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TITANIUM-LINED, CARBON COMPOSITE OVERWRAPPED PRESSURE VESSEL

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ABSTRACT

The design, manufacture, and test of a high performance composite overwrapped pressure vessel (COPV) utilizing a titanium liner is described. The vessel is unique in that the liner and end fittings are fabricated from commercially pure (cp) titanium with the attendant advantages of high corrosion and oxidation resistance, good low cycle fatigue characteristics (for pressure cycling) and good high cycle fatigue characteristics (for environmental testing). Liner manufacturing costs are reduced relative to other titanium alloys because cp titanium sheet is readily formed at room temperature and is easily weldable. The vessel is overwrapped with high performance carbon fiber impregnated with an epoxy resin. The tank was designed as a helium pressurant supply tank for a commercial communications satellite and has demonstrated P_bV/W ratio of approximately 1.5 x 10^6 inches. Topics covered include a brief overview of COPV technology, trade studies, tank design and analysis, performance summary, fabrication, qualification, and test results.

INTRODUCTION

Pressurized systems are required on many spacecraft in order to operate fluid management and propulsion systems. High performance composite overwrapped pressure vessels (COPVs) have been utilized in the aerospace and automotive industries for many years, providing an inherently safe, lightweight and cost effective storage source for pressurized fluids.

In a typical COPV design, an ultra-thin metallic liner is overwrapped with a reinforcing fiber and epoxy matrix. For typical operating pressures, this combination offers tremendous weight savings over equivalent monolithic metal vessels.

Low cycle fatigue characteristics are important for liner material selection because the liner plastically yields during each pressure cycle. Liner material ultimate strength and high cycle fatigue characteristics are equally important because the port openings and attachments are subjected to the full pressure load and external acceleration loads. Typical COPV's utilize 5086 and 6061-T6 aluminum as liner materials of choice. Recent research at Pressure Systems, Inc. (PSI) and Kaiser Compositek Corp. (KCC) has shown that commercially pure (cp) titanium specifically CP-3[*] for the liner and CP-70[†] for the port openings and attachments can serve as an excellent liner, eliminating many of the manufacturing and process problems associated with aluminum while improving the strength and fatigue properties of the liner.

This paper describes the design, manufacture, and test of a new COPV utilizing a cp titanium liner, Torayca T1000[‡] carbon fiber, Epon 826[§] epoxy resin, and a toughened epoxy film adhesive. This vessel is unique in its utilization of a titanium liner, with the attendant advantages of good low cycle fatigue characteristics (for pressure cycling), good high cycle fatigue characteristics (for vibration loading of end attachments), and high corrosion and oxidation resistance.

Figure 1 shows the basic configuration of the pressure vessel. The tank is intended to store approximately 6 lbs of helium at a maximum expected operating pressure (MEOP) of 4500 psi at 140°F. Proof pressure is 5625 psi and minimum burst pressure is 6750 psi.

[*] CP-3 Ti per MIL-T-9046J - 40 ksi yield strength
[†] CP-70 Ti per MIL-T-9047G - 70 ksi yield strength
[‡] T1000 carbon fiber is a product of Toray Industries, Inc., Tokyo, Japan.
[§] Epon 826 resin is a product of Shell Oil Co., Houston, Texas.

