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Structural Configuration Analysis of Crew Exploration Vehicle Concepts

V. Mukhopadhyay
NASA Langley Research Center, Hampton, VA

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Vivek Mukhopadhyay*
NASA Langley Research Center, Hampton, VA

Abstract

Structural configuration modeling and finite element analysis of crew exploration vehicle (CEV) concepts are presented. In the structural configuration design approach, parametric solid models of the pressurized shell and tanks are developed. The CEV internal cabin pressure is same as in the International Space Station (ISS) to enable docking with the ISS without an intermediate airlock. Effects of this internal pressure load on the stress distribution, factor of safety, mass and deflections are investigated. Uniform 7 mm thick skin shell, 5 mm thick shell with ribs and frames, and isogrid skin construction options are investigated. From this limited study, the isogrid construction appears to provide most strength/mass ratio. Initial finite element analysis results on the service module tanks are also presented. These rapid finite element analyses, stress and factor of safety distribution results are presented as a part of lessons learned and to build up a structural mass estimation and sizing database for future technology support. This rapid structural analysis process may also facilitate better definition of the vehicles and components for rapid prototyping. However, these structural analysis results are highly conceptual and exploratory in nature and do not reflect current configuration designs being conducted at the program level by NASA and industry.

I. Introduction

A comprehensive plan for the development of a lunar outpost was prepared by the Johnson Space Center Exploration study team[1] as early as in May 1992. In the recent Exploration and Space Architecture Study[2] (ESAS) report, details of the current space exploration strategy and technical approaches were outlined. Initial design specifications for the development of a Crew Exploration Vehicle (CEV) are also available in this report. The CEV internal cabin design pressure is specified to be the same as in the International Space Station (ISS) to enable docking with the ISS without an intermediate airlock[2,3]. In order to carry this pressure load in a conical shaped capsule without significant structural mass penalty[4,5], extensive structural analysis, design, fabrication and testing efforts would be required. With the current structural technology and design tools, there is opportunity for developing significantly efficient structural configurations for these vehicles. This paper will present some initial structural configuration modeling and analysis of a notional crew capsule inner shell and other pressurized components. First, some parametric solid models are developed[6] for visualization of vehicle configuration options. These parametric solid models are then used for the finite element analysis[7]. Uniform shell, isogrid plate and rib-spar constructions are investigated in order to examine their effects on the stress distribution, deflection, factor of safety distribution and mass. These results are presented as a part of lessons learned and to build up a structural sizing database for future technology support. However, these structural analysis results are highly conceptual and exploratory in nature and do not reflect the current configuration designs being conducted at the program level by NASA and the aerospace industry.

II. Structural Analysis Process

The initial assumptions for overall sizing and loads are as follows.

1. The CEV maximum outer diameter is 5.5 meters.
2. The CEV internal cabin pressure is 1.0139E+05 Newtons/meter^2 or 14.7 pounds/square inch (psi). The liquid oxygen tank internal pressure is 2.069E+05 N/m^2 (30 psi). The high pressure gas storage tanks carry 2.069E+06 N/m2 (300 psi) internal pressure.
3. Aluminum alloy material properties are used for the CEV crew cabin. For high pressure gas storage tanks, titanium alloy material properties are used. Thermal effects or external loads are not included at this stage.

The structural finite element analysis process steps are as follows.

1. Generate parametric solid models of the vehicle outer shell, and pressurized tank components.
2. Analyze solid and shell finite element models (FEM) of vehicle components and compute stress,

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*Research Engineer, Aero. Systems Analysis Branch/442, Associate fellow.

deflection and safety factors, based on the internal pressure load.

3. Resize solid or shell models, choose materials and skin thickness that would exhibit reasonable factor of safety based on the material yield stress.

4. Investigate packaging volume and interference issues of vehicles and internal components.

5. Show summary of results in Table 1.

6.Modify FEM models for future structural analysis with all external loads, for sizing, stability and optimization studies.

