DESIGN, OPTIMIZATION, AND FABRICATION OF A COMPOSITE GRID-STIFFENED AVIONICS ENCLOSURE

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SUMMARY

Using commercial-of-the-shelf components has many advantages for satellite applications; however, there are a number of challenges, including high vacuum compatibility, that must be addressed. To eliminate the need for high vacuum components, a pressurized, grid stiffened avionics enclosure was designed, optimized, and fabricated.

Keywords: spacecraft, avionics enclosure, grid-stiffened structures, composite fabrication

INTRODUCTION

Historically, satellite missions are expensive and labor-intensive because every system is custom-built and highly optimized. As a result, new technologies are difficult to implement and systems are often obsolete even before they are launched. One approach to address this problem is the utilization of highly capable small satellites that incorporate commercial off-the-shelf (COTS) components. The advantages of using COTS components are significant reductions in cost, a large industrial base from which to draw, and a larger supply chain to reduce component procurement times.

This approach has an added benefit from a thermal control standpoint because traditional terrestrial cooling systems for electronics can be utilized. In place of highly customized thermal control approaches currently used on satellites, simple heat sink and cooling fan approaches can be used. Additionally, cooling fans provide a means to actively control the heat transfer rates within the system, thereby providing a means to decouple the heat source from the sink during the cold orbit conditions. The result is a reduction in survival heater power and ultimately power system and battery mass. This approach is ideal for low-cost, experimental system where mission lifetime is limited to months. This approach is applicable to longer duration missions as well, but fan reliability becomes an issue that must be addressed either through improved, i.e. more expensive, systems or redundancy, which translates to added mass.

There are a number of advantages with this approach; however, there are a number of challenges to using COTS components on satellites as well. Most terrestrial components are not designed to withstand the harsh space environment. Environmental effects including high launch loads, high energy particles, atomic oxygen, ultraviolet radiation, and high vacuum, among others, complicates the use of COTS equipment. High vacuum poses a particular problem because of issues related to outgassing, thermal interfaces, whiskering, and the lack of air for convection cooling. COTS components can easily be ruggedized to survive launch, but surviving a high vacuum environment is another matter. To address this issue, a new design methodology based on pressurized, grid-stiffened avionics enclosures was developed.

PRELIMINARY ENCLOSURE DESIGN

Using sealed enclosures and forced air convection for cooling is not a completely new concept. It has been used before to cool electronics at high altitudes where there is insufficient air for adequate cooling [1]. The difficulties with space applications are the high vacuum condition the enclosure must endure coupled with the ever-present requirement to minimize mass. In addition, the enclosure must be able to survive high vibration, acoustic, and shock loads during launch. For these reasons, an alternate design to traditional electronics enclosures was pursued.

Traditional, avionics enclosures consist of either metallic or composite boxes to which the components are mounted. These boxes are optimized to survive high launch loads, to provide adequate thermal management, to minimize deflection, and to provide dimensional stability for the components. Obviously, they are not designed to endure high internal pressures. Adding an internal pressure requirement to traditional requirements would increase the wall thickness and mass to the point where the design would be either undesirable or infeasible. An alternative to this approach is to split the traditional enclosure requirements and the internal pressure requirement between two different enclosures so that each can be optimized to these drastically different sets of requirements. The first component is the traditional electronics enclosure, and the second is an optimized pressure shroud. This configuration is shown in Figure 1. An added benefit of this approach is its compatibility with legacy designs.

