Structural Design and Analysis of Un-pressurized Cargo Delivery Vehicle

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As part of the Exploration Systems Architecture Study, NASA has defined a family of vehicles to support lunar exploration and International Space Station (ISS) re-supply missions after the Shuttle's retirement. The Un-pressurized Cargo Delivery Vehicle (UCDV) has been envisioned to be an expendable logistics delivery vehicle that would be used to deliver external cargo to the ISS. It would be launched on the Crew Launch Vehicle and would replace the Crew Exploration Vehicle. The estimated cargo would be the weight of external logistics to the ISS. Determining the minimum weight design of the UCDV during conceptual design is the major issue addressed in this paper. This task was accomplished using a procedure for rapid weight estimation that was based on Finite Element Analysis and sizing of the vehicle by the use of commercially available codes. Three design concepts were analyzed and their respective weights were compared. The analytical structural weight was increased by a factor to account for structural elements that were not modeled. Significant reduction in weight of a composite design over metallic was achieved for similar panel concepts.

Nomenclature

$C_{A\alpha}$	=	axial force coefficient per degree alpha
$C_{N\alpha}$	=	normal force coefficient per degree alpha
d_r	=	launch vehicle 1st stage diameter
$dC_{N\alpha}/d(x/d_r)$	=	sectional load distribution of normal force coefficient per degree alpha
$dC_{A\alpha}/d(x/d_r)$	=	sectional load distribution of axial force coefficient per degree alpha
F_N	=	normal aerodynamic force on the vehicle
g	=	acceleration due to gravity
N_x	=	longitudinal load per unit length
N_y	=	circumferential load per unit length
q	=	dynamic pressure
S	=	cross sectional area of the Un-pressurized Cargo Delivery Vehicle
x	=	launch vehicle section
x_L	=	combined length of the Nose Cone and the Un-pressurized Cargo Delivery Vehicle
α	=	angle of attack

I. Introduction

As part of the Exploration Systems Architecture Study (ESAS) [1], NASA has defined a family of vehicles to support lunar exploration and International Space Station (ISS) re-supply missions after the Shuttle's retirement. The Un-pressurized Cargo Delivery Vehicle (UCDV) has been envisioned to be an expendable logistics delivery vehicle that would be used to deliver external cargo to the ISS. It would be launched on the Crew Launch Vehicle and would replace the Crew Exploration Vehicle. The estimated cargo would be 16,000 lbs of external logistics to the ISS.

Determining the minimum weight design of the UCDV during conceptual design is the major issue addressed in this paper. This task was accomplished using a procedure for rapid weight estimation that has been developed for evaluating new emerging vehicle concepts [2,3].

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Three design concepts were analyzed and their respective weights were compared. The general outline of the process used for estimation of structural weight during conceptual design of a new vehicle will be presented first. This process is based on Finite Element Analysis and sizing of the vehicle by the use of commercially available codes. Interaction between these codes has been facilitated through the integration and use of in-house developed programs in order to reduce the role of the user-in-the-loop.

The three structural wall design concepts were: metallic semi-monocoque, composite semi-monocoque, and composite monocoque. The iterative procedure produced an axially asymmetric vehicle design due to the nature of asymmetric loading conditions. For ease of manufacturing and because of uncertainties in payload mass distributions at the conceptual level, a more robust common-gauge design was desired. That was achieved with an additional loop in the iterative process.

The analytical structural weight does not account for structural elements that are not modeled, such as fittings, panel edges, fillets and so on. This additional weight was treated as so-called “non-optimum” weight and was accounted for in the conceptual phase with a factor that varies for different panel concepts.

Weights of three designs were summarized and compared at the sub-assembly and assembly level as well as between analytical and “as-built” designs.

II. Un-pressurized Cargo Delivery Vehicle Design

The UCDV is composed of the following three elements (Fig. 1): Service Module (SM), Un-pressurized Cargo Delivery Module (UCDM) and Orbital Replacement Unit (ORU) Pallets. The SM will also be used by the Constellation Program architecture for different kinds of tasks, such as delivery of the Crew Exploration Vehicle (CEV) to its destination. Design and analysis of the SM is not covered in the present work. Attached to the SM is the Un-pressurized Cargo Module. This tuning-fork-like module is a major load carrier, transferring loads from the payload to the SM and the launch vehicle. Two ORU Pallets provide the interface between the payloads and the UCDM. The pallets are also the interface between payloads and dedicated storage locations on the ISS truss assembly. The UCDV will attach to the ISS and will remain attached while the ORU Pallets are transferred between UCDM and locations on the ISS truss. In order to reduce the UCDV stay time at the ISS, whole pallets containing the payloads will replace existing units in space.

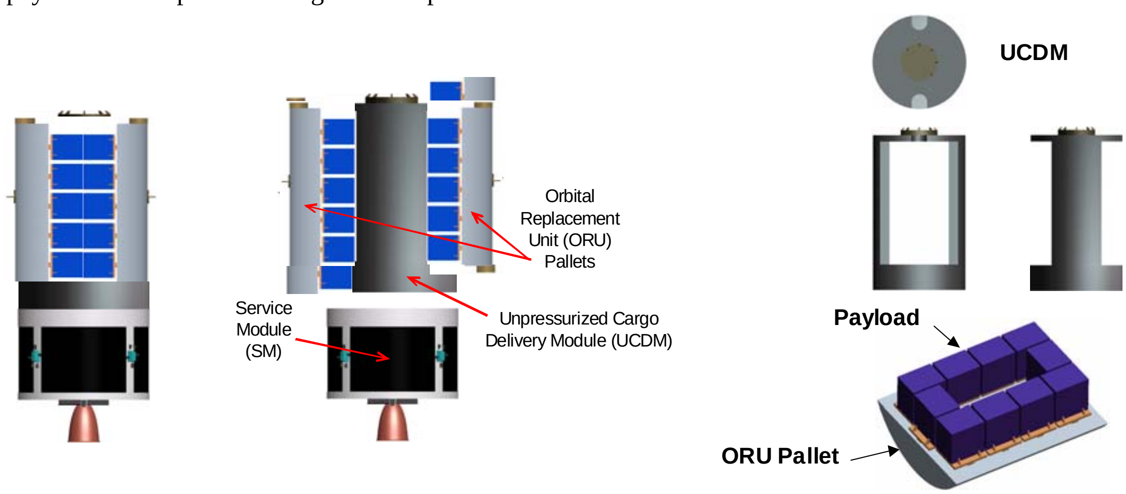

Figure 1. Un-pressurized Cargo Delivery Vehicle

III. Design Process

The major objective of this work was to determine the structural weight of two elements of the UCDV: the UCDM and ORU Pallets. In addition, design and weight estimates of the vehicle Nose Cone were done only to the level necessary to introduce the aerodynamic loads into the analysis process. Three design concepts were analyzed and their respective weights compared. The general outline of the process used for estimation of structural weight during conceptual design of new design vehicles will be presented first. This process is based on Finite Element Analysis (FEA) and sizing of the vehicle by the use of commercially available codes. Interaction between these codes has been facilitated through the integration and use of in-house developed programs in order to reduce the role of the user-in-the-loop (Fig. 2).