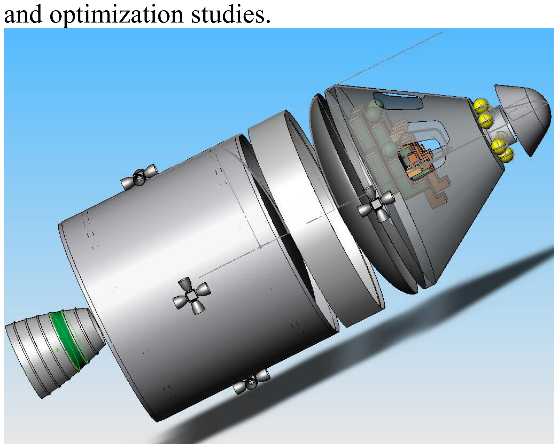

Figure 1. A conceptual crew exploration vehicle with crew capsule, inter-stage skirt, heat shield, service module and engine.

Figure 1 shows a modular crew exploration vehicle concept, with the service-module, inter-stage skirt and heat shield. The emergency escape rocket is not shown. This figure also shows six titanium tank with each capable of carrying 2.069E+6 Newtons/m^2 (300 psi) internal pressure with a factor of safety of 11. The tank outer diameter is 0.4 meters and the skin thickness is 3mm. Each tank has another integral 3mm thick 30 mm wide band around the circumference. This helps in containing additional pressure and serves as an anchor for attaching the tank to the vehicle. Mass of each spherical tank is 7.36 kg.

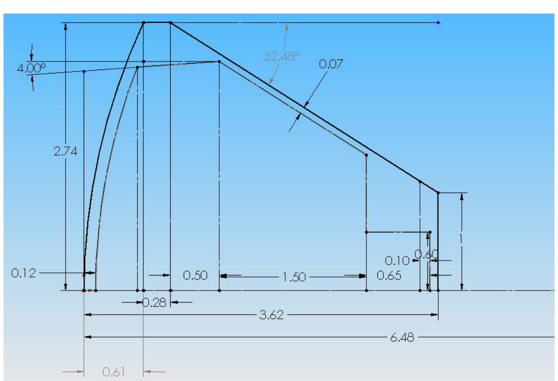

Figure 2. A notional command module section drawing.

The solid modeling tool[6] used for this analysis can also be used for computation of mass, moment of inertia, center of gravity location, etc. It can also be used for creating engineering drawing for prototype development. A sample drawing of the command module cross section is shown in Figure 2. The inner pressurized module is used for the structural analysis.

III. Inner Shell with Cutouts

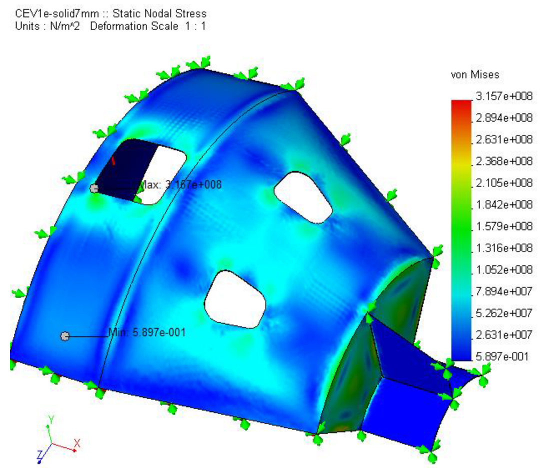

Figure 3. Stress distribution of the CEV with 7mm thick skin at 1.0139E+5 N/m^2 (14.7 psi) internal pressure using FEM analysis with solid 3D tetrahedral elements.

For the command module shell analysis, high strength Aluminum AL7050-T73651 alloy is used. This aluminum alloy has high fracture toughness. For very high pressure tank analysis, aluminum and Titanium 6AL4VA alloy material properties are used[6]. Material properties are shown in Table 2. Finite element model stress analysis was performed using SolidWorks[7] for solid models and with CosmosDesignStar[8] for both solid and shell models. First a solid model of the conical inner shell with cutouts for access hatch and windows is built. Using this solid model, a detailed stress analysis was performed. The internal pressure was assumed to be 1.0139E+5 N/m^2 (14.7 psi). The solid shell of revolution was assumed to have a 7 mm thick skin. For this material property and skin thickness, the structural mass, without cutouts, is 1014 kg. This first quadrant of the inner pressurized shell with cutouts was modeled with a dense mesh containing 41635 tetrahedral 3D elements. Fixed boundary condition was imposed on the front and rear ends and edges. From this finite element analysis, the maximum von Mises nodal stress was 3.16E+08 N/m^2 (45790 psi), as shown in Figure 3.