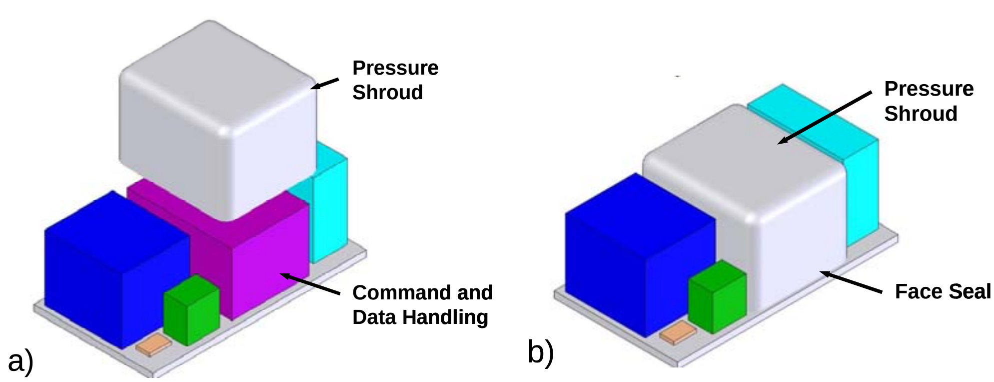

Figure 1: Depiction of the pressure shroud for the command & data handling subsystem a) exploded view and b) assembly view

One of the challenges with this approach relates to geometric considerations. Typically, satellite components and electronics are planar and are best enclosed in rectangular prisms. However, the ideal geometries for pressure vessels are spheres and cylinders. It is unlikely that satellite components would be redesigned for optimal packaging inside a sphere or cylinder, this it will be required for the pressure shroud to contain the pressure, must do so with a packing approach compatible with current satellite components and legacy systems. A modest improvement can be made by utilizing rounded corners, but this must be balanced with efficient volumetric packaging of the satellite components.

Design Requirements

As with all satellite structures, the driving design requirement is to maximize the stiffness-to-weight ratio for the enclosure. However, since the only requirement for the shroud is to resist the internal pressure, the structural and stability requirements are significantly reduced when compared to an enclosure that must both support the components and resist the pressure. As a result, the allowable maximum deflection at the center of the panel was relaxed. A design acceptance value of 0.127 cm was chosen.

The internal pressure of the enclosure is dependent on the thermal performance requirements of the system. By increasing the pressure in the box, the effective heat transfer coefficient can also be increased. For this application, we assumed that standard atmospheric pressure would provide adequate cooling. The design pressure was defined as 10% above atmospheric to ensure that the enclosure was slightly pressurized under all cases.

Since the shroud will be pressurized, the design must comply with MIL-STD-1522A, "Safe Design of Pressurized Space Systems." This standard requires a proof pressure of 1.5 times the maximum expected operating pressure (MEOP) and a design burst pressure of 2 times the MEOP without rupture. Since the shroud will be designed to contain a pressure 112 kPA at standard temperature, the MEOP is 121 kPa because of the temperature rise from 298 K to 323 K. The temperature rise results from the difference between standard temperature and the allowable operating temperature for most avionics. The requirements for the shroud are summarized below.

1. Component size 28.58 cm (W) x 33.66 cm (L) x 21.59 cm (H)
2. Fit within an envelope extending 2.5 cm from all sides of the component
3. Maximum transverse deflection less than 0.127 cm at 120 kPa (MEOP)
4. Withstand a proof pressure of 181 kPa without detrimental deformation
5. Withstand a burst pressure of 240 kPa without rupture
6. Use only space qualified materials
7. Withstand a 15g random vibration launch environment in all axes
8. Mount to a flat plate
9. Provide a hermetic seal with a leak rate of less than 10% loss per year
10. On-orbit lifetime of three years

The component size selected is representative of a moderate to large electronics or avionics box. Because the primary load case was internal pressure, components with large planar surfaces pose the greatest challenge and were the best choices to stress the design and test the limits of utility for this approach.

As a first step, four different potential enclosure structural configurations were considered including welded-metallic, laminated composite, honeycomb sandwich, and grid-stiffened composite enclosures. Each concept has its advantages and disadvantages, which are summarized on Table 1.

