DESIGN OF A COMPOSITE GRID STIFFENED ENCLOSURE FOR SATELLITE AVIONICS

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Abstract

There is a need in the space community to drastically reduce the time it takes to design and analyze a satellite's thermal control system. One approach to achieving this goal is a forced air convection thermal switch (FACTS). For this approach to be practical, a pressurized, hermetically sealed enclosure with minimal mass is required. The first step in the structural analysis of the enclosure was a preliminary comparison between four different configurations including monolithic metal panels, laminated composite panels, honeycomb panels, and advanced grid-stiffened panels. The honeycomb panel actually had the highest stiffness-to-weight ratio; however, the advanced grids-stiffened panel was chosen for a detailed analytical study because it provided comparable stiffness-to-weight but was more amenable to the pressurized design. Detailed structural sizing of the ribs yields an optimal configuration of five vertical ribs and three horizontal ribs on each face. This configuration provides adequate design margin and center panel deflections.

1 Introduction

Historically, satellite missions are expensive and manpower intensive. Every system is custom built and highly optimized. In addition, the harsh environment and the challenges associated with getting to and operating in space have led to a conservative operational paradigm. As a result, new technologies are difficult to implement and satellite applications tend to be niche or strategic in nature. There is a critical need in the entire space industry to reduce the development time and overall cost of satellite missions to quickly take advantage of new or emerging technologies and to enhance the overall utility of space.

One area of focus that could significantly improve the cost and responsiveness of space operations is the utilization of highly capable small satellites. Because of the reduced complexity of these systems when compared to large, monolithic satellites, they provide a means to quickly integrate new technologies and to enhance space operations to meet new or emerging needs; an example might be regional monitoring during times of natural disasters. Small satellites present a means to transition from a slow, strategic operating paradigm to a more agile, more responsive operating paradigm.

For responsive space operations to be feasible, the primary design drivers for satellites must change. The ultimate goal is a low cost system that can be designed and launched in a matter of days or weeks to meet emerging or changing needs. Classically, mass is the primary design driver followed closely by reliability. For responsive space operations, mass and reliability will remain important, but time and cost will become the primary design drivers. Consequently, to meet this challenge, time and cost must be traded with mass, reliability, and other variables. For responsive space operations to be practical, a new paradigm for satellite design is required.

To achieve these goals, two significant changes to the satellite design process must be made. First, satellite bus designs must be robust and modular so that they can be designed and fielded much more rapidly. Instead of highly optimized, one-off systems, satellites must be built more like commercial systems, such as computers, where custom machines can be built for customers quickly.

A good example of this business model is provided by Dell. Second, commercial off-the-shelf (COTS) components must be used in place of expensive, space qualified components. It has been shown that using COTS components can significantly reduce cost for many systems. In addition, incorporating COTS technology enables rapid technology insertion.

Obviously, there are technical challenges with the successful implementation of each of these changes. Whereas some subsystems lend themselves to modular architectures, others do not. For example, the computer industry has shown through plug-and-play (PnP) functionality that electrical and software interfaces can be modularized. A similar concept, Space PnP Avionics (SPA), is being investigated by the Air Force Research Laboratory for the command and data handling system [1]. On the other extreme, the thermal control subsystem (TCS) represents a complex, distributed subsystem that is not compatible with the concept of system modularity. The TCS is an integrated system that is carefully balanced to ensure proper operation; any small change at the component level has significant ramifications at the system level.

As for the challenge of implementing COTS components into satellite design, the harsh space environment often precludes the use of COTS equipment. The fact that space is a high vacuum complicates the use of COTS equipment because of outgassing, thermal interfaces, convection cooling, and whiskering. COTS components can easily be ruggedized to survive launch, but surviving a high vacuum environment is another matter. To address these issues as well as the thermal modularity issue, a new design methodology based on a Forced Air Convection Thermal System (FACTS) was investigated.

2 FACTS Concept Description

There are three primary elements to the FACTS concept. The first element is that the individual components are grouped by functionality and integrated into a single avionics enclosure as shown on Figure 1. This element provides a more modular design because the bus can be tailored by swapping out different enclosures. For example, if more precise attitude control is required, the attitude determination and control enclosure is replaced with another system that provides better capability. This functional grouping and integrated enclosure concept also simplifies the overall design of the TCS because it limits the number of interfaces that must be controlled. Thermal design is generally separated into two parts: overall bus design and component specific design. A natural breakpoint occurs at the interface between the bus and the subsystems. Rather than having to control the interface for every type of component, only the interface between the subsystem enclosure and the satellite structure would have to be controlled. The detailed thermal design of the components within the enclosure would be completed ahead of time.