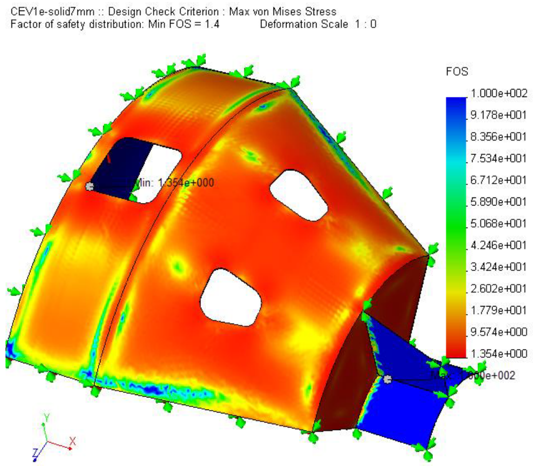

Figure 4. Factor of safety distribution of a 7 mm thick shell due to the 1.0139E+5 N/m^2 (14.7 psi) internal pressure load.

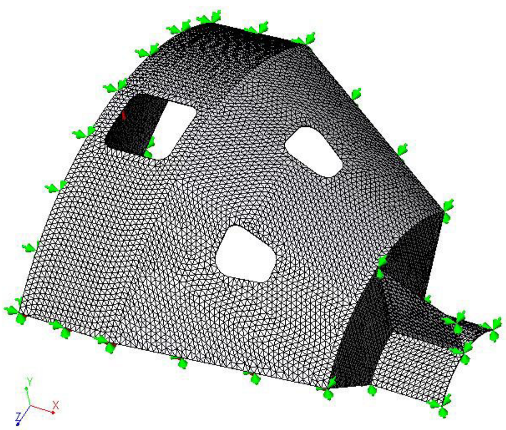

Figure 5. Solid finite element mesh model with 41635 tetrahedral 3D elements for the 7.0 mm thick conical inner shell with cutouts.

The maximum stress usually occurs at the edge of the cutouts, which are usually reinforced. The maximum factor of safety (FOS) defined as the ratio of material Yield stress/von Mises stress was computed to be 1.35. The FOS distribution is shown in Figure 4. The high density tetrahedral mesh is shown in Figure 5.

IV. Ribbed Shell

For a better understanding of the structural behavior under uniform internal pressure loading, a detailed solid model of a quadrant of the shell with longitudinal and radial ribs and stiffeners was developed and analyzed. Again AL7050-T73651 aluminum alloy was used for the shell and rib material. This analysis used a quadrant of the capsule with 0.005m thick skin and 0.7m wide 0.005m thick ribs at 10 degrees interval. Although the ribs are modeled as extending inside, they need to extend outside to fill the 0.7m gap between the inner pressurized shell and outer mold line. Then the ribs could be used to hold the thermal protection layer on the conical face. This could also prevent ribs buckling, which would generally be under tension from internal pressure load. The stress distribution is shown in Figure 6. The maximum von-Mises stress is 1.72E+08 N/m^2 (25000 psi) and the minimum factor of safety is 2.48.

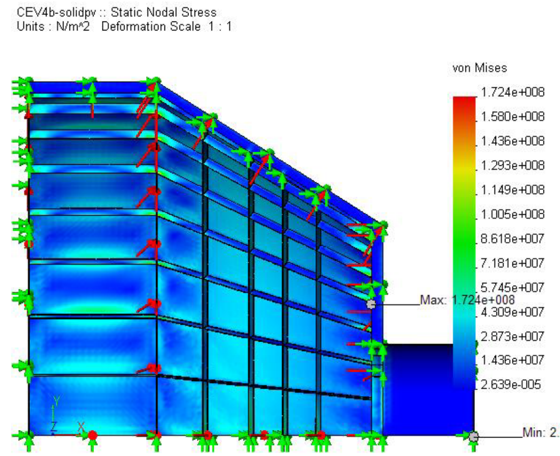

Figure 6. Von Mises stress distribution at element nodes due to the 1.0139E+5 N/m^2 (14.7 psi) internal pressure.