Table 1: Advantages and Disadvantages of Pressure Shroud Fabrication Approaches

Fabrication Approach	*Advantages*	*Disadvantages*
Welded metal	• Low-cost • Simple to design, analyze & fab • Straightforward face seal design	• Very heavy • Hermetic welds
Laminated composite	• Improved stiffness-to-weight • Straightforward to analyze • Extensive fabrication experience	• Heavy • Hand lay-up or bonding required to seal at edges
Sandwich structure	• Excellent stiffness-to-weight • Minimal weight and cost	• Heavy hermetic joints • Moisture entrapment • Complicated analysis
Grid-stiffened laminated composite	• Good stiffness-to-mass • Monolithic structure for sealing • Structurally efficient	• Complex design & analysis • Complex load path • Hand lay-up

Methods of Analysis

Because of the number of potential configurations for the shroud and the complicated techniques required for detailed analysis and optimization, a preliminary down selection was performed using first principles and first-order design tools. The purpose of the analysis was to determine which configuration provided the best stiffness-to-weight ratio starting with a simple design that could then be further optimized through detailed analysis. The analysis methods for the preliminary down-selection, including initial assumptions, are discussed below.

For the preliminary down selection, the maximum deflection of the largest panel for each of the fabrication approaches was evaluated. It was assumed that the edge boundary conditions were most closely represented by a fully clamped (CCCC) condition because of the symmetry of the box and the pressure load. The true conditions are not fully clamped; however, assuming simply supported conditions yielded unreasonably high transverse deflections. The load applied to the panel was 18.4 psi, which represents the MEOP with 5% margin for temperature variations.

The metallic structures were analyzed using the finite element analysis program CosmosWorks. For the laminated composite (LC) panel, it is difficult to develop a Finite Element Analysis (FEA) model quickly for many of the cases. Often it is more efficient to use non-FEA predictive models for approximation and sizing. For any of the configurations utilizing a laminate construction, CompositePro® was used to determine the laminate properties based on the lamina properties. CompositePro® was also used to determine the maximum deflection at the center of the panel for a CCCC boundary condition using Whitney's one term approximate solution (m=n=1) [2]. However, it does not provide any moment resultants (M_x, M_y, M_z), which are required to size the face-sheets. For this aspect of the analysis, CosmosWorks was used.

To analyze the sandwich structure (SS) panel, the face-sheet and honeycomb core were input as a lamina into CompositePro®, which was then used to analyze the sandwich laminate with the [face-sheet/core/face-sheet] construction and to determine the full sandwich laminate bending stiffness matrix D_{ij}. The laminate D_{ij} terms were then input into the approximate laminated specially orthotropic plate bending solution and the maximum transverse displacement of the panel and the moment resultants were determined. Lastly, CompositePro® was used to find the minimum safety factor based on First Ply Failure Criteria and the Maximum Stress Failure Criteria.

Finally, for the grid-stiffened laminated composite (GSLC) enclosure, a rectangular grid spacing of 5" by 5" sections was used, and the face-sheet thickness was varied. Because of the rib stiffening effect, the critical panel size evaluated was a 5" by 5" section only. This amounts to a face-sheet sizing analysis, and it is assumed that the rib structure takes the majority of the load. An analysis method similar to that used for the laminated panel was used for the grid stiffened panel. For this simple preliminary analysis, there was no attempt made to size the rib thickness, width, deflection, or stresses. The analysis would have required detailed FEA modeling.

Results and Discussion

Using the methods outlined above, each configuration was analyzed. For some of the configurations, variations from the baseline were also evaluated. Most notably these were for the grid-stiffened, carbon-epoxy, laminated composite construction where different face-sheet thicknesses were evaluated and for the monolithic aluminum panels where two different thicknesses were evaluated. The results from the analysis are presented on Table 2.

Table 2: Summary of Preliminary Analyses

Case	*Construction Type - Material*	*Height [cm]*	*Mass [kg]*	*Deflection [cm]*	*Meets Req's*
A	Welded metal – Al 6061-T6	0.508	4.21	0.160	No
B	Welded metal – Al 6061-T6	0.635	5.26	0.081	Yes
C	Welded metal – 1025 Steel	0.381	9.07	0.132	No
D	LC – 44 uniply layers, $[45/-45/(0/90)_5s]$	0.559	3.81	0.193	No
E	SS – Al w/ 1.016 cm honeycomb core	1.321	3.86	0.013	Yes
F	GSLC – 10 uniply layers, $[0_3/90/0]s$	0.127	0.98	0.406	No
G	GSLC – 16 uniply layers, $[0_3/90/0]s$	0.203	1.59	0.102	Yes
H	GSLC – 20 uniply layers, $[0_3/90/0]s$	0.254	1.95	0.053	Yes
I	GSLC – 44 uniply layers, $[0_3/90/0]s$	0.559	4.04	0.018	Yes