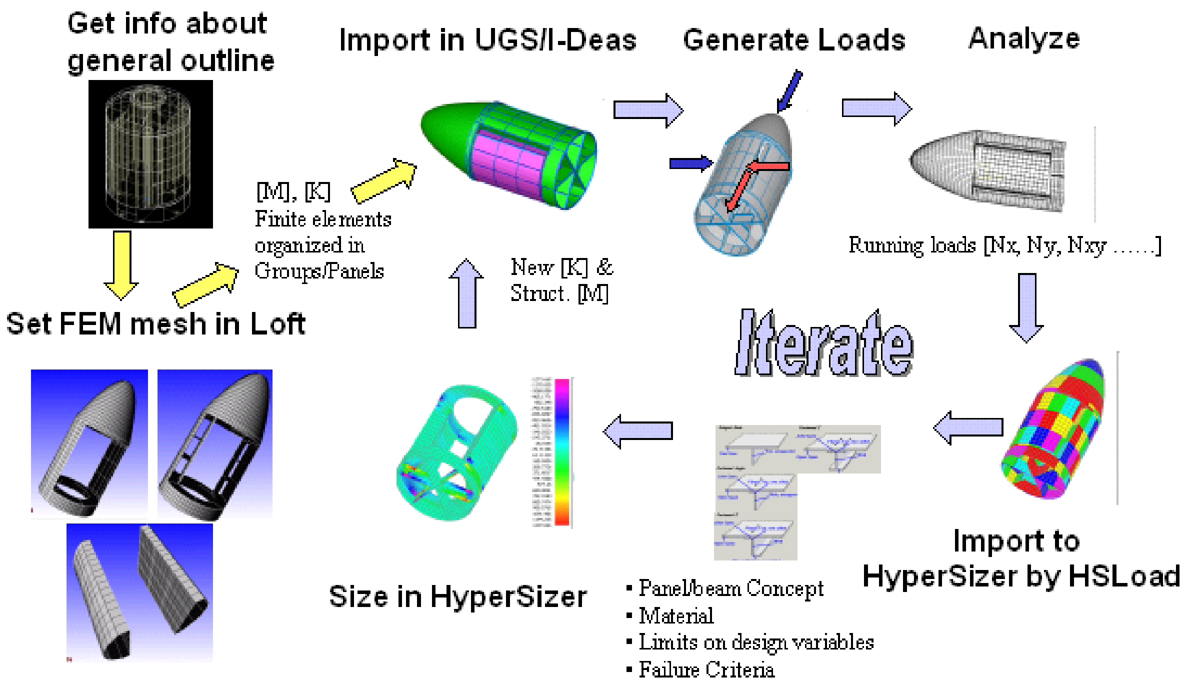

Figure 2. Process outline

The process starts from a collection of information about geometric configuration, structural arrangement and design information that includes knowledge of subsystem masses that will be used in the load definition process [4]. An in-house developed Finite Element Model (FEM) mesh generating program (LOFT), that parameterizes the vehicle geometry and groups the mesh into vehicle structural panels and beams, runs next. LOFT output is loaded into the UGS/I-Deas® [5] commercial CAE program where preliminary values of structural mass and stiffness are introduced. External loads are modeled and the FEM analyzed for different load cases. Results of each load case run, in form of running loads (or line loads), as well as FEM geometry, are imported to another commercial software from Collier Research/HyperSizer® [6]. FEM qualities are changed in HyperSizer in such a way that structural panels and groups are further grouped into groups or sub-assembles with common design concepts, materials, limits on design variables and failure criteria. This grouping is facilitated again through the use of an in-house developed computer program (HSLoad) and application of HyperSizer templates. HyperSizer is a structural component design and analysis program that sizes each of the panels/beams to minimum weight within a pre-defined design variable range. The newly sized vehicle will have updated structural mass and stiffness that are representative of sizing changes required to satisfy margins of safety for multiple FEM defined load cases. The updated FEM has to be imported to I-Deas to resolve inconsistencies between the FEM input and the re-sized panel and beam geometry. Several iterations between this analysis and sizing process will be necessary to arrive at a converged design state.

IV. Analytical Model

Shell and beam finite elements were used to model the Nose Cone, ORU Pallets and the UCDM Support structure. These finite elements were grouped into components that were physically equivalent to: vehicle panels, webs, beam caps, frames and stiffeners (Fig. 3). The length of the UCDV was 281 inches and the diameter was 216.54 inches. The nose cone was 200 inches long. Initial stiffness and mass properties were assigned to them. Several parts of the vehicle, such as passive Common Berthing Mechanism (CBM) and Payload, were modeled with lump masses (Fig. 4). They were respectively attached to the UCDM and ORU by rigid elements. Two ORU Pallets were attached to the UCDM Support structure with four rigid elements each. These elements transfer loads like hinges or ball joints in sockets. Six attachments of the model to the SM transferred only forces. ORU Pallets and the UCDM Support structure were designed with outer and inner skin, bulkheads and longerons (Fig. 5).

