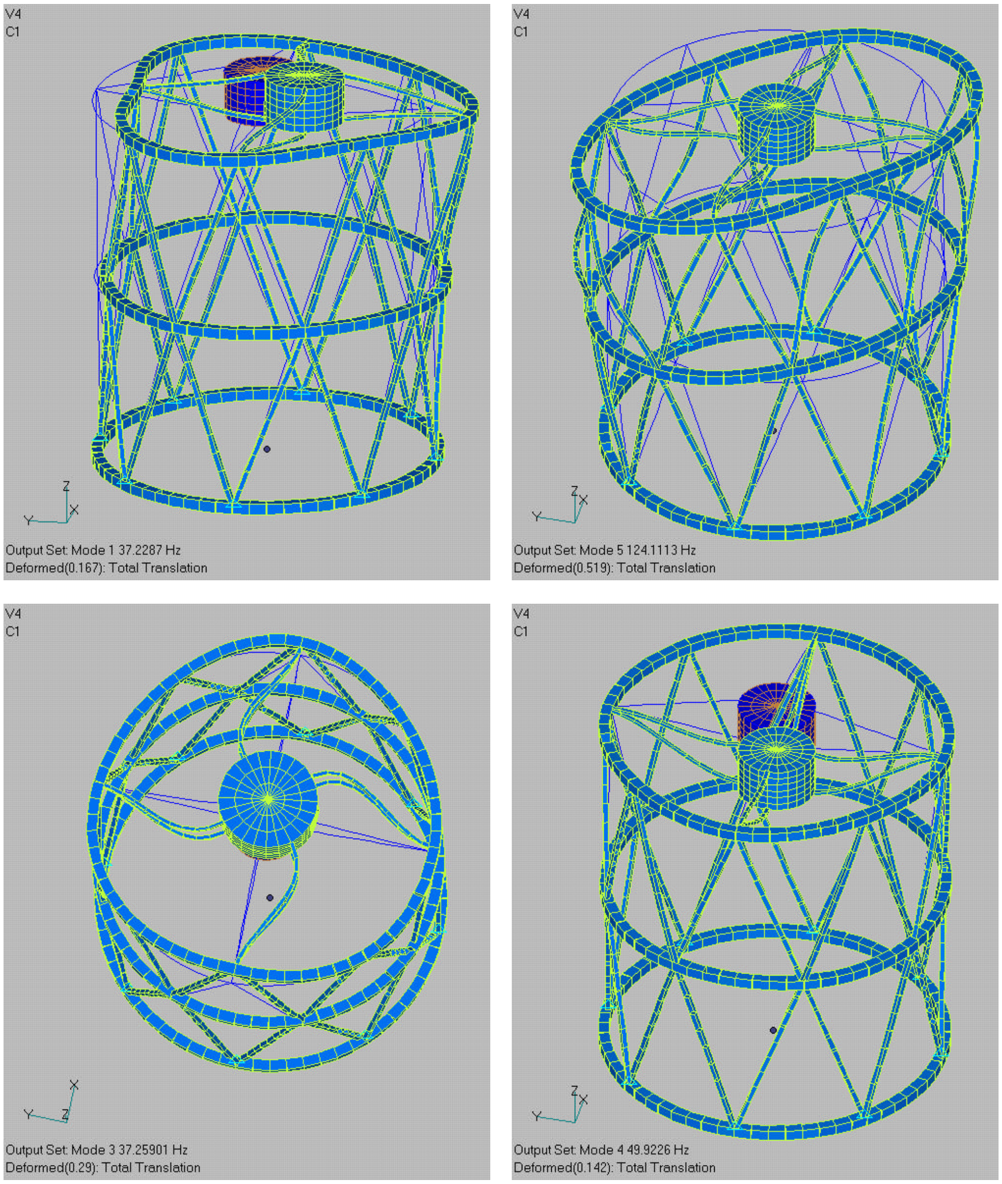

Figure 6: First few fundamental vibration modes of Hubble-like barrel metering truss optimized for minimum mass and 35Hz fundamental frequency.

5.2 FORTE-like Flat Panel Shell

This concept is based on using a low cost composite structure fabrication technology developed by LANL and Composite Optics, similar to what was used for the FORTE satelleite bus. The approach is based on flat sandwich panels built with flat panel composite faces and joined by flat panel composite ribs. This approach is cost effective as it does not require complex molded components. The faces and ribs are simply cut out of graphite fiber reinforced flat panels. They incorporate “snap” together groove and tab features for self-jigging and are joined together by bonding with room temperature cure adhesives. An example of such structure, dimensioned for the SNAP telescope is shown in Figure 7.