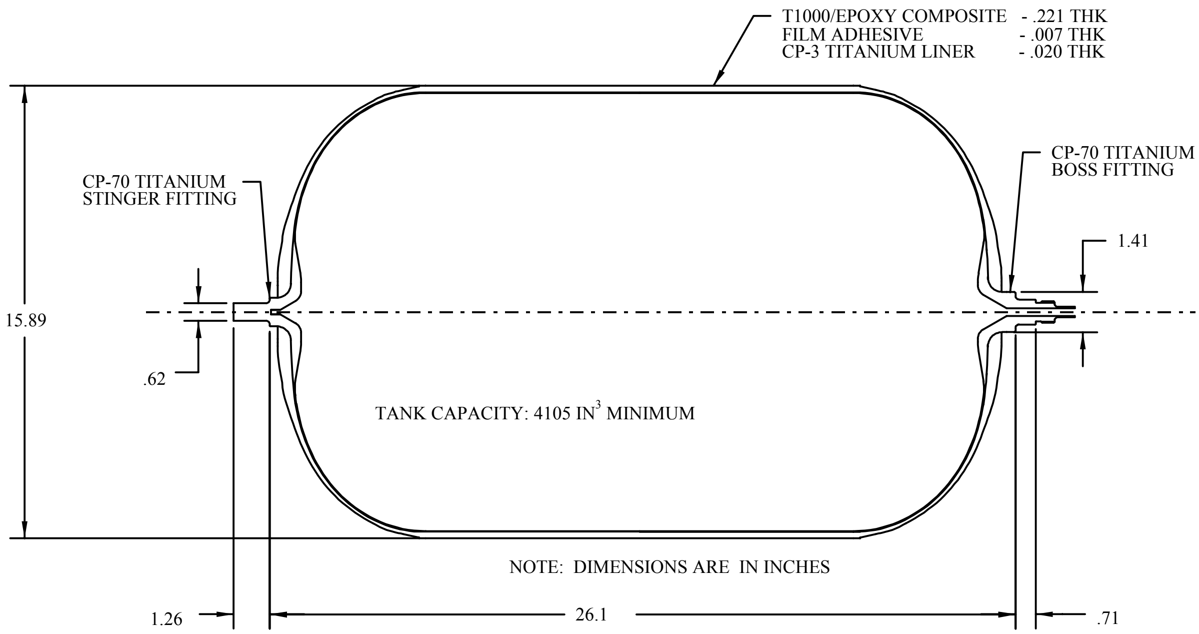

FIGURE 1. TITANIUM LINED COMPOSITE OVERWRAPPED PRESSURE VESSEL

TRADE STUDIES

Trade studies were performed to select the liner and composite overwrap materials. Since minimum weight was a primary design goal, an ultra-thin metallic liner overwrapped with a high strength carbon/epoxy composite was selected as the baseline design. The metallic liner serves as a pressure barrier while the more weight efficient carbon/epoxy carries the pressure membrane loads.

Choice of the carbon fiber was straightforward because a high specific tensile strength (fiber tensile strength ÷ fiber density) would produce the lowest weight composite overwrap which could accommodate the pressure membrane loads. A commercially available carbon fiber with a high specific tensile strength is Torayca T1000. T1000 has a typical tensile strength in excess of 800 ksi, a modulus of 42 msi and a density of 0.065 lb/in^3.

Liner material trades focused on three main categories: technical factors, cost, and risk parameters. The technical tradeoffs compared manufacturing process such as formability and welding, material compatibility, as well as strength and fatigue characteristics. Cost trades evaluated different manufacturing techniques gathered from experienced liner vendors. The risk analysis highlighted potential problem areas such as welding, corrosion, and contamination. The study focused on the three common materials used in aerospace pressure vessels; cp titanium (6Al-4V titanium was not considered a viable candidate for a non-load sharing liner nor was it considered cost effective), 6061-T6 aluminum, and 301 stainless steel. 301 stainless steel was eliminated immediately because its density (0.29 lb/in^3) was considered too high relative to aluminum (0.10 lb/in^3) and titanium (0.16 lb/in^3). The basic design relies on a minimum gage liner and the weight penalty of the relatively dense steel was considered too great.

The study concluded that cp titanium was the material of choice due to several factors:

1. Titanium exhibits excellent corrosion and oxidation resistance over the typical operating temperature ranges.
2. Titanium is less susceptible to stress corrosion cracking. Aluminum on the other hand is often found to exhibit pitting and stress corrosion particularly when subjected to severe cold forming, chemical machining, and prolonged exposure to test fluids.
3. Although aluminum is relatively inexpensive with good machinability, most liner configurations necessitate some welding. Aerospace quality welds in aluminum pressure vessels are difficult to achieve especially in thin wall cross-sections. Titanium on the other hand is readily weldable with lower inherent porosity.
4. Titanium has a lower galvanic potential with carbon filaments than aluminum and need not be isolated from the composite.
5. Titanium has a higher strength to weight ratio.