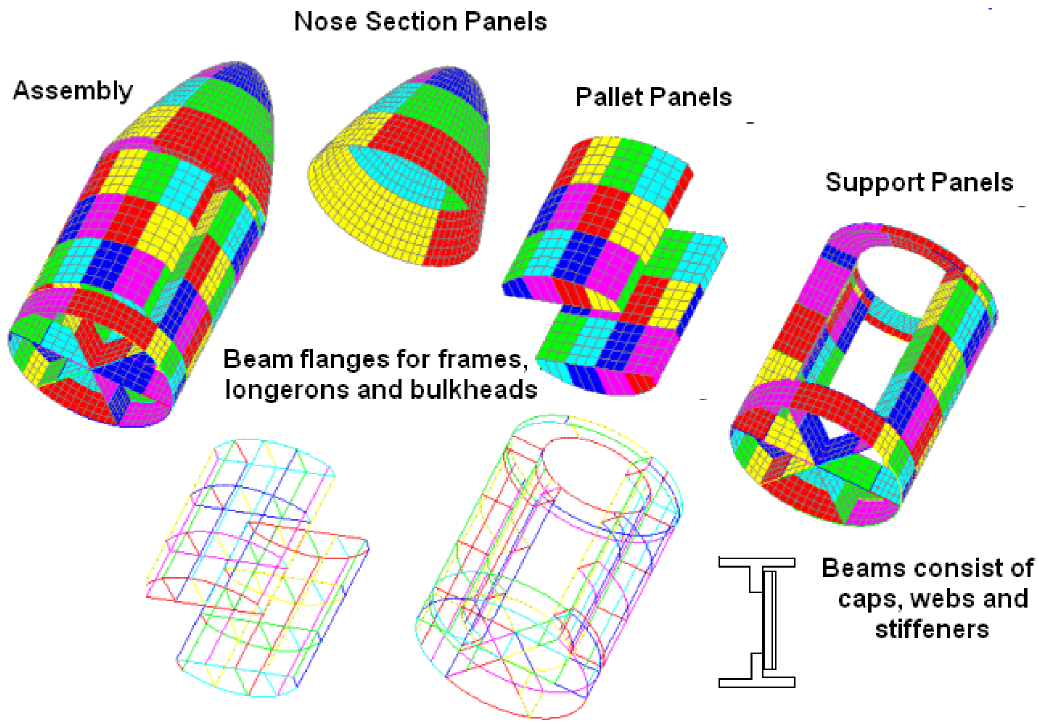

Figure 3. Finite elements grouped in components

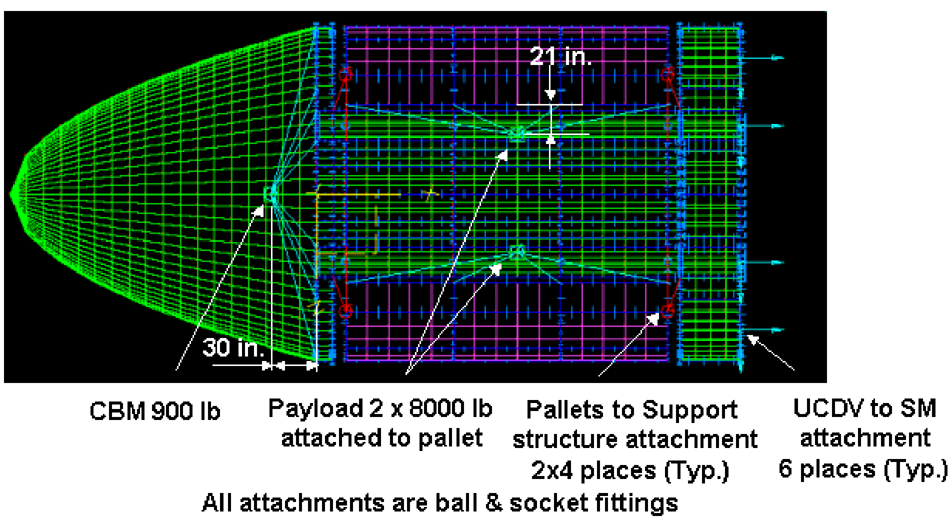

Figure 4. Finite Element Model

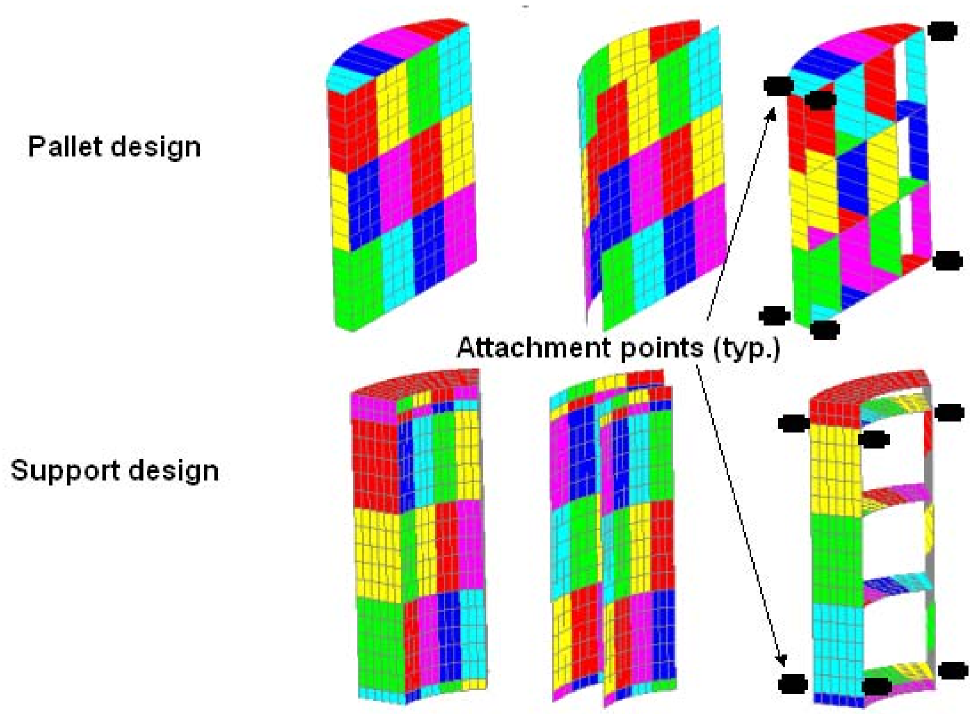

Figure 5. FEM of Pallet and Support Structure

V. Loads

Three points on the ascent flight profile of the launch vehicle were considered for load case definition (Fig. 6). They were: liftoff, maximum dynamic pressure and maximum axial acceleration.

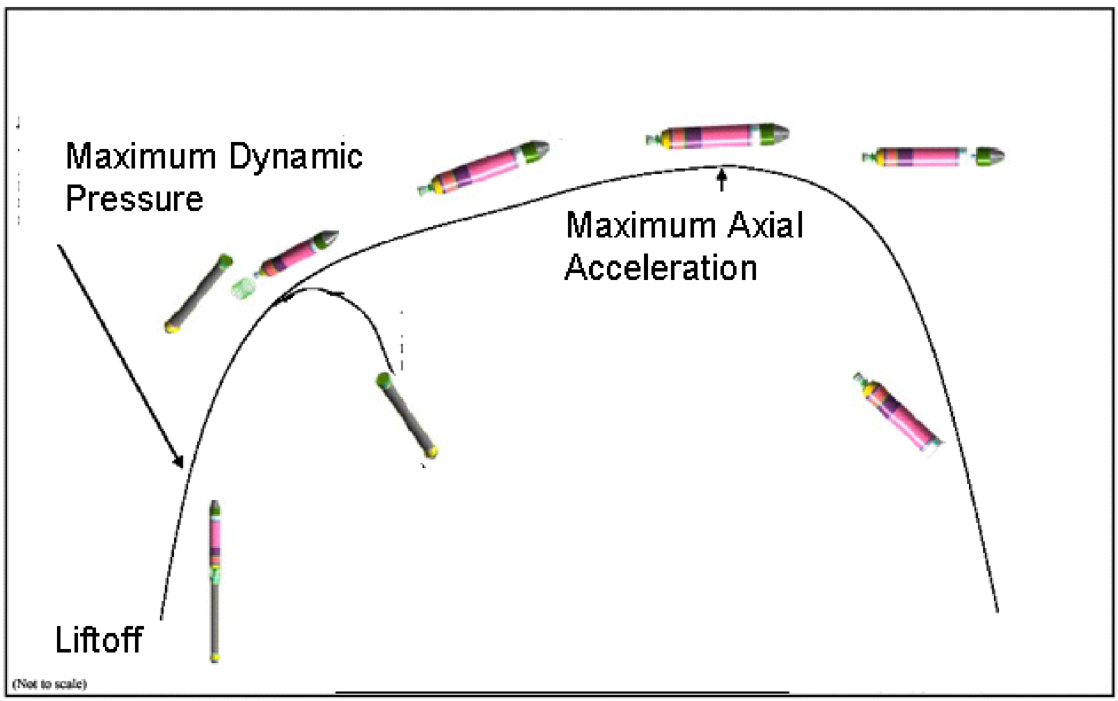

Figure 6. Three flight load points on the launch vehicle flight profile

Aero loads and inertial loads were combined to get the total load when the vehicle reaches the maximum dynamic pressure point on the ascend flight path (Fig. 7). The inertial loads are modeled with 1.9g axial acceleration and 0.141g lateral acceleration. The following information was used to derive aero forces for the Nose Cone and UCDV: maximum dynamic pressure q = 832 lb/ft^2, the average value of the sectional load distribution of normal force coefficient per degree alpha along the upper part of the vehicle $[dC_{N\alpha}/d(x/d_r)]_{avr}$ = 0.04, the average value of the sectional load distribution of axial force coefficient per degree alpha along the upper part of the vehicle $[dC_{A\alpha}/d(x/d_r)]_{avr}$ = 0.00314, where x represented the launch vehicle section, x_L was equal to the combined length of the Nose Cone and UCDV (i.e. 481 inches), and d_r was the vehicle 1st stage diameter equal to 146.08 in. The cross sectional area of the UCDV was S = 36,827 in^2. Dispersion of the angle of attack was α = 5°. Equation (1) defines the normal aerodynamic force on the vehicle. This force was mapped to the FEM. The axial force was defined similarly.

$$F_N = q\,S\,\alpha\,[dC_{N\alpha}/d(x/d_r)]_{avr}\,(x_L/d_r) \tag{1}$$

Inertia loads due to the acceleration were used to model the liftoff loads. Axial acceleration equal to 2g was applied along the vehicle axis. Lateral acceleration of 1.56g was applied separately along two mutually perpendicular directions and along the diagonal in a plane normal to the vehicle axis. Three resultant liftoff load cases were assembled from the axial and lateral inertial loads.