The second element incorporates hermetically sealed, pressurized enclosures into the design which enables the use of COTS components. By hermetically sealing the enclosure, the vacuum compatibility and outgassing requirements for components are eliminated opening, up the design space for a wider array of low cost components and materials choices. In addition, the pressurized enclosure reduces the importance of cleanliness and thermal joints that tend to drive up satellite costs through requirements for specialized testing and facilities. Two obvious disadvantages of using pressurized enclosures include the complexity and reliability of a hermetic seal and the added structural mass required to contain the internal pressure.

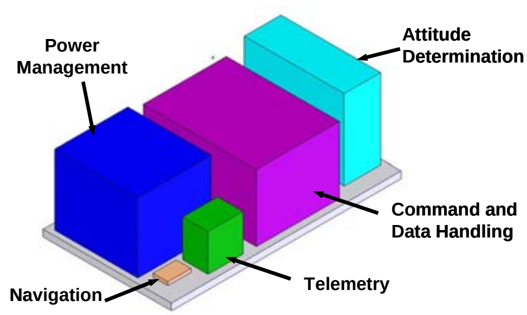

Fig. 1. Subsystem enclosures for a small satellite bus

The third element of the FACTS concept is the use of forced air convection cooling to control the heat transfer rate from the components to the base plate of the enclosure and ultimately to the radiator panels. There are two advantages with forced convection. First, forced convection provides higher heat transfer rates when compared to conduction thus improving the efficiency of the system. Second, a simple DC axial fan can be used as a thermal switch. When heat loads are high, the fan is switched on and provides additional cooling through convection. When loads are reduced, the fan is turned off, and heat is only transferred through conduction and radiation. Disadvantages include the increased power requirement and the associated vibration signature of the fan.

Since mass will always be important for satellites, the added structural mass of the enclosures could eliminate the utility and benefits of the FACTS concept. For this reason, the design and fabrication of the enclosure were the primary focuses of this effort.

3 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 [2]. 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 increases the wall thickness and mass to the point where the design is 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: one is a traditional avionics enclosure and the other is a pressure shroud. This configuration is shown in Fig. 2. An added benefit of this approach is its compatibility with legacy designs.

3.1 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 0.025” was chosen.

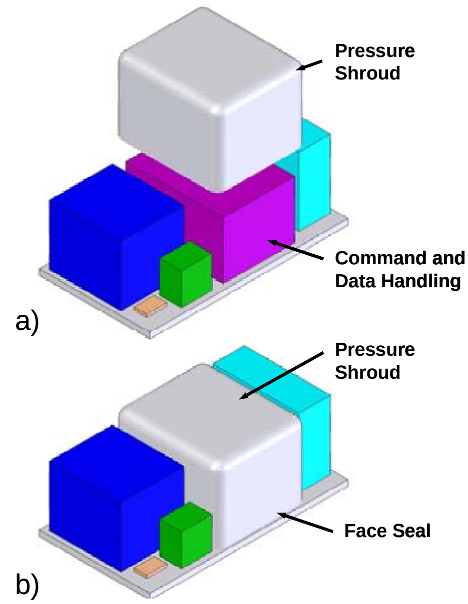

Fig. 2. Depiction of the pressure shroud for the command and data handling subsystem

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 military 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 16.2 psi at standard temperature, the MEOP is 17.5 psi because of the temperature rise from 537 R to 582 R. 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. Maximum transverse deflection less than 0.05” at 17.5 psi (MEOP)

2. Withstand a proof pressure of 26.3 psi without detrimental deformation
3. Withstand a burst pressure of 35.0 psi without rupture
4. Wall thickness shall not exceed 1”
5. Use only space qualified materials
6. Withstand a 15g random vibration launch environment in all axes
7. Mount to a flat plate
8. Provide a hermetic seal with a leak rate of less than 10% loss per year
9. On-orbit lifetime of three years

3.2 Enclosure Configuration Options

Four different structural configurations were considered including hermetically welded metal, bonded and monolithic laminated composite panels, bonded sandwich panels, and monolithic grid-stiffened composite enclosures. Each of the various configurations will be discussed briefly and their potential advantages and disadvantages will be highlighted.