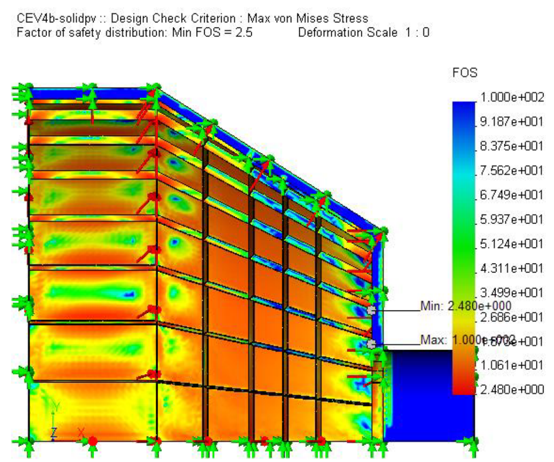

Figure 7. Factor of safety distribution due to the 1.0139E+5 N/m^2 (14.7 psi) internal pressure.

The factor of safety distribution due to the internal pressure is shown in Figure 7. Most of the high stresses occur at the junction of the cylindrical and conical shell, and at the flat front end. This end has four additional ribs support as shown in Figures 3-5. The total structural mass is 996 kg for the 4 quadrants (4x249 kg or 4x549 lbs), not including ribs for the back shell end. The back shell end is assumed to be fixed for all the FEM analyses boundary conditions. The maximum deflection under this boundary condition is 1.93 mm. Minimizing deflection is important in order to avoid de-bonding of the thermal protection material.

V. Isogrid Panel

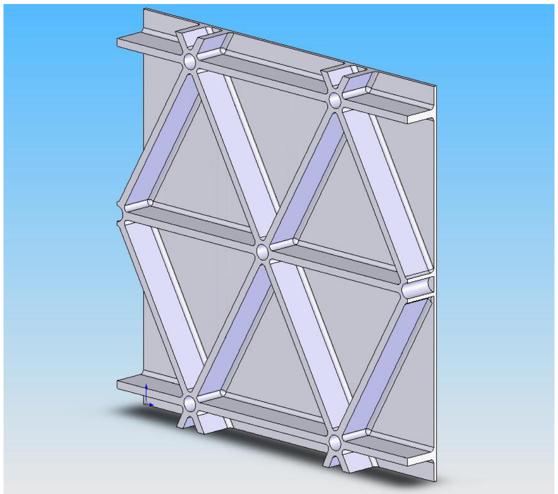

Figure 8. Isogrid solid model of a 25.4x25.4 cm square AL 7050 alloy panel with weight 1.02 kg.

For efficient load carrying capacity and mass reduction, isogrid structures are commonly used for launch vehicles. A solid model of an isogrid panel is shown in Figure 8. This Isogrid solid model represents a 25.4x25.4 cm (10 inch) square panel, with skin thickness 2.54mm (0.1 inch). The rib thickness of each of the machined isogrid is 5.08mm (0.2 inch). Total skin plus rib height is 25.4 mm (1 inch). The material is AL 7050 alloy. The panel weight is 1.0614 kg, with standard 2.54 mm (0.1 inch) fillet and 1 cm (0.4 inch) diameter drilled holes at each rib junction. Each side of the isogrid triangle has a length of 12.7 cm (5 inch). The specific mass per unit surface is 16.452 kg/meter^2. With this isogrid panel geometry and material, a panel specific mass translates into 16.452 kg/m^2 panel area.

Figure 9 shows von-Mises stress distribution on this isogrid panel due to the uniform 1.0139E+5 N/m^2 (14.7 psi) pressure, with the ideal boundary condition of two opposite edges -fixed, and other two edges -free. With this more detailed high fidelity finite element analysis with about 20000 solid tetrahedral elements, the local stress at the spar skin junction filet dominate the design. The maximum von Mises nodal stress is 8.233E+07 N/m^2 (11940 psi).