Maximum vehicle's axial acceleration was the third loading condition and it was represented in the analysis with inertia force caused by a 5g axial acceleration and no aero loads were modeled.

Since the UCDV has two planes of symmetry, the lateral loads at liftoff and maximum dynamic pressure may be applied in each principal direction and at 45 degrees to the symmetry planes. The three loading conditions were therefore modeled with seven load cases so that the major vehicle axial-cross-section inertial planes were loaded (Fig. 7).

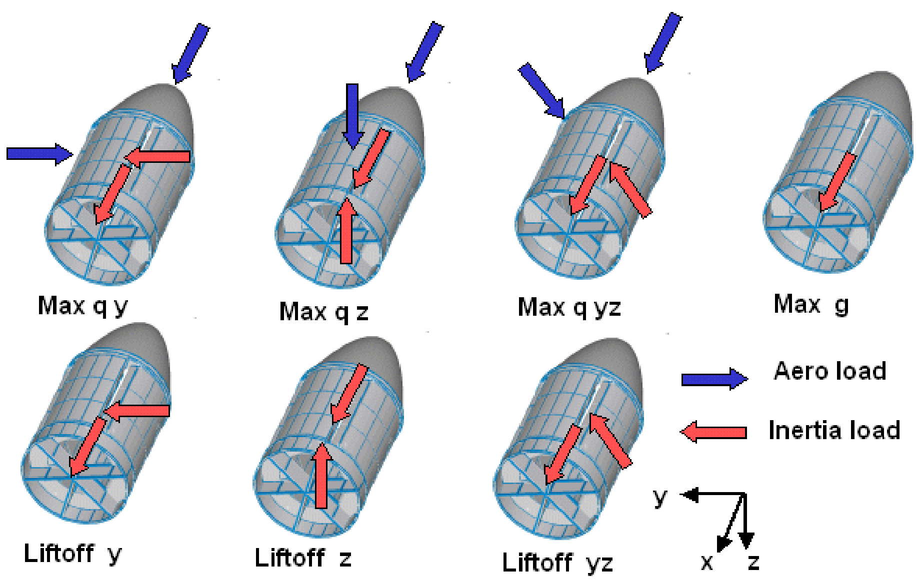

Figure 7. Seven load cases

VI. Panel Sizing Preliminary Study

A quick assessment of different panel structural concepts was conducted before performing the more detailed FEA based approach. HyperSizer was used to get panel structural design gauges and mass estimates for a representative curved panel whose size, end conditions and loading are shown (see Fig. 8). Five different panel families were investigated with metallic and composite materials and the resulting weight per unit area summarized below. Equivalent orthotropic material properties of composite materials are used in HyperSizer for quick optimizations. Several ply layups were considered with Graphite Epoxy IM7/977-2 plies. This study indicated that the analytical weight of sandwich panels is the smallest and that, predictably, composite material panels are lighter than aluminum panels. Controlling failure modes were buckling for sandwich panels and local buckling for uni-axially stiffened panels.

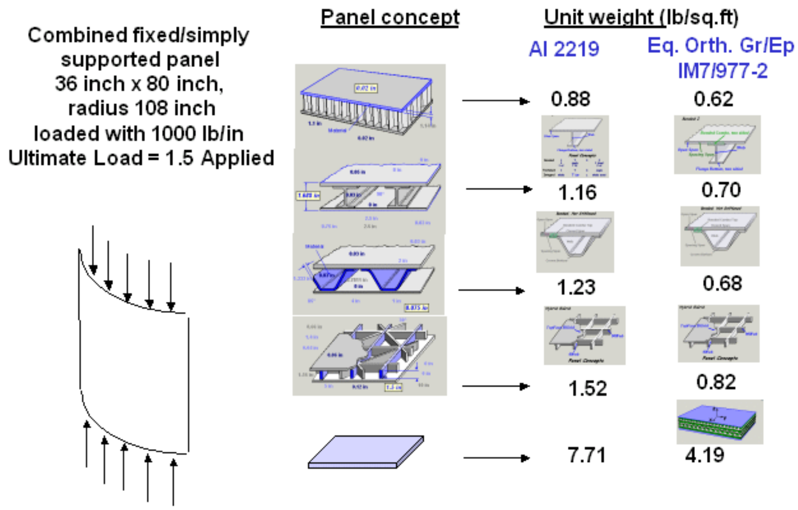

Figure 8. Results of the preliminary study

VII. Three Design Concepts of UCDV

Based on the results of the preliminary study, three structural wall design concepts were considered: metallic semi-monocoque, composite semi-monocoque, and composite monocoque. Figure 9 summarizes the three designs and initial concepts that were applied to the vehicle except to the Nose Cone. Uniaxial stiffened panels were selected for stiffened skin because the sandwich panel's actual weight maybe higher than predicted by the preliminary study because of high non-optimum weight factors. The design process started with nine different aluminum skin and panel concepts, and five aluminum beam shapes (Fig. 9). The sizing process produced optimum panel and beam shapes, thickness and stiffener sizes for different parts of the vehicle. The composite material monocoque design used the beam shapes that were optimal for the aluminum design, and optimized beam thickness, size and selected materials. Skin thickness and material were also optimized. The composite semi-monocoque design was achieved by optimizing the beams, as for the monocoque design, and by retaining the same stiffened panel concepts as in the optimum aluminum design. Selection of panel materials, and optimization of panel thickness, size of stringers and distance between stringers was also done.

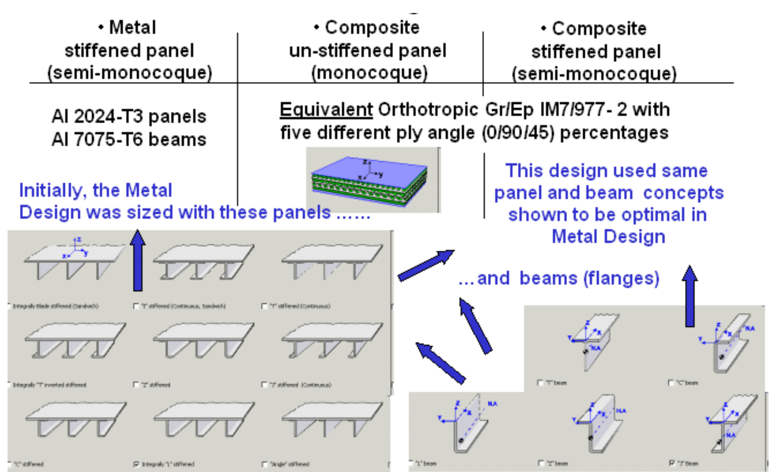

Figure 9. Three designs

The process, shown in Fig. 2, produces an axially asymmetric vehicle due to the nature of asymmetric loading conditions that may also act along opposite sides of the vehicle and cause a different material distribution. Also, for ease of manufacturing, it may be required to build assemblies of panels with common gauges. Further, at the conceptual level, the modeler typically will not cover all possible scenarios such as payload distributions that may cause additional design loads. Additions to the outlined analysis process are required that will distribute material so that the final design encompasses these uncertainties and omissions. The current process consists of applying an identical design to groups of panels and causes re-distribution of weight that has to be accounted for in the iteration process (Fig. 10). The user first reviews the magnitude of the running loads and regroups the current distribution of panels/beams into new groups that preserve vehicle axial symmetry and running loads magnitude. If unsure about the distribution of subsystem mass, such as a payload, the user may expand a group of panels with new panels along the vehicle axis to account for uncertainties in subsystem inertia loads. This regrouping starts even before the standardization process and continues during the subsequent iterations. The standardization process follows after the HyperSizer optimization run and consists of one or more reanalysis runs. The user reviews margins of safety after the optimization and assigns to a group of panels/beams the design of the component that was most critical to violate one of failure criteria. The reanalysis run follows and, if no constraints are violated, the new vehicle design is exported to I-Deas and the new iteration process starts. If the reanalysis results in one or more failed panels/beams, then the failed component affects the design of the whole group and the new reanalysis would follow.