Of the different design configurations, the best performance was achieved with the aluminum honeycomb sandwich panel configuration, which weighed 3.86 kg and deflected 0.013 cm. However, after careful consideration of all aspects of the requirements including manufacturing, integration, and sealing, the aluminum honeycomb panel was eliminated because it would be nearly impossible to carry the loads at the corners and to provide a hermetic seal without significantly increasing the mass. Instead, the grid stiffened approach was chosen because of its good SWR and the ability to fabricate the structure as a single piece with radius corners, thereby reducing the stress and providing a good seal. The mass for this approach was estimated at 1.59 kg, and the deflection was 0.102 cm.

DETAILED ENCLOSURE DESIGN AND OPTIMIZATION

The objective of the detailed design effort was to improve the fidelity of the analysis of the grid-stiffened enclosure so that a proper shroud design could be completed as opposed to the simple analysis performed in the preliminary effort. The detailed design of the shroud focused on the optimal rib design to contain the internal pressure. The number and orientation of the plies for the face-sheet as well as the rib height was unaltered from the preliminary design work. The focus of the analysis was a qualitative comparison and optimization, thus assumptions were made and are listed below.

1. The grid is rectangular
2. Corner radius limited to 2.5 cm
3. The width of the ribs are fixed at 0.47 cm
4. The ribs all have the same height
5. The height of the ribs at the intersections is the same as elsewhere
6. The allowable stresses are 1/3 of the flexure and shear ultimate strengths

There are two important points the must be highlighted in regards to the assumptions above. First, the rib width at 0.47 cm is a result of previous experience manufacturing grid-stiffened structures at AFRL and corresponds to a single width prepreg toe after autoclave curing. Second, there are twice as many layers at the rib intersections, so the ribs will not have uniform height. Since this was used for qualitative comparisons between designs, this assumption should have minimal impact on the optimization process.

The size and spacing of the ribs is dependent on the number and type of ribs chosen. Because of manufacturing limitations, only vertical and horizontal ribs were considered. The vertical ribs started at the base of the shroud (near the open end) and continued over the top of the box to the base on the opposite side of the shroud. The horizontal ribs were continuous around the circumference of the shroud. A grid nomenclature was chosen to make it easier to describe the grid design. The nomenclature is of the form j x k x l. j is the number of vertical ribs on the long side. k is the number of vertical ribs on the short side. l is the number of horizontal ribs.

To determine the appropriate number of ribs, rib spacing and rib height, the pressure shroud was modeled in CosmosWorks 2006 using tetragonal elements and static 3-D analysis techniques. The material properties for the panel are provided on Table 3. The pressure load was applied to the interior surfaces. Two load cases were analyzed. The MEOP was used to compare the maximum displacement for each of the cases, and the burst pressure was used to compare the maximum VonMisses stress for each of the cases. Again clamped (fixed) boundary conditions were used for this analysis. The clamped boundary condition was considered appropriate because of the requirement to seal the base of the pressure shroud against the structural panel.

Table 3. Material properties used for the sizing and finite element analysis. These values are conservative approximations of actual material properties. The design stresses are conservative assumptions.

Ribs		$[(\pm 45/0/90)_3]_s$ Laminate	
Construction	Carbon-Epoxy	Construction	Carbon-Epoxy
Layup	unidirectional tape	Layup	$[(\pm 45/0/90)_3]_s$
Tensile Strength	2.76 GPa	Thickness	0.120 inches
Tensile Modulus	169 GPa	Tensile Strength	945 kPa
Flexural Strength	1.65 GPa	Tensile Modulus	73.1 MPa
Flexural Modulus	148 GPa	Flexural Strength	240 kPa
Short-Beam Shear	128 MPa	Flexural Modulus	44.8 MPa
Design Tensile Stress	550 MPa (1/3 flex)	Design Tensile Stress	80 kPa (1/3 flex)
Design Shear Stress	32 MPa (1/4 shear)		