The first configuration investigated was a hermetically welded metallic enclosure consisting of either steel or aluminum walls. The advantages of this system are its low cost and its simplicity to design, analyze, and machine. In addition, it would be relatively easy to seal the box using an integrated flange and a standard o-ring. The disadvantages are the mass required to contain the pressure and the requirement for hermetic welds.

The second configuration was a laminated composite enclosure using 44 0.005” thick carbon-epoxy uniplies with a lay-up of $[45/-45/\ (0/90)_5s]s$. The total thickness was 0.22”. The advantage of using a laminated composite structure is the improved stiffness-to-weight ratio compared to metal walls. In addition, solid composite laminates are fairly straight forward to analyze, and there is a wealth of fabrication experience to draw from. Disadvantages include weight and cost, especially considering expensive hand lay-up processes. Another challenge would be the construction of the box. Secondary bonding of the panels at adjoining edges would not provide adequate strength; fabrication of a monolithic box would be expensive and time consuming.

The third configuration evaluated was a honeycomb panel with two 0.06” aluminum face-sheets and a 0.40” thick aluminum honeycomb core. The system could be improved with carbon composite face-sheets and/or honeycomb cores, but they present added fabrication and bonding challenges. The advantage of a honeycomb structure is the significant improvement in the stiffness-to-weight ratio for minimal impact to weight and cost. The disadvantages include complicated analysis, difficult and costly manufacturing processes, and moisture entrapment within the cells. In addition, the complex face-sheet and core structure make honeycomb panels difficult to join, and it would be nearly impossible to provide a hermetic seal at the joints without significantly adding mass.

A grid-stiffened composite enclosure was the final configuration evaluated. The system consisted of a single face-sheet with a rectangular grid of ribs oriented horizontally and vertically. The face-sheet lay-up was the same as the laminated composite enclosure, and three different thicknesses were evaluated: 44, 20, and 10 plies. The ribs were 0.5” tall, 3/16” thick and were spaced 5” apart starting from a rib at the center. The advantages of this concept are high structural efficiency and stiffness to weight, its amenability to automated processes, and lack of moisture entrapment. The most significant disadvantage is the challenge of accurate modeling and analysis of the structure as detailed FEM models are required to account for non-linear effects and complex load paths between the thin face-sheet and the thick rib structure.

4 Preliminary Down-selection

4.1 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.

The first step in the analysis was to determine the overall size requirement for the pressure shroud. This was accomplished by conducting a short survey of various components and subsystem sizes to determine the enclosure size that needs to be contained. Since there are many system configurations, it is important to initially design the enclosure for the worst case condition. Therefore, from the component and subsystem survey, the maximum size of the box was chosen to be 13.8” x 11.8” x 9.84”, which corresponds to the size of a moderately large command and data handling (CDH) system.

Performing even a first order analysis on a five sided pressure shroud is complex with some of the configurations previously described. For this reason, the analysis was performed on a single, flat panel. It was assumed that the edges of the panel behaved as a fully clamped (CCCC) boundary condition. This assumption was assumed because of the symmetry of the box, the internal pressure exerting equal force on opposite faces, and the bolted flange at the base of the shroud. Realistically, the true boundary conditions are not exactly CCCC; however, assuming fully simply supported conditions yields unreasonably high transverse deflections.

The load applied to the panel for the analysis was 18.4 psi, which represents the atmospheric pressure at sea level plus a design factor of 1.25. Variations in pressure caused by temperature change, as well as, additional pressure requirements for testing and qualifying the design were not considered in the preliminary analysis. However, both of these factors were taken into consideration during the detailed analysis and in defining the design requirements for the shroud.

The metal structures were analyzed using CosmosWorks a finite element analysis program. For the laminated 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) [3]. 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.

Whitney provides two solutions for the structural analysis of laminated plates, dependent upon the lay-up of the laminate [2]. The solution for specially orthotropic laminates ($B_{ij}=0$, $D_{16}=D_{26}=0$) uses either 1) the characteristic shapes found as solutions for natural vibration of a beam with clamped ends or 2) polynomial functions. Solution 1 is quite involved when finding coefficients A_{mn} terms. The one term (m=n=1) solution gives accurate results for displacement, but additional terms are required for accurate maximum moment predictions. Whitney’s second solution, applicable to symmetric plates ($B_{ij}=0$), is also quite involved requiring a minimum of 49 terms (m=n=7) for reasonable accuracy. For the initial sizing of the box, the solution for specially orthotropic laminates with one term polynomial approximation was used. Symmetric laminates were used to minimize coupled bend-twist distortions resulting from hygrothermal effects. Validation of the analysis, coded within Excel with Visual Basic modules, was performed; good agreement was achieved when comparing results for a graphite/epoxy [0/90]s laminate with those of CompositePro®.