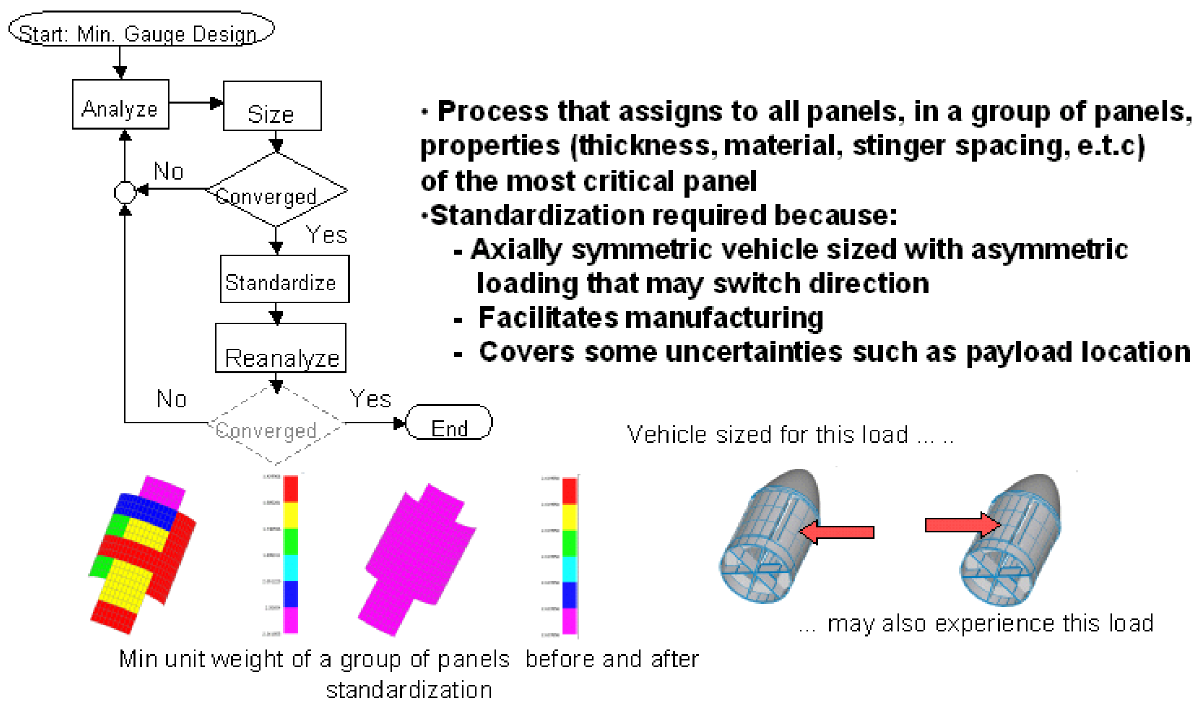

Figure 10. Design standardization process

VIII. Results

A. Metal Semi-Monocoque Design

The initial all-metal semi-monocoque design was evaluated first (Fig. 11). UCDV panels and stringers had material properties of aluminum 2024-T3. Beam material was aluminum 7075-T6. Of the nine initially considered HyperSizer panels, some had stringers that were integrally machined with the skin, while the other panels had stringers fastened to the skin. Flange and web thickness were linked by the sizing process both for the stringers and for the beams. The Nose Cone was modeled with sandwich panels that had aluminum 2024-T3 face sheets and an optimized aluminum core (Fig. 11).

Minimum gauge and size limits were applied to the design variables (Table 1). Local optimization was applied in HyperSizer and individual vehicle panels were optimized to satisfy failure criteria based on strength, stiffness, global and local buckling, crippling, face sheet wrinkling and dimpling, core crushing and other industry standards.

A safety factor of 1.5 was applied to the limit loads. All failure analyses had to result in positive margins of safety for each panel or beam. The optimization process started with the minimum gauge design.

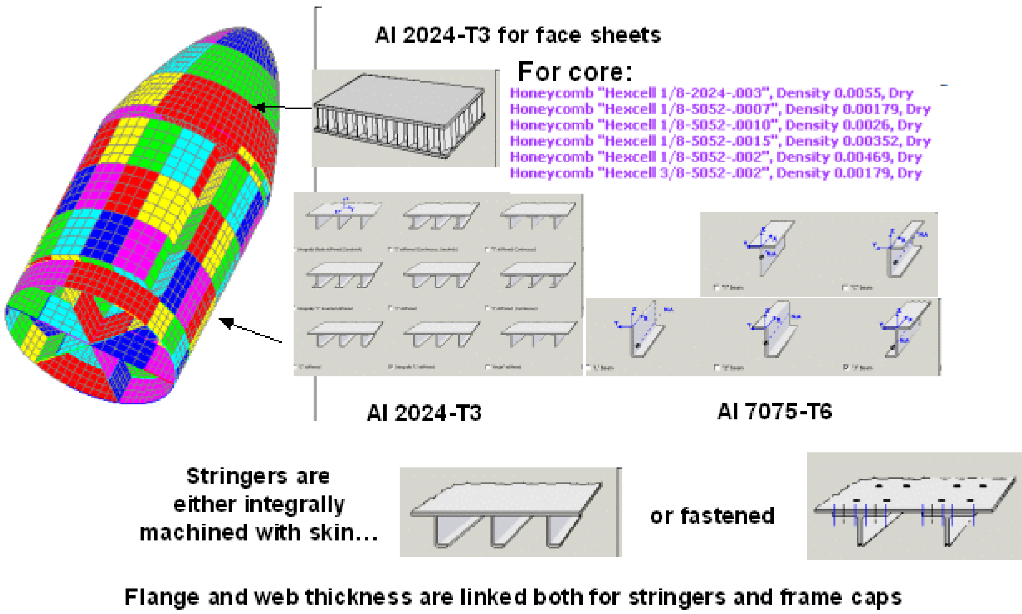

Figure 11. Initial metal semi-monocoque design

Table 1. Panel and beam design limits

Panel	Beam
Minimum skin thickness: • 0.02 inch for sandwich panel face sheets • 0.03 inch for stiffened panel skin	
Minimum sandwich panel core thickness = 0.5 inch	Minimum flange/web thickness = 0.04 inch
Minimum distance between stringers = 2.5 inch	Minimum flange width = 1 inch
Maximum combined height skin/stringer = 2 inch	Minimum beam height = 1 inch
Typical panel size: • 30 x 70 inch for Support structure • 43 x 70 inch for Pallet structure • 108 inch panel radius	

The first six design iterations resulted in a vehicle structural weight equal to 6,991 lbs when the weight change between the two successive iterations was less than 1%. The structural standardization process followed. Design variables of the most critically designed panel/beam were applied to a group of panels/beams and the vehicle was reanalyzed in HyperSizer without changing/optimizing the design variables. This process causes an increase in weight and re-distribution of structural mass. Therefore, the model had to be reanalyzed in I-Deas because of the revised inertial forces. Table 2 lists the weight convergence during structural standardization process.