Two different rib heights and a number of different rib configurations were considered. Because of the thickness limitation of the shroud panels, only 1.27 cm and 1.91 cm rib thicknesses were considered. The 1.91 cm ribs provided the best performance; however, the improvement was not significant enough to constitute the added mass because the same number of ribs was required in either case. As for the rib spacing, a number of variations were evaluated ranging from one to seven ribs evenly spaced across the faces of the shroud. The results are summarized on Table 4. The configuration that provided the lowest mass while still providing adequate design margin was the five vertical ribs by three horizontal ribs by one.

Table 4: VonMisses Stress and Maximum Displacement for Various Rib Cases

Case	*Mass [kg]*	*VonMisses @ Burst [Mpa]*	*Displacement @MEOP [cm]*	*Factor of Safety*
0x0x0	1.81	470	7.730	Fail
1x1x1	1.99	532	0.179	1.04
3x1x1	2.08	438	0.121	1.26
5x1x1	2.18	448	0.120	1.23
7x1x1	2.27	465	0.092	1.19
3x3x1	2.18	329	0.099	1.68
3x3x3	2.26	297	0.095	1.86
5x3x1	**2.27**	**266**	**0.076**	**2.08**
7x3x1	2.37	270	0.075	2.04
5x5x1	2.38	264	0.075	2.09
5x5x3	2.45	289	0.091	1.91
5x5x5	2.61	296	0.094	1.86
7x5x1	2.47	267	0.073	2.07
7x5x3	2.55	258	0.072	2.14
7x5x5	2.70	236	0.075	2.34

The next variable that was evaluated was the spacing between the ribs. The previous analysis assumed the rib spaced evenly across the faces of the shroud as shown in Figure 2a. Since the maximum deflection occurs at the center of the panel, concentrating the ribs towards the center reduced the maximum deflection at the center of the panel, but increases the stress in the face-sheet between the outside ribs and the edge of the shroud because of the increased surface area. The spacing of the ribs was modified to identify the optimum spacing to reduce maximum deflection while maintain the stress in the face-sheet below the design stress of 59 MPa. The results are provided in Table 5, while Figure 2 compares the performance of the evenly spaced ribs, the 3.81 cm spacing, and the 1.27 cm spacing. The optimal spacing for this design is 3.81 cm. Below this spacing, the loading on the face-sheet exceeds the design stress.

Table 5: VonMisses Stress and Maximum Displacement for Various Rib Spacing

Rib Spacing	*VonMisses @ Burst [MPa]*	*Displacement @MEOP [cm]*	*Factor of Safety*
1.27	184	0.060	3.00
2.54	215	0.057	2.57
3.81	225	0.064	2.45
5.08	238	0.076	2.32
6.35	277	0.089	1.99