To analyze the sandwich panel, two methods were investigated. The first method was similar to the previous method used for the laminate panel with a modified bending stiffness term, D_{ij}, to account for the sandwich construction.

$$(D_{ij})_{sandwich} = \frac{1}{2} h^2 b (a_{ij})_{facesheet} \quad (1)$$

where h is the thickness of the core plus one face-sheet thickness, b is the beam width, and $(a_{ij})_{facesheet}$ was found from inverting the in-plane laminate [ABD] stiffness matrix associated with the facesheet laminate. This approach yields significant errors when non-symmetric flanges are used. For the second method, the honeycomb core and its associated material properties were input as a lamina into CompositePro®, which was then used to analyze the sandwich laminate with the [face-sheet/core/face-sheet], outputting 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 with the output being w_{max}, the maximum transverse displacement of the panel, and the moment resultants. Lastly, CompositePro® was used to find the minimum safety factor based on First Ply Failure Criteria and the Maximum Stress Failure Criteria. The inputs to CompositePro® were the moment resultants along with temperature changes of 0F, 100F, and -100F.

Finally, for the grid stiffened 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 as 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.

4.2 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 laminate 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 1.

Using solid aluminum 6061-T6 panels (Configuration H) required a wall thickness of 0.25" to reduce panel deflection to an acceptable value. The resulting mass was 11.6lbs. For the solid gr/ep laminate (Configuration B), more than 44 0.005" plies are required to achieve acceptable mid-point panel deflections. By comparison, implementing 5" x 5" stiffened sections (Configuration C) significantly reduces the deflection to 0.007". Finally, the aluminum honeycomb sandwich panel had the lowest deflection at 0.005".

Table 1. Summary of the Preliminary Analyses

Case	***Construction***	***Material***	***Dimensions [in]***	***Estimated Weight [lbs]***	***Max Deflection [in]***
A	Aluminum Honeycomb	0.06 in Al face-sheets with 0.40 in Al honeycomb core	13.78 x 9.84 side panel	8.5	0.005
B	Solid Carbon-Epoxy Laminate	44 - 0.005 in uniplies [45/-45/(0/90)$_5$s]s, 0.22 in thick	13.78 x 9.84 side panel	8.4	0.076
C	Grid-Stiffened Carbon-Epoxy Laminate	44 - 0.005 in uniplies [45/-45/(0/90)$_5$s]s, 0.22 in thick	5 x 5 panel	8.9	0.007
D	Grid-Stiffened Carbon-Epoxy Laminate	20 - 0.005 in uniplies [45/-45/(0/90)$_4$]s, 0.10 in thick	5 x 5 panel	4.3	0.021
E	Grid-Stiffened Carbon-Epoxy Laminate	10 - 0.005 in uniplies [0$_3$/90/0]s, 0.05 in thick	5 x 5 panel	2.5	0.160
F	Grid-Stiffened Carbon-Epoxy Laminate	16 - 0.005 in uniplies [(0/90)$_4$]s, 0.05 in thick	5 x 5 panel	3.5	0.040
G	Aluminum	6061-T6 t = 0.200 in	12.20 x 11.40 side panel	9.3	0.063
H	Aluminum	6061-T6 t = 0.250 in	12.20 x 11.40 side panel	11.6	0.032
I	Steel	1025 Steel, t = 0.150 in	12.20 x 11.40 side panel	20.0	0.052

Of the different design configurations, the greatest stiffness-to-weight ratio was achieved with the aluminum skin/aluminum honeycomb core sandwich panel configuration (Configuration A). However, the grid stiffened design (Configuration E) provided the lowest overall mass. If the design of the sandwich panel had been optimized, it would have been more mass competitive with the grid-stiffened design and most likely would have been the number one design choice. However, after careful consideration of all aspects of the requirements including manufacturing, integration, and hermetic sealing, the grid-stiffened design was chosen to proceed with. The primary reasons for this decision were the fact that the sandwich panel construction would be nearly impossible to seal at the edges and that when the full shroud configuration was considered a significant amount of mass would have to be added at the corners to stiffen the structure so that it could carry the load. Further, the grid-stiffened enclosure design provided the ability to radius the corners of the shroud thereby reducing the stress at these locations.