Table 2. Weight convergence during structural standardization design process for metal semi-monocoque design

Design Iteration	Optimized Structural Weight (lbs)	Standardization Structural Weight (lbs)
6	6,991	12,544
7	7,294	11,182
8	7,179	11,182
9	7,199	11,039

Tables 3 and 4 summarize results of the 9th design iteration. Maximum dynamic pressure was the dominant load condition and global buckling was the dominant failure constraint. The failure mode weights column (Table 3) summarizes the component weights under three general groups of failure modes. The optimization program sums the weights of panels and beams whose controlling failure modes belong to one of these three groups. The standardization process almost doubled the beam weight and caused the average panel unit weight to increase from 1.27 lb/ft^2 to 1.78 lb/ft^2. The standardization reanalysis increased the average panel and beam thickness so that the final design (Table 4) had higher margins of safety. The final weight of 11,039 lb did not include lump weights of Common Berthing Mechanism and payload but included 554 lbs of the Nose Cone weight.

Table 3. Metal semi-monocoque design after the 9th iteration and before the final standardization run

Weight per Load Case (lb)		Beam Weights		Panel Weights		Failure Mode Weights (lb)	
Max q in y	2,948	Unit (lb/ft)	1.03	Unit (lb/ft^2)	1.27	Strength	1,099
Max q in z	2,060	Total (lb)	1,793	Total (lb)	5,406	Buckling	4,595
Max q in yz	1,313					Local Buckling	1,365
Max g	356					Min Gauge	140
Liftoff in y	191						
Liftoff in z	246						
Liftoff in yz	85						
Total Weight (lb)	7,199						

Table 4. Metal semi-monocoque design after the 9th iteration standardization reanalysis

Weight per Load Case (lb)		Beam Weights		Panel Weights	
Max q in y	4,494	Unit (lb/ft)	1.97	Unit (lb/ft^2)	1.78
Max q in z	3,561	Total (lb)	3,429	Total (lb)	7,610
Max q in yz	2,017				
Max g	598				
Liftoff in y	117				
Liftoff in z	205				
Liftoff in yz	47				
Total Weight (lb)	11,039				

After the 8th iteration, the panel and beam design was frozen and the same families of concepts were used in the following iterations as outlined in Section VII. Therefore, during the 9th iteration, HyperSizer optimized design variables such as thickness, stiffener spacing and size of the retained concepts of panels and beams. Figure 12 shows the effects of standardization that started after the 6th iteration. Distribution of optimal panel unit weights is shown on the left and redistribution of the unit weights after the standardization reanalysis is shown on the right. Figure 13 shows the retained panel concepts that were obtained after 8th iteration. This was the most weight efficient design and consisted of: integral inverted L section, integral inverted T section and fastened T section for the Pallets; integral blade, fastened I section, fastened angle and fastened Z section for the Support structure; honeycomb "Hexcell 3/8-5052-0.002" for the core of the sandwich panels of the Nose Cone.

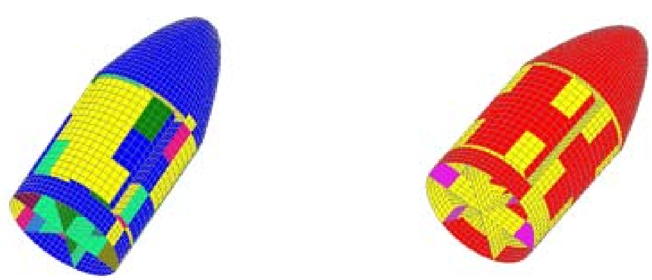

Figure 12. 6th Iteration panel unit weight distribution before and after the standardization run

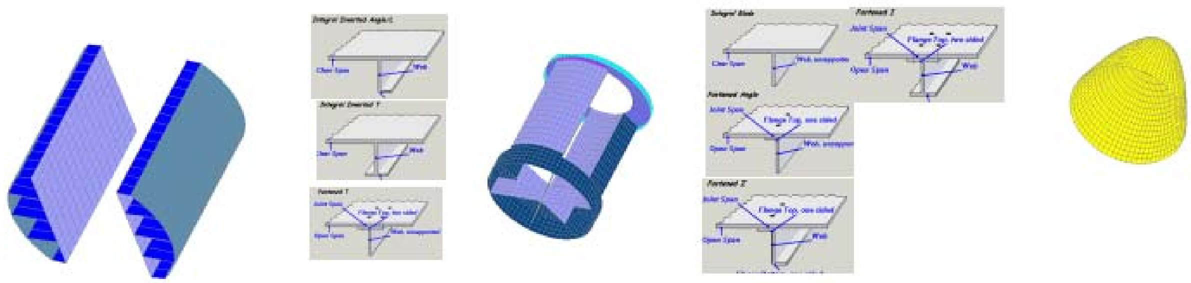

Figure 13. Vehicle design after the 8th iteration

B. Composite Monocoque Design

The second vehicle design was an all-composite design. For simplicity of part manufacturing this design was mostly an un-stiffened panel concept. Five different lamina layup patterns of Graphite/Epoxy IM7/977-2 are used and the optimizer had to select the one with the minimum weight (Fig. 14). Four Aramid/Phenolic composite cores were selected for Nose Cone sandwich panels as the starting design. The two major load-carrying bulkheads were modeled with bonded skin-stringer panels. Optimal cross section types of frame and longeron caps were retained from the aluminum design but using the new material, Graphite/Epoxy IM7/977-2, their size and thickness were subject to optimization.

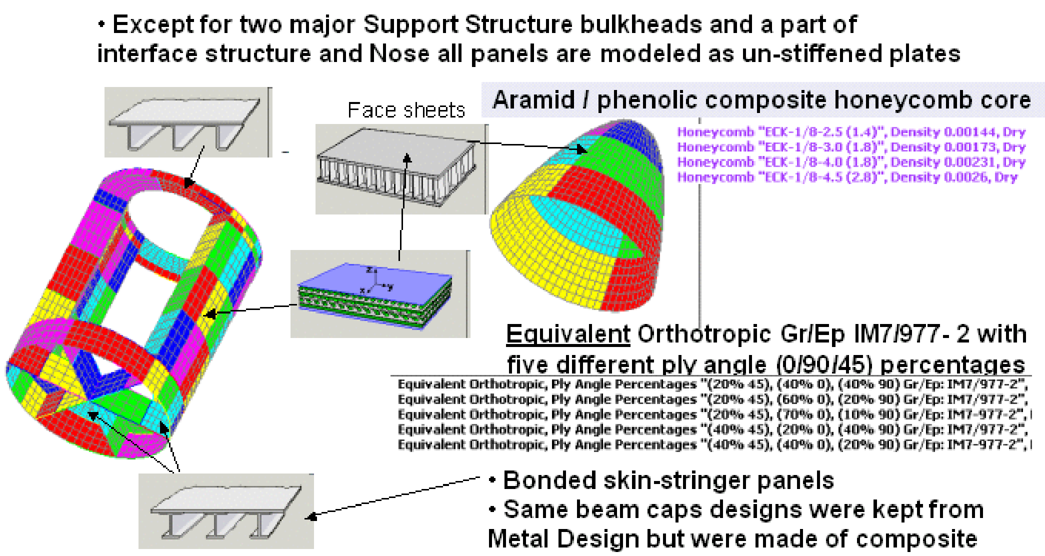

Figure 14. Un-stiffened composite panel design

The optimization process for this design started with the minimum weight design that was achieved after the 6th iteration of the all-metal design. As shown in Section II, only running loads obtained after the 6th iteration and the vehicle FEM geometry were imported from I-Deas into HyperSizer. Hypersizer uses only this information from I-Deas to select the optimum layup and panel thickness. Two more iterations produced a converged minimum weight design. After that, two iterations, coupled with the standardization process, produced a minimum weight design of 7,495 lbs, and standardization generated a final weight of 10,043 lbs. Controlling failure mode for most of the structure was global buckling. Maximum dynamic pressure was the dominant load condition. Tables 5 and 6 summarize results of the last iteration.