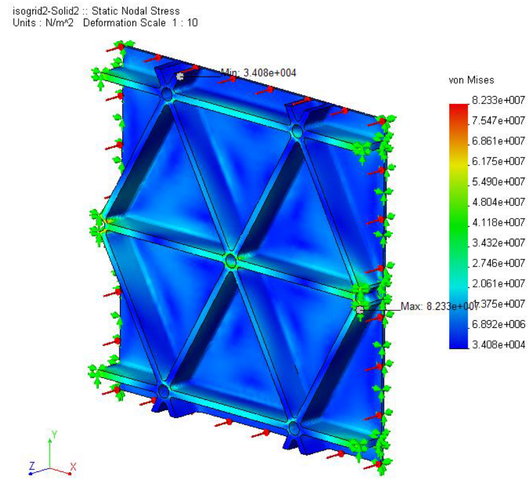

Figure 9. Von-Mises Stress distribution due to the 1.0139E+5 N/m^2 (14.7 psi) normal pressure.

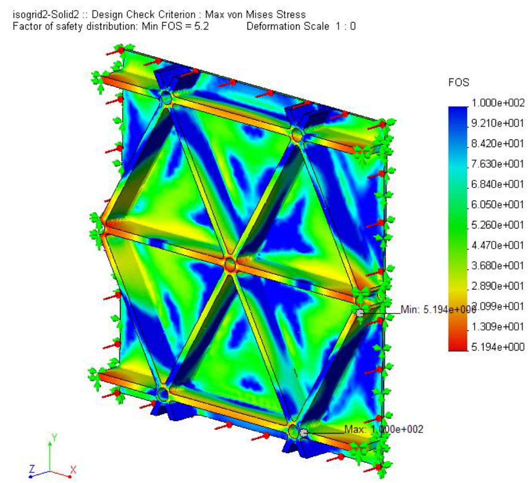

Figure 10. Factor of safety distribution due to the 1.0139E+5 N/m^2 (14.7 psi) normal pressure.

Figure 10 shows the factor of safety distribution based on the ratio of yield stress/von-Mises stress. The minimum factor of safety is 5.19. The inner shell has a 60 square meter surface area, and its mass would then be 987 kg with AL7050-T73651 material isogrid skin construction. Thus much higher structural stress factor of safely can be achieved for this higher internal design pressure load, without increasing the structural mass. However for greater accuracy, analytical and empirical buckling analysis of flat and curved cylindrical and spherical segments needs to be performed. An empirical procedure for buckling check and optimization under combined loading is described in Ref. 9.

VI. Pressurized Tanks

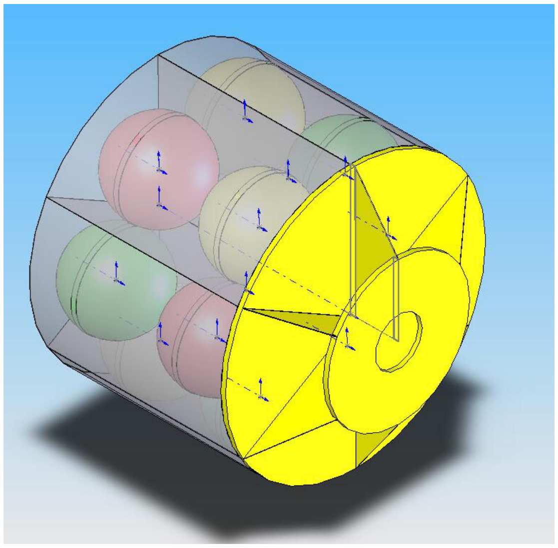

Figure 11 Partitioned Service modules with internal spherical tank assembly and docking adapter.

Preliminary results from the structural analysis of pressurized components of a service module concept is presented in this section. Figure 11 shows this service module concept consisting of six segments. Each segment houses two spherical tanks. These 12 spherical tanks are sized to make maximum use of the volume available within the 5.5 diameter outer shell. Each 1.8 meter diameter spherical tank is modeled with 3 mm thick skin made of titanium. Each tank has a retaining band for strength and attachment to main frame. The mass of each tank is 147.7 kg. Each tank can withstand an internal pressure of 30 psi, with a yield stress factor of safety of 3.8. Each spherical tank provides an internal volume of 3 cubic meters. The total structural weight is 5916 kg.