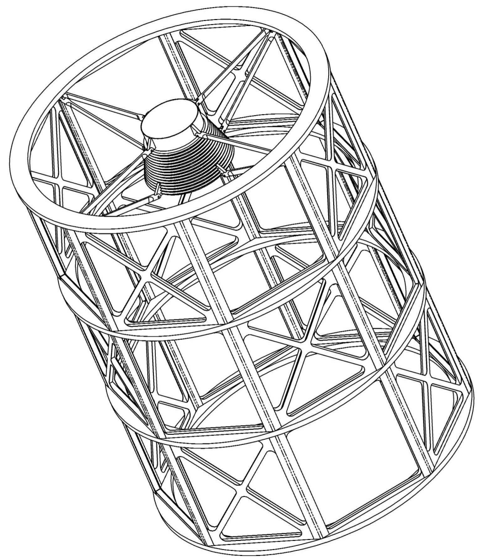

Figure 7: Flat panel metering structure concept.

We have not at this time performed any analysis or sizing calculations on this approach. The minimum total mass of composite material using this approach can be roughly estimated by assuming a panel thickness of 0.75mm for example (thinner panels would likely not be practical). Built with 0.75mm thick composite panels, the structure of Fig. 7 contains about 69kg of composite material. Depending on the details of the assembly technique, a number of metal connection pieces may be required (Forte used a large number of aluminum joints).

5.3 Tripod Frame

A simple support concept often shown in sketches and sometimes used in smaller ttelscopes consists of a tripod connecting the secondary mirror assembly to the outer periphery of the primary mirror strongback. It is important to realize that such tripod system does not constitute a truss and relies on the bending stiffness of the beams and stiff build-in end conditions to provide high stability and stiffness. Because of this, a tripod suppoort is likely to require very large diameter beams to achieve the 35 Hz requirement.

First, we calculated the minimum diameter of the beams to avoid low frequency (<35Hz) violin modes: using Eq. (2), the minimum mean diameter is found to be 19mm.

$$L \approx 2.8m, E = 250GPa, \rho = 1800kg/m^3 \rightarrow f_1^{f-f}(Hz) = 1866 \times D(m) \qquad (3)$$

$$f_{\min} = 35Hz \rightarrow D_{\min} = 19mm \qquad (4)$$

An FE model was constructed assuming 20mm OD and 18mm ID of the 250Gpa hypothetical composite layup mentioned in Section 4.2. Characteristic modes of that structure are illustrated in Fig. 8. The "violin" mode of the legs is about 35 Hz as expected (right side of Fig.), but the lowest mode, involving transverse and pitching motions of the secondary mirror, occurs at about 1.8 Hz.

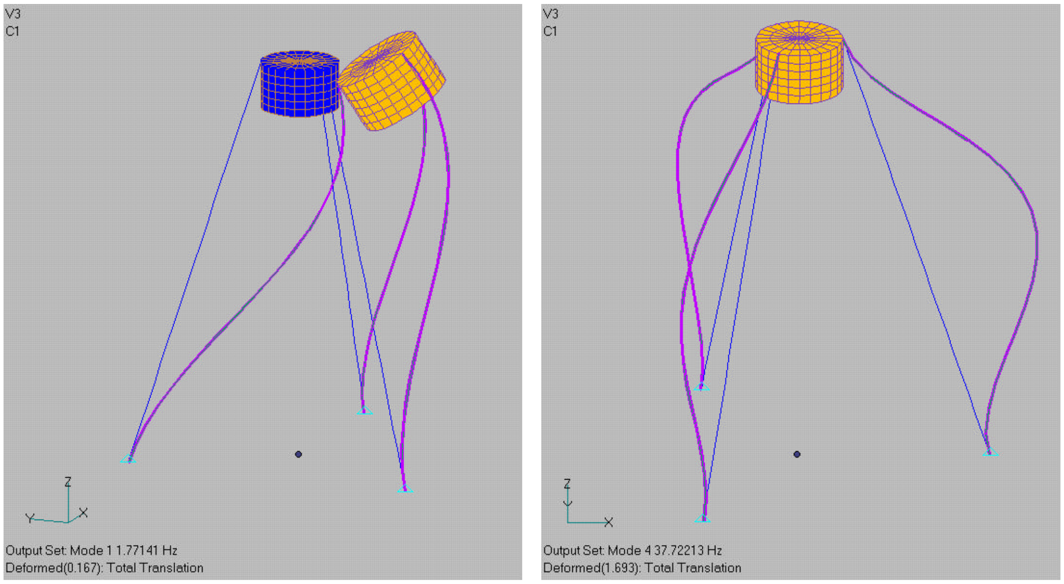

Figure 8: fundamental modes of tripod design with 20mm OD x 1mm wall GFRP legs.