Table 5. Composite monocoque design before the final standardization run

Weight per Load Case (lb)		Beam Weights		Panel Weights		Failure Mode Weights (lb)	
Max q in y	3,221	Unit (lb/ft)	0.47	Unit (lb/ft^2)	1.57	Strength	193
Max q in z	2,413	Total (lb)	812	Total (lb)	6,683	Buckling	6,961
Max q in yz	833					Local Buckling	273
Max g	529					Min Gauge	68
Liftoff in y	46						
Liftoff in z	269						
Liftoff in yz	184						
Total Weight (lb)	7,495						

Table 6. Composite monocoque design after final standardization reanalysis

Weight per Load Case (lb)		Beam Weights		Panel Weights	
Max q in y	4,132	Unit (lb/ft)	0.96	Unit (lb/ft^2)	1,96
Max q in z	3,209	Total (lb)	1,669	Total (lb)	8,374
Max q in yz	1,218				
Max g	769				
Liftoff in y	84				
Liftoff in z	463				
Liftoff in yz	168				
Total Weight (lb)	10,043				

C. Composite Semi-Monocoque Design

A stiffened composite panel design was the third design evaluated. It consisted of the same stiffened panel and beam concepts obtained as in the optimal all-metal design, but the material was composite Graphite/Epoxy IM7/977-2 instead of aluminum (Fig. 15). Also stringers were bonded to the skin instead of fastened by rivets.

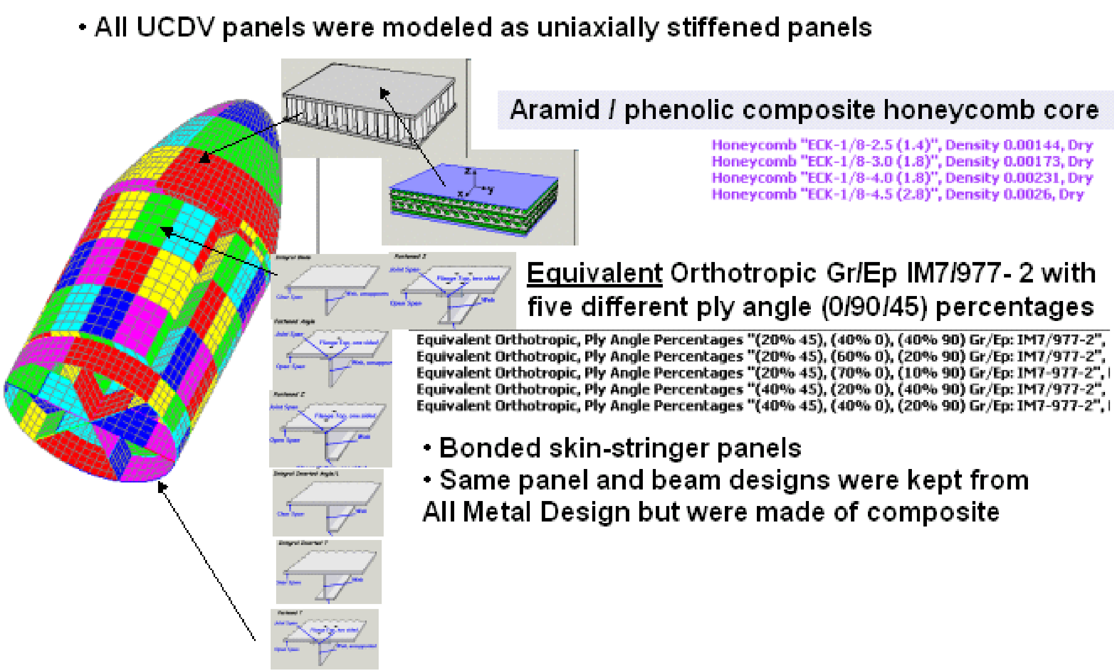

Figure 15. Stiffened composite panel design

The starting point for optimization was again the all-metal design reached after the 6th iteration. Because of the close proximity of the minimum weight design, the standardization process was initiated immediately and after two iterations achieved a weight of 3,961 lbs (Table 7). Final standardization produced a weight of 6,011 lbs (Table 8). This was a significant weight reduction when compared to the first two designs. It is worth noting that the average panel unit weight for of the minimum standardized design was reduced to 0.99 lb/ft^2 as compared to 1.78 lb/ft^2 of the aluminum design.

Table 7. Composite semi-monocoque design before final standardization reanalysis

Weight per Load Case (lb)		Beam Weights		Panel Weights		Failure Mode Weights (lb)	
Max q in y	1,575	Unit (lb/ft)	0.51	Unit (lb/ft^2)	0.72	Strength	353
Max q in z	1,333	Total (lb)	882	Total (lb)	3,079	Buckling	2,216
Max q in yz	626					Local Buckling	1,042
Max g	216					Min Gauge	350
Liftoff in y	58						
Liftoff in z	99						
Liftoff in yz	54						
Total Weight (lb)	3,961						

Table 8. Composite semi-monocoque design after final standardization reanalysis

Weight per Load Case (lb)		Beam Weights		Panel Weights	
Max q in y	2,579	Unit (lb/ft)	1.02	Unit (lb/ft^2)	0.99
Max q in z	2,033	Total (lb)	1,781	Total (lb)	4,230
Max q in yz	899				
Max g	352				
Liftoff in y	25				
Liftoff in z	109				
Liftoff in yz	14				
Total Weight (lb)	6,011				

Figures 16 and 17 show the summary of running loads. The units on the bar charts are lb/in for the longitudinal N_x and the circumferential N_y loads. These loads generally confirm the assumption about the magnitude of the load that was used in the preliminary study.

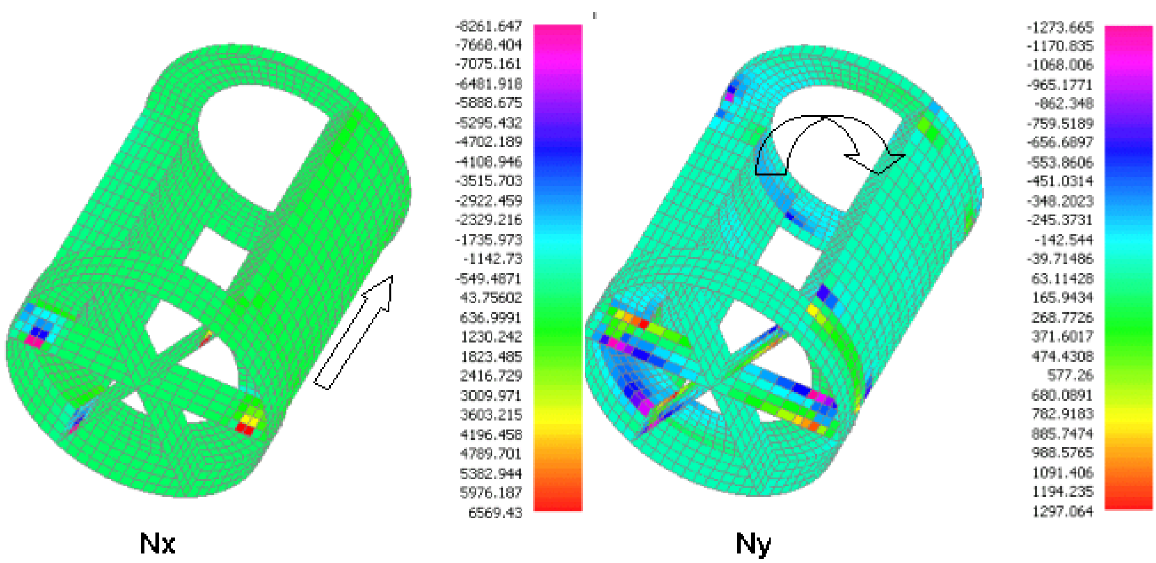

Figure 16. Running loads on stiffened composite panel design (lb/in) of Support structure