Based on the preliminary analysis, the grid-stiffened carbon-epoxy structure was chosen for further analysis. This design showed the greatest potential for least mass while achieving reasonable amounts of deflection. This structure also avoided moisture entrapment, joining at corners, and other core problems associated with the aluminum skin on honeycomb structure.

5 Detailed Design and Analysis

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 sizing the ribs to contain the internal pressure. The number and orientation of the plies for the face-sheet was unaltered from the preliminary design work. The required information was the rib height and the rib spacing in both the horizontal and vertical directions. For the analysis, several assumptions were made and are listed below.

1. The grid is rectangular
2. The enclosure has rounded corners with a 1" radius to reduce stress in these corners
3. The width of the ribs are fixed at 0.187"
4. The ribs all have the same height
5. The height of the ribs at the intersections are 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 0.187" is a result of previous experience manufacturing grid-stiffened structures at AFRL and corresponds to a single width prepreg tow after autoclave curing. Second, even though there are twice as many layers at the rib intersections, the overall height at these sections is the same as other rib sections. The uniform height is a result of the fabrication process in which the ribs are constrained by male and female mandrels, which forces excess resin out of the rib intersection creating a fiber rich zone.

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, and l is the number of horizontal ribs.

To determine the appropriate rib height and rib spacing a single panel was evaluated to simplify the analysis. The panel was modeled in CosmosWorks 2006 using tetragonal elements and static 3-D analysis techniques. The material properties for the panel are provided in Table 2. For the panel, only the ribs were modeled. This was done to size the ribs so that they carry the entire load and the face-sheets are only required to resist deflection over the area between ribs. To determine the load on the ribs, the total load on the ribs was calculated by multiplying the area of the face by the proof pressure. The total load was then applied as a uniform load over the face of the ribs. This approach will provide a conservative design.

Table 2. Material properties used for the sizing and finite element analysis.

Ribs		$[(\pm45/0/90)_3]_s$ Laminate	
Construction	Solid Carbon-Epoxy	Construction	Solid Carbon-Epoxy
Layup	unidirectional tape	Layup	$[(\pm45/0/90)_3]_s$
Tensile Strength	400 ksi	Thickness	0.120 inches
Tensile Modulus	24.5 Msi	Tensile Strength	45 ksi
Flexural Strength	240 ksi	Tensile Modulus	8.0 Msi
Flexural Modulus	21.5 Msi	Flexural Strength	35 ksi
Short-Beam Shear Strength	18.5 ksi	Flexural Modulus	6.5 Msi
Allowable Tensile Stress	60 ksi (1/4 flexural)	Allowable Tensile Stress	8.5 ksi (1/4 flexural)
Allowable Shear Stress	4.5 ksi (1/4 shear strength)		

Again clamped (fixed) boundary conditions were used for this analysis. The clamped boundary condition was considered appropriate because the opposite panel had an equal and opposite force. These equal and opposite forces are joined with unidirectional carbon-epoxy ribs loaded axially. Minimal displacement in this axially loaded rib was assumed to act as a clamped boundary.

Two different rib heights and a number of different rib configurations were considered. Because of the thickness limitation of the shroud panels, only 0.5” and 0.75” rib thicknesses were considered. The 0.75” 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 including no horizontal ribs, one horizontal rib, three horizontal ribs, and between one and seven vertical ribs. The configuration that provided the lowest mass while still providing adequate design margin was the five vertical ribs by three horizontal ribs. The results are shown in Fig. 3.