The figure also clearly shows that the frequency of that first mode is dictated by the bending stiffness of the legs. Equation (5) shows relationship between the first mode frequency and the bending stiffness (or diameter) of the legs (still assuming a 1mm wall). Using this relation, we expect the diameter required to bring the first modal frequency to 35 Hz to be 145mm.

$$f(\mathrm{mod}\,e1) \div \sqrt{EI} \div D^{1.5} t^{0.5} \tag{5}$$

$$f(\mathrm{mod}\,e1) : 1.8Hz \rightarrow 35Hz \Rightarrow D : 19mm \rightarrow 145mm \tag{6}$$

The FE model was then updated to represent 145mm diameter legs; the resulting modes are illustrated in Fig. 9. The first mode now satisfies the 35 Hz requirement exactly, and the first violin mode of the legs is 286 Hz as could also be calculated with Eq. (2).

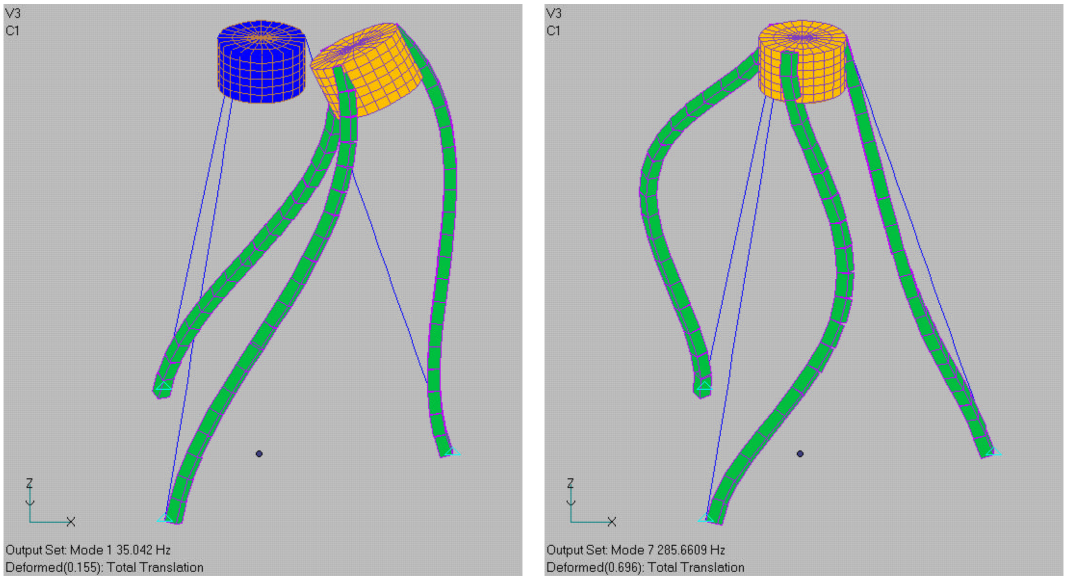

Figure 9: fundamental modes of tripod design with 145mm OD x 1mm wall GFRP legs.

Clearly, the tripod concept is far from optimal for a stiffness driven structure such as this. The reliance on bending stiffness of the legs forces one to use large diameter posts which block a significant portion of the aperture.

5.4 Hexapod

The optimal (i.e. highest stiffness and lowest mass) structural approach to supporting the secondary mirror directly from the periphery of the primary mirror strongback is to use a statically determinate truss (6 struts, stabilizing the 6 d.o.f. of the mirror). Such minimum truss is illustrated in Fig. 10 and reacts the inertial forces on the mirror assembly through pure traction and compression in the legs.

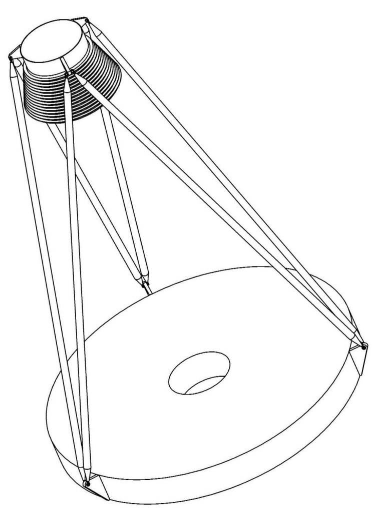

Figure 10: Hexapod metering truss concept, sized for SNAP.