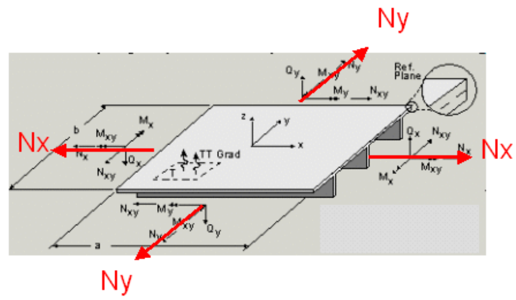

Figure 17. Direction of running loads on stiffened composite panel design of Support structure

D. Non optimum weight

The obtained structural weight is analytical and does not account for structural elements that are not modeled, such as fittings, panel edges, fillets and so on. This additional weight is treated as so-called “non-optimum” weight and is usually accounted for in the conceptual phase with a factor that varies for different panel concepts as shown by Eq. (2). It was assumed to be 15% of the analytical weight in this study (Table 9). This factor had relatively small value mostly because the standardization process already increased the structural weight to produce the symmetric vehicle and to cover for other modeling uncertainties. The 15% value was based on author’s unpublished work with application of this process on another vehicle and correlation of analytical weights of that vehicle with a detailed weight breakdown of an existing launch vehicle. It was obtained for uniaxial stiffened metallic panels and, for lack of better information, was adopted for composite panels in this study.

$$\text{As Built Weight} = \text{Analytical Weight x Non-optimum Factor} \quad (2)$$

Table 9. As built weight

Design	Analytical Weight (lb)	Non-optimum Factor	As Built Weight (lb)
Metal semi-monocoque	11,039	1.15	12,695
Composite monocoque	10,043	1.15	11,549
Composite semi-monocoque	6,011	1.15	6,913

E. Summary of results

Vehicle weight is summarized in Table 10 for the three designs and compared at the sub-assembly and assembly level as well as between analytical and “as-built” designs. The first four rows of results in Table 8 refer to the analytical weights of the Nose Cone, UCDV Pallet, Support structure and total analytical weight that is a sum of the three weights. Weight contribution in percent of total analytical weight of each of the sub-assemblies is shown in adjacent columns. The total as built weight is the total analytical weight increased by 15%. The CBM weight of 900 lbs was added to get structural weight of the Nose Cone and UCDV. The last row represents the minimum structural weight of the vehicle that the optimizer produced and shows how the final estimated weight almost doubles pure analytical calculation. The comparison between the three designs indicates a definite weight advantage of the composite design. This can only be achieved if the composite design uses the same semi-monocoque design as the aluminum panels. The advantages of the composite monocoque design over the aluminum semi-monocoque design are inconclusive for the seven load cases that were used in this study. This result confirms predictions reached in the preliminary study for un-stiffened panels.

There are two weights marked with an asterisk in Table 10 that illustrate effects of the user-in-the-loop in the standardization process. The first asterisk is placed near the weight of the Nose Cone of the composite semi-monocoque design. The weights of the Nose Cone for two composite designs are not the same although they should be the equal for identical load cases and design concepts. In this case, the designer allowed optimization to taper the core of the Nose sandwich panels for the monocoque vehicle and did not allow it in the semi-monocoque design. The second asterisk is over the minimum weight of composite monocoque design. Although that weight is larger by 300 lbs than the equivalent weight of the metal semi-monocoque design, total weight of the composite monocoque design, achieved by the standardization process, is smaller by 1,000 lbs than the equivalent metallic design weight. Tables 3 through 6 show that the standardization process almost equally doubled the weight of beams both for the metallic and the monocoque composite design because there were no changes in beam design concepts. Only beam material and size of cross sections were changed. Standardization increased the weight of metallic semi-monocoque panels by 41% and increased the weight of composite monocoque panels by only 25%. This unequal increase in panel weight is due to the fact that the standardization process is not a formal optimization process and affects different panel designs differently. The designer was more judicial in selecting the standardized design for the composite panel design and indicates certain arbitrariness in this last step of the process that may be improved with more strict application of HyperSizer capabilities.

Table 10. Summary of results

Sub-assembly	**Metal Semi-Monocoque**		**Composite Monocoque**		**Composite Semi-Monocoque**	
	W (lb)	% Total	W (lb)	% Total	W (lb)	% Total
Nose	554	5	321	3	343*	6
Pallet	3,656	33	3,659	37	1,915	32
Support	6,829	62	6,063	60	3,753	62
Total (analytical)	11,039	100	10,043	100	6,011	100
Total as built	12,695		11,549		6,913	
Total as built with CBM	13,595		12,449		7,813	
Min W Total (analytical)	7,199		7,495*		3,961	

F. Global Constraints

At the time of this study, HyperSizer did not have the capability to deal with the problem of global constraints. The vehicle was sized at the panel level and global constraints such as vehicle displacement or vehicle natural frequencies were not enforced. The designer would typically check if these constraints were not violated after the design was completed. In this study the two lowest global natural modes of the vehicle were analyzed for metallic and composite stiffened panel designs (Fig. 18).

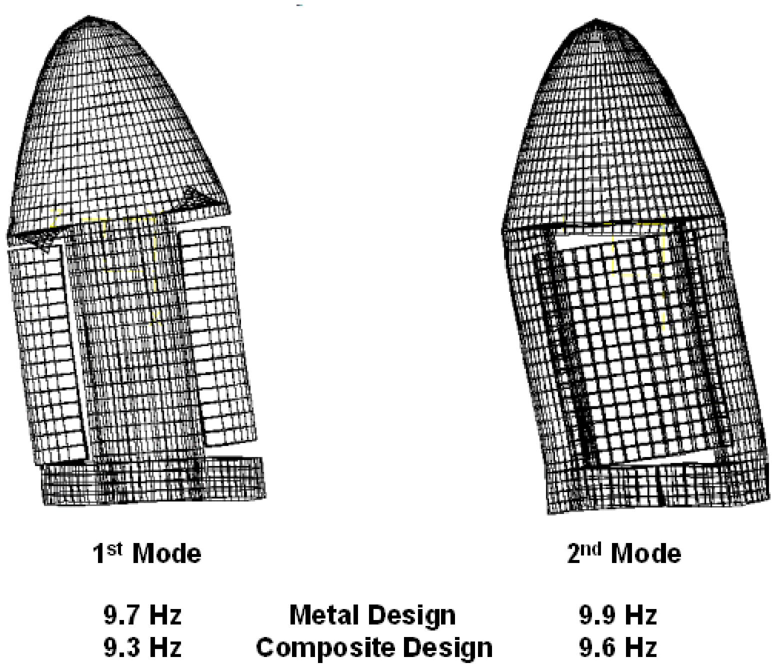

Figure 18. First two natural modes of vibration for two semi-monocoque designs

IX. Conclusions

Three design concepts of the Un-pressurized Cargo Delivery Vehicle were analyzed and their respective weights were compared. The composite semi-monocoque design had the minimum structural weight when compared with the other two designs. Use of un-stiffened composite panels showed that they did not produce weight improvement over the design based on stiffened aluminum panels for the selected load cases.

An earlier developed procedure for rapid weight estimation that was based on Finite Element Analysis and sizing of the vehicle by the use of commercially available codes was amended with new analysis iterations. The additional process increased the design weight to account for manufacturing and uncertainties in subsystems inertial load distribution. Added weight due to un-modeled structure was also accounted through the use of non-optimum weight factor that was based on previous studies. These additions almost doubled the weight of the theoretical optimal design and were based on weight estimates at the subassembly level that are more physically plausible than application of a general non-optimum factor.

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