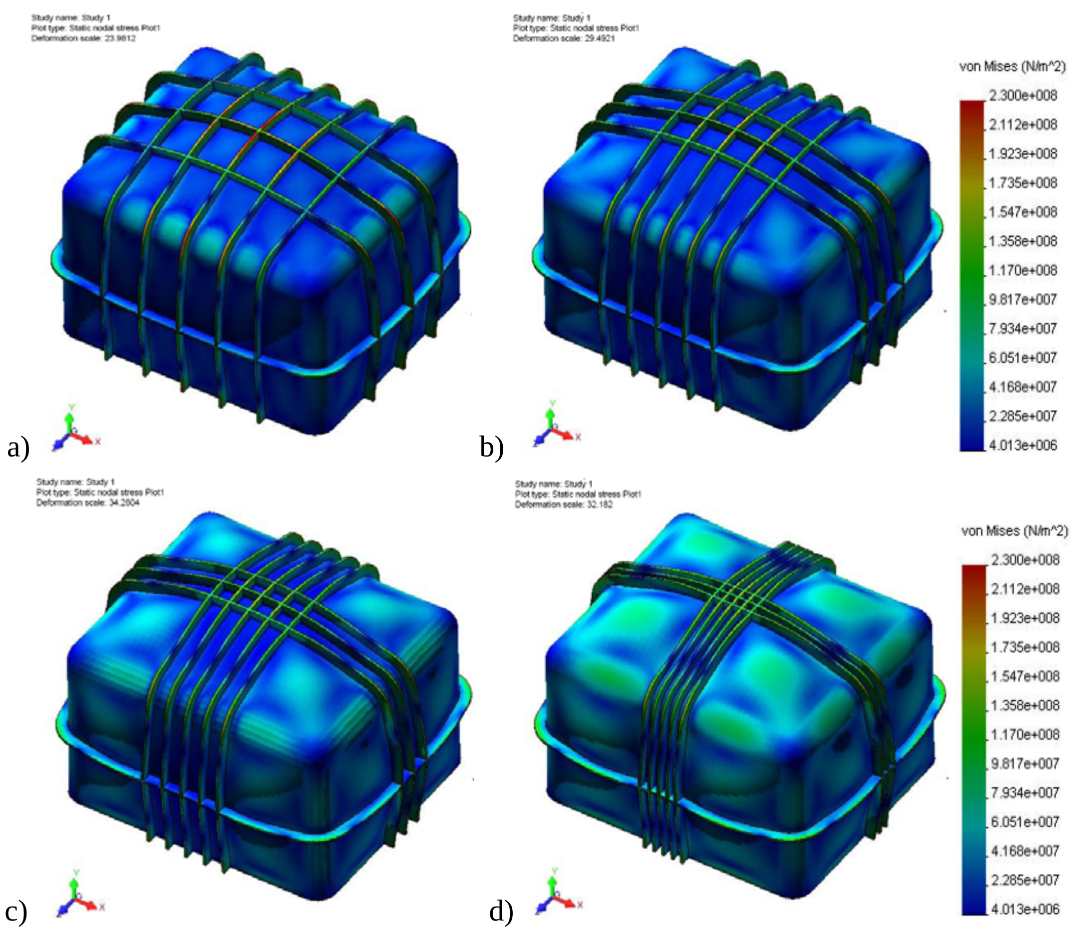

Figure 2: Von Misses stress distribution from CosmosWorks a) for the uniformly space ribs across the surfaces and b) for a rib spacing of 3.81 cm c) for a 2.54 cm spacing and d) for a 1.27 cm spacing

FABRICATION

The pressure shroud was fabricated using the approach outlined in US patent 7479201 for the fabrication of grid-stiffened structures [3]. The face-sheets were laid up on a solid aluminium male mandrel. The face-sheet material was IM7/977-2 plain weave fabric, and the layers were cut using a CNC Gerber cutter. The ribs were laid up using IM7.977-2 12k unidirectional toes. Aircast 3700 rubber was used to provide compaction during the curing process. The advantage of the Aircast 3700 is the very high coefficient of thermal expansion of the material. Finally, a female mandrel made from a composite tooling material was used to contain the entire assembly. Figure 3 shows various stages of fabrication, and the final product is shown in Figure 4. It can clearly be seen in the latter figure that the rib height is not uniform over the entire length of the rib. In addition, the width of the rib is not consistent, which is a result of the compaction process caused by the Aircast 3700 rubber. The consistency of the rib width could be improved by adding rubber between the ribs and the exterior mandrel to account for the height difference around the corners of the shroud. The final mass of the pressure shroud was 1.64 kg.