With this concept, the bending stiffness of the struts need only be high enough to avoid violin modes below 35 Hz. With a length of about 2.8 meters and pinned ends, Eq. (1) gives a minimum mean diameter of about 43 mm:

$$L = 2.8m, E = 250GPa, \rho = 1800kg/m^3 \rightarrow f_1^{p-p}(Hz) = 823 \times D(m) \qquad (3)$$

$$f_{\min} = 35Hz \rightarrow D_{\min} = 43mm \qquad (4)$$

A FE model was constructed representing tubular members with an outside diameter of 45mm and a 1mm wall. The modes obtained from that model are shown in Fig. 11. The lowest frequency is the violin mode of the legs slightly above 35 Hz as designed. The lowest mode involing significant mirror motion occurs at 43.5Hz. Note that the curvature of the legs seen in that mode is due to dynamic coupling with the 35 Hz violin mode of the legs – there is no bending moment transfer between the mirror assemblies and the legs.

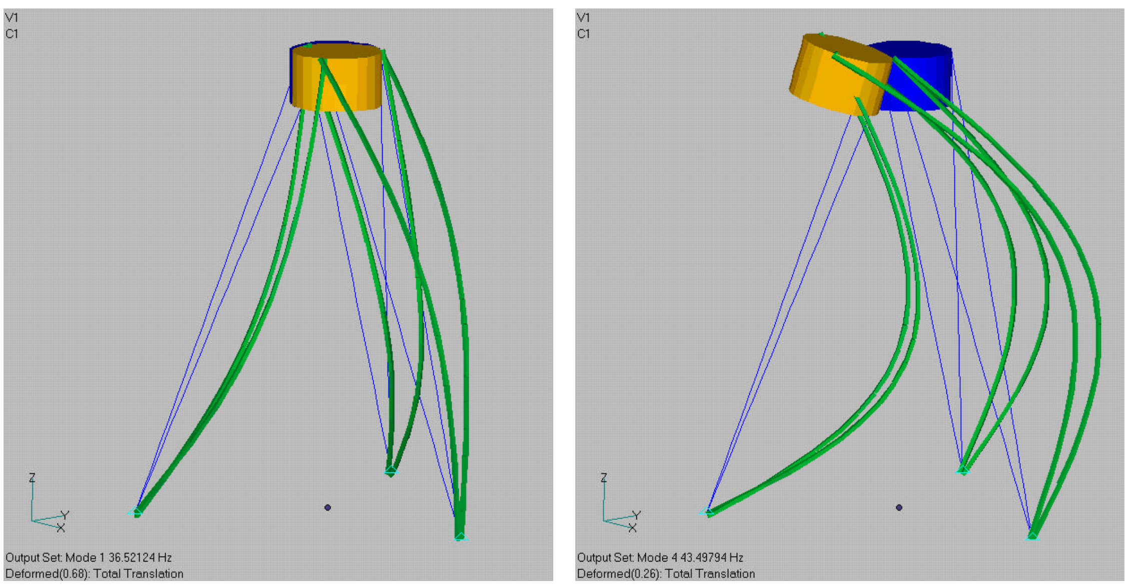

Figure 11: Fundamental vibration modes of hexapod structure built with circular thin wall GFRP tubes (45mm OD by 1mm wall, E_{axial}=250GPa)

The model was also used to verify the underlying assumption that these design are entirely stiffness driven (i.e. stresses will automatically be very low as soon as the design statisfies stiffness requirements). Figure 12 shows static load solutions for load cases 1 and 2 as defiend in Section 3.2. The stresses/strains do not exceed 12.2 MPa / 50με in either case. The ultimate strain for XN80 uniaxial composites is about 5000με and the yield strain for 5052 aluminum alloy for example is around 2000με. This gives safety factors greater than 40 to ultimate strains for pseudo-static design limit load factors.