Figure 12 shows the solid model of an integral fuel tank concept. The tank is made of 20 mm thick sheet of aluminum with integral extension for inter-stage skirt attachment. Figure 13 shows the stress distribution and boundary conditions from the finite element analysis of a quadrant section of this tank. This integral tank can withstand an internal pressure of 2.069E+5 N/m2 (30 psi) with a minimum factor of safety of 5.5. The maximum von-Mises stress of this tank is 7.7E7 N/m^2 under this pressure load, and the prescribed boundary conditions. Since the minimum factor of safety is 5.5, and the desired factor of safety is 1.5, a redesign for mass reduction is required. The total internal volume of this tank is 142 cubic meters. The total structural mass is 7105 kg. However, a detailed design is necessary with all external loads and thermal effects.

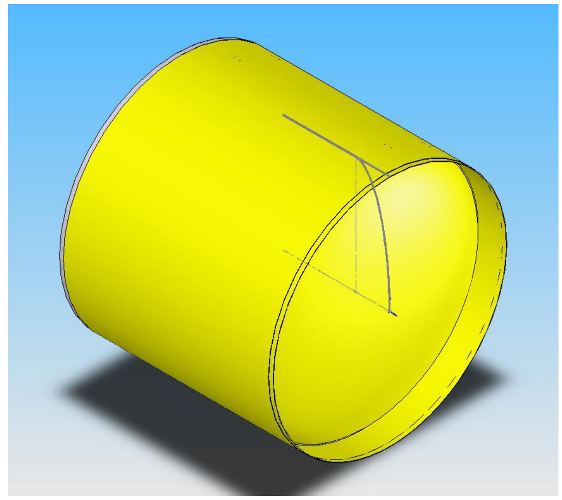

Figure 12. Integral fuel tank solid model.

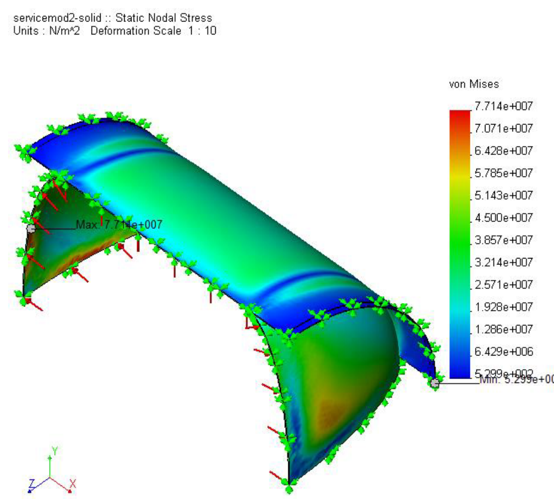

Figure 13. Von-Mises stress on the quadrant tank due to 2.069E+5 N/m2 (30 psi) internal pressure.

Conclusions

Structural analysis results of a notional crew exploration vehicle are presented. The effect of the internal cabin pressure load on the stress distribution, factor of safety, mass and deflections are investigated. Uniform 7 mm skin shell, 5mm thick shell with ribs and frames, and isogrid skin construction options are investigated. The isogrid construction appears to provide most strength/weight ratio for this limited study. Titanium construction for the very high pressure spherical tanks is recommended. However, additional sizing and optimization studies with all possible design loads are necessary.