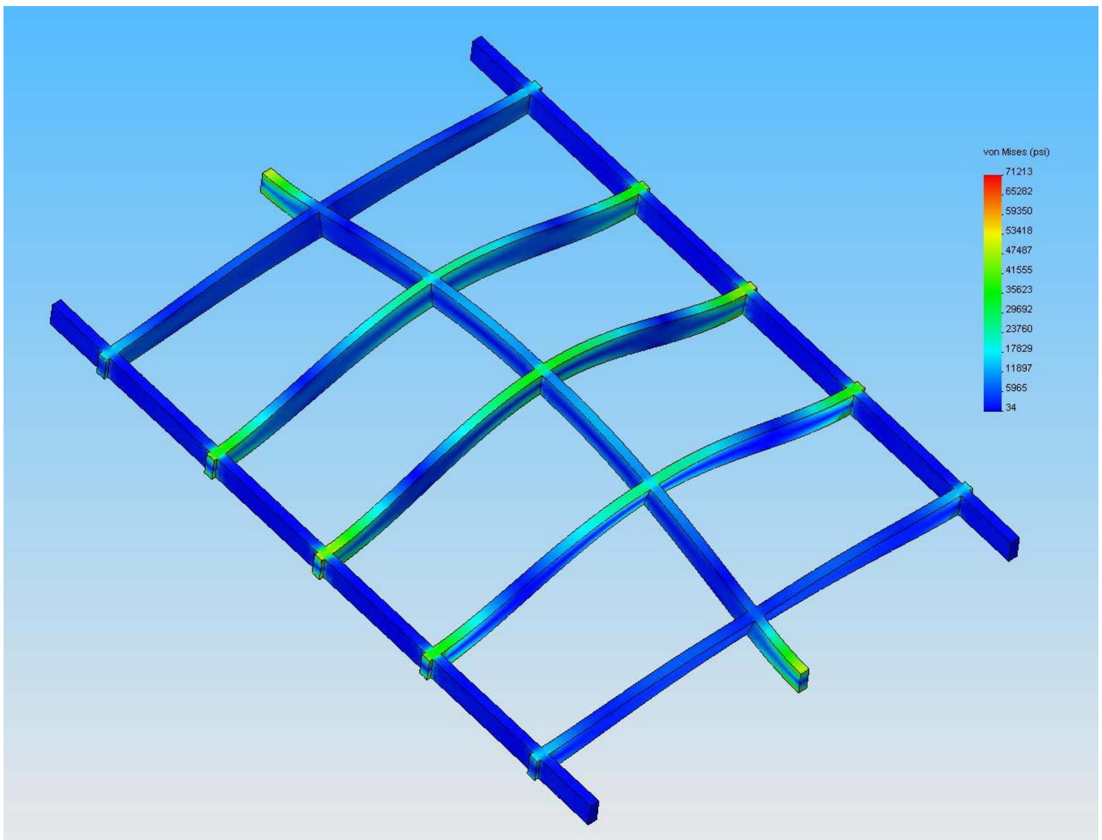

Fig. 3. Von Mises stress in a 5 x k x 3 panel with 0.5 in tall ribs. Individual panels were modeled before doing a complete model of the structure. The maximum Von Mises stress here is acceptable.

Once the rib spacing was decided, the maximum stress and deflection of the face-sheet could be calculated. The stress and deflections were calculated using the following equations for a uniformly loaded rectangular plate with all edges clamped:

$$S = \frac{kwR^2}{t^2} \text{ and } y = \frac{k_1 wR^4}{Et^3},$$

where S is the stress, y is the deflection, k is a constant from 0.308 to 0.500 which is based on the lengths of the sides, k_1 is a constant from 0.0138 to 0.028 which also depends on the length of the sides, w is the distributed load, R is the length of the longest side, E is Young’s modulus, and t is the thickness. Young’s modulus values were obtained using CompositePro®. These equations are only valid where flexure stresses dominate and where all transverse deflections are small, typically less than 0.4*t*, where *t* is the thickness of the skin. All deflections in these analyses remained under this limit. Additional limitations may exist when applying these equations to laminated, though quasi-isotropic, composites. These limitations were assumed to be minimal. The maximum stress and deflection for the worst case critical location were 9 ksi and 0.006", respectively.

Using the results from the single panel analysis, a solid model of the entire shroud was developed. The model was used to verify the suitability of the 5 x 5 x 3 configuration as well as to determine the stress concentrations at the corners of the shroud. Because of the 1" radius requirement at the corners, there is a significant stress concentration as the rib tries to resist the bending loads at the center of the plate. The radius of the ribs was also the location of the highest stress concentrations, which was to be expected. As with the previous analysis, only the ribs were modeled. Again, this was done to size the ribs such that they carried the entire load of the structure. The final results are presented below in Fig. 4.