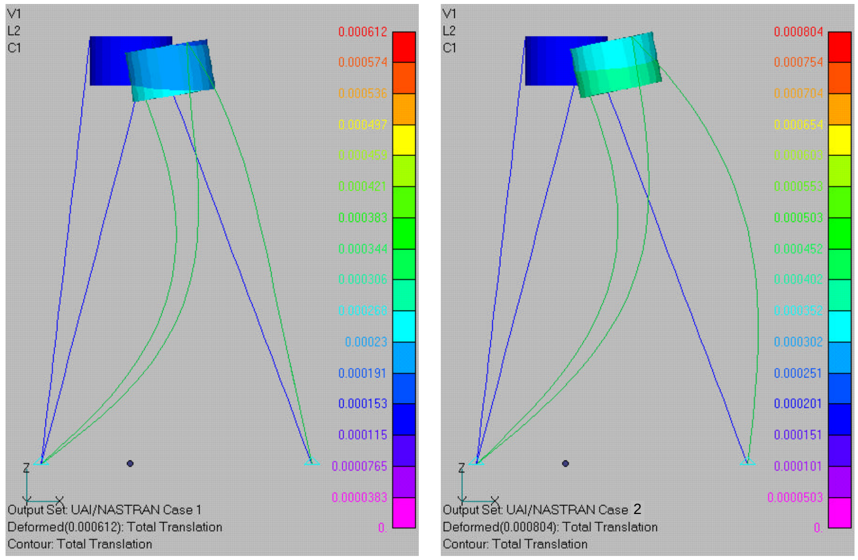

Figure 12: Static deflections of hexapod structure under acceleration limit load cases 1 and 2.

5.5 Summary Table

Table 2 summarizes mass estimates for the four concepts presented above. Note that in all cases, the estimated mass of metal fittings (Invar was assumed) exceeds the mass of composite materials. The table also lists the % reduction in unobstructed light collection area of the primary mirror.

Concept	Mass (kg)			% obscuration of primary mirror area
	composites	fittings	TOTAL	
Hubble style truss	16	42	58	3
FORTE style frame	70	45	115	3
Tripod frame	7	24	31	11
Hexapod truss	4	8	12	7

Table 2. summary of approximate masses and obscurations for secondary mirror assembly support concepts.

6. Primary Mirror Baffle

The primary mirror baffle restricts light input into the telescope to the field of view of the instrument. It is a large cylindrical shell with a number of ring-shaped baffles on the inside. The entire inside surface is coated with optical black paint to minimize reflection of stray photons.

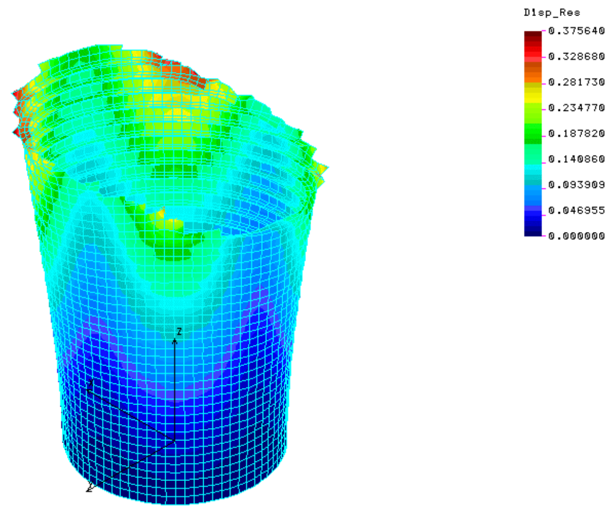

Figure 13: fundamental vibration mode of a primary baffle structure for SNAP made entirely of aluminum alloy; thickness of main shell and baffle rings is 1.5mm; frequency is 56 Hz; total mass is 124 kg.

We analysed a zero'th order concept for this baffle to evaluate the required thickness (and mass) of the shell while maintaining a minimum fundmental frequency of 35 Hz. For simplicity, we assumed a baffle structure made of aluminum plate, with the optical baffle rings also serving as mechanical stiffeners. A model of a baffle with 8cm wide baffles/stiffeners every 17cm was constructed. Wall thickness was set to 1.5 mm throughout. The lowest natural frequency is about 56 Hz and the total mass about 124 kg. Figure 13 illustrates the mode shape obtained with this model.

7. References

1. Supernova / Acceleration Probe (SNAP), A Science Proposal to the Department of Energy and the National Science Foundation, Version 1.00, SNAP Collaboration, February 8, 2000.
2. *Delta IV Payload Planners Guide,* doc. # MDC99H0065, The Boeing Company, Huntington Beach, CA, October 1999.
3. Private Fax Communication from M. Levi, LBNL, February 16, 2000.