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Table 1. Summary results from the set of models analyzed

model description analysis	element type	no of elements	load	BC	material	skin thickness	max VM stress		min FOS	max FOS	max deflection mm	mass
							psi	N/m2				
						conversion	1	6897.2286				
solid Shell with cutouts	3D tetrahedral	98000	14.7 psi	flat back & tunnel end fixed	AL 7050-T73651 plate	5 mm	186800	1.29E+09	0.33	8.6	267mm	1014 KG without cutouts
solid outer Shell without cutouts	3D tetrahedral	100000	14.7 psi	ends fixed	AL 7050-T73651 plate	5 mm	20000	1.38E+08	3.9	12	2.2mm	1014 kg
surface outer Shell without cutouts	2D Triangular	6866	14.7 psi	ends fixed	AL 7050-T73651 plate	5 mm thick skin	14590	1.01E+08	7.9	20	7 mm	1014 KG without cutouts
isogrid panel	3D tetrahedral	16047		two edge fixed two edge free	AL 7050-T73651 plate	2.5mm skin 0.5mm web 25.4 cm skin+web	11940	8.24E+07	5.19	36	0.018 mm	16.452kg/msq, 987 kg for 60 sqm surface area kg/sqm
solid			300 psi	fixed mid band	Ti-6AL4VA	0.4m dia 3mm thick	14100 psi	9.73E+07 N/m2	11 mm		0.1 mm	7.36kg
solid quadrant, new geom, inner shell	3D tetrahedral	with new cutouts	14.7 psi	ends&edges fixed	AL 7050-T73651 plate	7 mm thick skin	42730 psi	2.95E+08 N/m2	1.5	18	12.8 mn	1107 kg 4quads
solid quadrant, new geom, inner shell	41635 elements - 3D tetrahedral	with new cutouts & corner ribs	14.7 psi	tunnel ends, corner rib & edges fixed	AL 7050-T73651 plate	7 mm thick skin & tunnel rib	45790 psi	3.16E+08 N/m2	1.35 FOS	18	20 mm	1124.22 kg 4 quad
outer shell-flat end 700 iteration solver 10 min in20gH PC	3D tetrahedral 336000 dof	109198 elements, old outer OML	14.7 psi	tunnel ends, & edges fixed	AL 7050-T73651 plate	5 mm skin, 0.7mm x 5 mm ribs at 10 deg	21910	1.51E+08	3.2	19	3 mm	1309.52 kg 4 quad
solid quadrant, new geom, inner shell	100000 -3D tetrahedral, 330000 dof	flat end (no ribs) , new goem inner shell	14.7 psi	tunnel ends, & edges fixed	AL 7050-T73651 plate	5 mm skin, 0.7mm x 5 mm ribs at 10 deg	25000	1.72E+08	2.48	18	1.93 mn	996 kg, 4 quad

Table 2 Material properties:

material	type	prop	propsymbol	unit	psi	kgf/cm/cm	N/m2 (pascal)
AL7050-T73651	metal	Ex	EX		1.03000E+07	7.24174E+05	7.10415E+10
plate		Nu	NUXY	0.33 psi conv	1	7.030814E-02	6.897229E+03
		G	GXY		3.87218E+06	2.72246E+05	2.67073E+10
		dens	DENS	lb/cuin	0.10200	2.82340E+03	kg/mcube
		ult ten	SIGXT		7.10000E+04	4.99188E+03	4.89703E+08
		ult comp	SIGXC		6.00000E+04	4.21849E+03	4.13834E+08
		sigyield	SIGYLD		6.20000E+04	4.35910E+03	4.27628E+08
AL-6061 T651	metal	Ex	EX		9.90E+06	6.96051E+05	6.82826E+10
plate		Nu	NUXY	0.33 convers	14.7	1.033530	101389.260339
		G	GXY		3.72180E+06	2.61673E+05	2.56701E+10
		dens	DENS	lb/cuin	0.098		
		ult ten	SIGXT		4.20000E+04	2.95294E+03	2.89684E+08
		ult comp	SIGXC		3.50000E+04	2.46078E+03	2.41403E+08
		sigyield	SIGYLD		3.60000E+04	2.53109E+03	2.48300E+08
Ti6AL4VA	metal	Ex	EX		16000000	1.12493E+06	1.10356E+11
plate		Nu	NUXY	0.33 convers	30	2.109244	206916.857834
		G	GXY		6.01504E+06	4.22906E+05	4.14871E+10
		dens	DENS	lb/cuin	0.16		
		ult ten	SIGXT		160000	1.12493E+04	1.10356E+09
		ult comp	SIGXC		145000	1.01947E+04	1.00010E+09
		sigyield	SIGYLD		150000	1.05462E+04	1.03458E+09