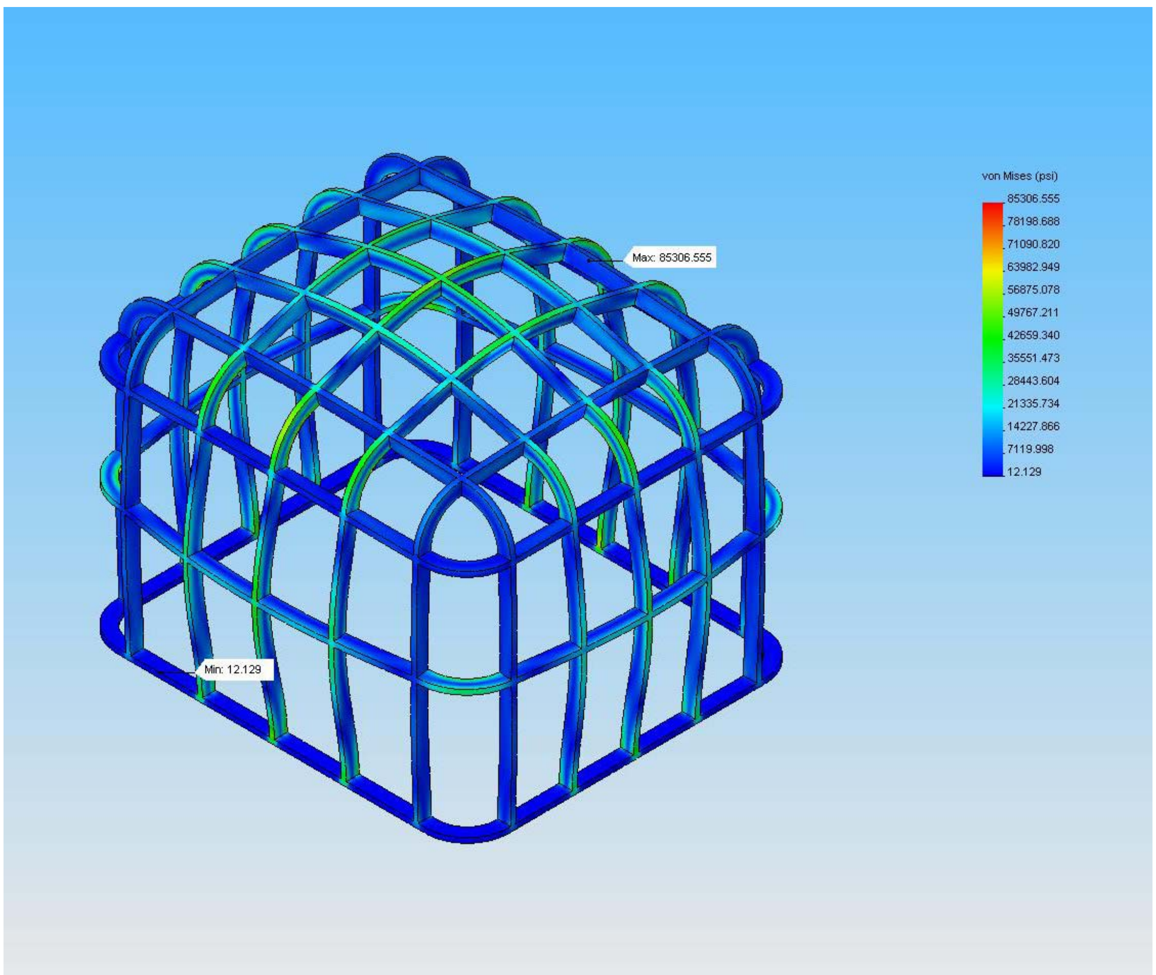

Fig. 4. Final rib analysis using finite elements. This 5 x 5 x 3 design using 0.5 in ribs meets the stress allowables. The largest Von Mises stress is 85 ksi and the maximum deflection is 0.083 in at the proof pressure of 25.5 psi.

6 Conclusions

The analysis shows that a 5 x 5 x 3 rib configuration provides adequate design margin for a proof pressure of 25.5 psi. The safety factor for the pressure shroud was on the order of 3 with a maximum stress of approximately 80 ksi. This design is assumed to be conservative because it does not integrate the effect of the ribs and face-sheet into a single model. Because of the thin face-sheet and the thick ribs, it is difficult to properly mesh the system for accurate results. 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. The final step will be to optimize the system using the refined model. It should be noted that this is a conservative estimate, and the mass can be reduced by optimizing the grid spacing. In addition, controlling the joint design between the skin and the ribs is critical [4].

Because of the requirement for a pressurized, hermetically sealed enclosure, the FACTS concept will increase the structural mass of the satellite system by ~5%. However, the increased mass is tolerable considering the significant strategic advantages associated with the proposed modular, robust system.

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