6. CP titanium is readily cold formable and can be economically manufactured into a very uniform thin wall liner.
7. Titanium has excellent low and high cycle fatigue characteristics

A substantial amount of analyses was focused on Item 7. A detailed assessment of low cycle fatigue performance for 6061-T6 aluminum, CP-3 titanium, and CP-70 titanium was made. Table 1 shows the relevant data used in the assessment. A geometric and material non-linear axisymmetric finite element model was used to model the vessel. Orthotropic properties for each composite layer were assigned to corresponding elements in the model. Elastic / plastic material properties for each candidate liner material were assigned to the liner elements. For each liner material, the nominal membrane strains and plastic strains were computed. As expected, the membrane strains were essentially the same for all liner candidates because the composite wrap carries nearly all the pressure membrane loads. The liner plastic strain magnitude ε^{PEQ} , however, was different for each liner material becuase of differing yield strength and elastic moduli.

Using the liner plastic strain magnitudes from the finite element analysis, the expected number of proof or operation cycles for each liner material was estimated using the Coffin-Manson Law[1]:

$$N = {}^{1}/_{4} \left[\varepsilon_F / \Delta\varepsilon^{PEQ} \right]^2$$

Where:

N= number of pressure cycles

ε_F = the minimum fracture strain of the metal liner = ln [1 / (1– RA)]

RA = the minimum reduction in area of the metal liner

$\Delta\varepsilon^{PEQ}$ = the change in plastic strain magnitude during a pressure cycle

The expected number of cycles was computed for both the nominal strains in the cylindrical section and for the peak strains near the cylinder / dome transition. Table 1 summarizes the estimated number of proof or operational cycles for each liner material.

Figures 2 and 3 summarize the percent cycles to failure for each candidate liner material for a typical spacecraft cycle life requirement of 8 proof cycles and 50 operation cycles. The percent cycles to failure is computed by a simple Minor's rule approach as shown below:

$$PCTF = \Sigma \left(n_i / N_i \right) \times 100\ \%$$

Where:

PCTF = Percent of cycles to failure

n_i = Number of required pressure cycles for pressure cycle P_i

N_i = Number of pressure cycles to failure for pressure cycle P_i

P_i = Pressure cycle -

i = 1 : proof pressure cycle

i = 2 : operation pressure cycle

Figure 2 shows that all the candidate liner materials have large margins on cycle life for the nominal strains calculated in the cylindrical section of the vessel. For example, only 15% of the low cycle fatigue life of 6061-T6 aluminum has been used after 8 proof cycles and 50 operating cycles. Figure 3 shows CP-3 and CP-70 titanium exhibit superior low cycle fatigue performance relative to 6061-T6 aluminum in areas of peak strain in the vessel.