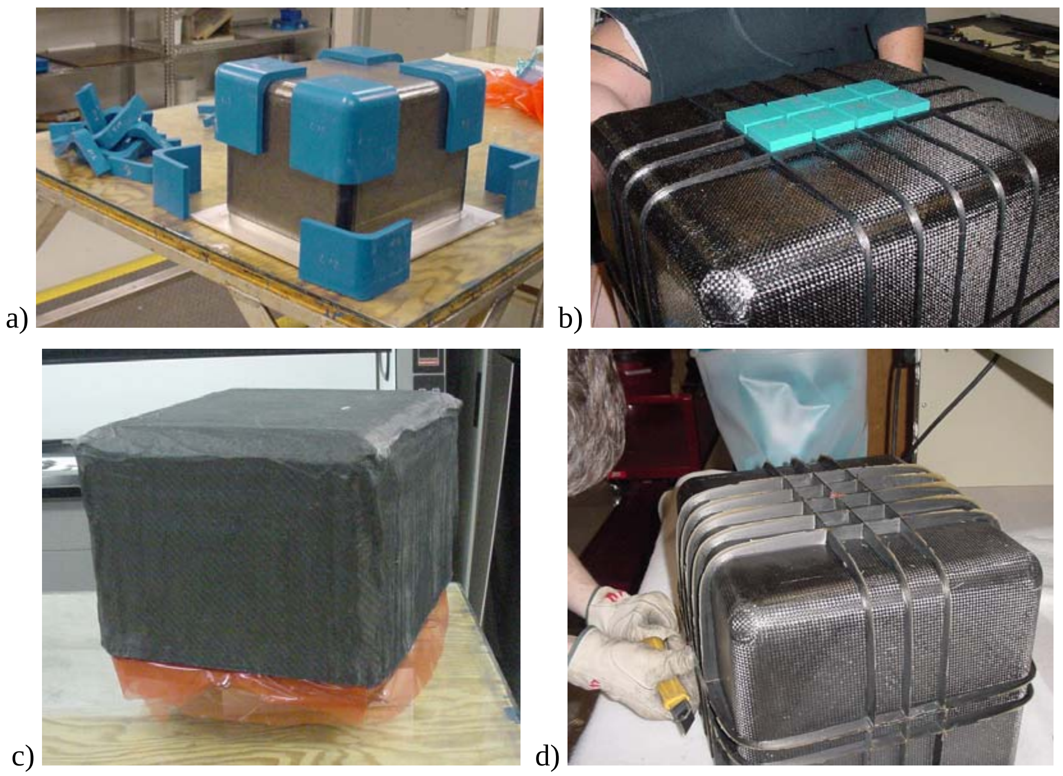

Figure 3: Fabrication process for the grid-stiffened composite shroud: a) a wax film was applied to the aluminum mandrel to account for the face-sheet thickness for fabricating the Aircast 3700 molds, b) the process of laying up the ribs on the face-sheet, c) the final assembly with the composite tooling mandrel before going into the autoclave, and d) removing the excess resin from the cured part.

Figure 4: The final grid-stiffened composite pressure shroud

CONCLUSIONS

The analysis shows that a 5 x 3 x 1 rib configuration provides good design margin for a proof pressure of 241 kPa. The safety factor for the pressure shroud was on the order of 2 with a maximum stress of approximately 225 kPa and a maximum deflection at the center of the top surface of 0.064 cm. This design is assumed to be conservative because the design stress is 1/3 of the flexural stress; however, the current model does not take into consideration the joint design between the rib and the face-sheet. This is an important design consideration, but the nature of the load case from the face-sheet to the rib should minimize the effect of the joint [3]. Experimental testing will be used to validate the model. The system will be pressurized to the proof pressure while measuring the strain on the face-sheets. In addition, the enclosure will be tested to failure to determine the actual burst pressure of the system. These results will then be used to validate and refine the model.

Because of the requirement for a pressurized, hermetically sealed enclosure, this approach will slightly increase the structural mass of the satellite. However, the increased mass is tolerable considering the significant advantages associated with the proposed low-cost, COTS-based system.

References

1. Stienberg D. “Cooling techniques for electronic equipment”. 2nd edition, John Wiley & Sons, 1991.
2. Whitney J. “Structural analysis of laminated anisotropic plates”. 1st edition, Technomic Publishing Company Inc, 1987.
3. Huybrecht, S. et. al., “Method for making advanced grid-stiffened structures,” US Patent 7479201
4. Higgins J., et. al., “Design and testing of the Minotaur advanced grid-stiffened fairing”. Composite Structures, Vol. 66, pp 339-349, 2004.