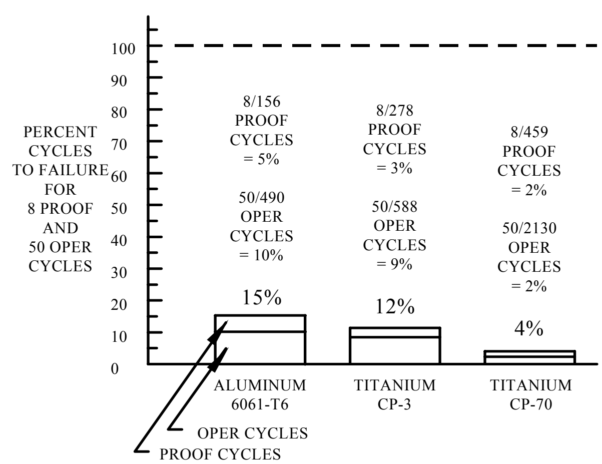

FIGURE 2. LOW CYCLE FATIGUE LIFE NOMINAL STRAINS

TABLE 1. MATERIAL TRADE STUDY - TANK LINER

	ALUMINUM 6061-T6	CP TITANIUM CP-3	CP TITANIUM CP-70
MATERIAL PROPERTIES			
YIELD STRENGTH - KSI	41.5	40.0	70.0
ULTIMATE STRENGTH - KSI	47.0	50.0	80.0
ELASTIC MODULUS - MSI	10.0	15.5	15.5
PLASTIC MODULUS - MSI	0.076	0.178	0.188
MIN ELONGATION - e_F - % *	10	20	15
MIN REDUCTION OF AREA - RA - % *	24	38	30
MIN FRACTURE STRAIN - ε_F - % *	27	48	36
MEMBRANE STRAINS - NOMINAL			
ε_{HOOP} - PROOF PRESS - %	1.35	1.34	1.32
ε_{LONG} - PROOF PRESS - %	0.86	0.85	0.83
ε_{HOOP} - ZERO PRESS - %	0.03	0.03	0.05
ε_{LONG} - ZERO PRESS - %	0.03	0.03	0.05
ε_{HOOP} - OPER PRESS - %	1.07	1.06	1.04
ε_{LONG} - OPER PRESS - %	0.68	0.66	0.65
LINER PLASTIC STRAIN MAGNITUDE - NOMINAL			
ε^{PEQ} - PROOF PRESS - %	1.69	1.85	1.54
ε^{PEQ} - ZERO PRESS - %	0.61	0.41	0.70
ε^{PEQ} - OPER PRESS - %	1.22	1.40	1.09
LINER Δ PLASTIC STRAIN MAGNITUDE - NOMINAL			
$\Delta\varepsilon^{PEQ}$ - 0 to PROOF PRESS - NOMINAL - %	1.08	1.44	0.84
$\Delta\varepsilon^{PEQ}$ - 0 to OPER PRESS - NOMINAL - %	0.61	0.99	0.39
LINER Δ PLASTIC STRAIN MAGNITUDE - PEAK			
$\Delta\varepsilon^{PEQ}$ - 0 to PROOF PRESS - PEAK - %	2.91	3.15	1.93
$\Delta\varepsilon^{PEQ}$ - 0 to OPER PRESS - PEAK - %	1.90	2.20	1.06
PREDICTED LINER CYCLES - NOMINAL STRAINS			
0 to PROOF PRESS	156	278	459
0 to OPER PRESS	490	588	2130
PREDICTED LINER CYCLES - PEAK STRAINS			
0 to PROOF PRESS	22	58	87
0 to OPER PRESS	50	119	288
* Specimen thickness 0.02 to .25 inch			

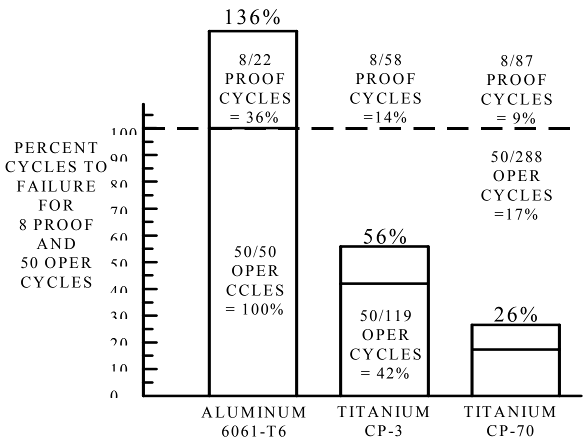

FIGURE 3. LOW CYCLE FATIGUE LIFE PEAK STRAINS

TANK DESIGN AND ANALYSIS

Tank Requirements

Table 2 shows the basic tank requirements.

TABLE 2. TANK DESIGN REQUIREMENTS

CHARACTERISTIC	REQUIREMENT
Max Operating Pressure	4500 psig 50 cycles
Proof Pressure	5625 psig 8 cycles (design) 4 cycles (qual)
Min Burst Pressure	6750 psig
Pressurant	5.8 - 6.6 lbm He
Size	16.6 in dia x 26.62 in long
Tank Weight	29.5 lbm max
Tank Capacity	4105 in^3 min
Shell Leakage	1.0 x 10^{-6} scc/sec @MEOP
Limit Acceleration	50 g each axis
Random Vibration	18.1 g rms 120 sec / axis 50 g peak response
Operating Temperature	-140°F to 140°F

The tank is intended to store approximately 6 lbs of helium at a maximum expected operating pressure (MEOP) of 4500 psi at 140°F. In addition to pressure cycling requirements, the pressurized tank must be capable of withstanding environmental loadings including 50 g quasi static acceleration, sinusoidal vibration, and random vibration. Of these environmental loading conditions, the random vibration requirement was the most critical. Random vibration response was limited to 50 g peak at the first major mode frequency, thus for design purposes an alternating 50 g quasi static fatigue load could be assumed for design of the stinger and port end attachments. Given that the first resonant mode of the vessel was in excess of 400 Hz, and the random vibration duration was 120 seconds each axis, 118000 cycles of 50 g quasi static acceleration loading was conservatively assumed for design of the stinger and port attachments.

Structural Configuration

Figure 1 shows the basic configuration of the pressure vessel. The tank is mounted to the spacecraft using the ported boss fitting and the unported stinger fitting. The ported boss fitting supports the tank in all three directions and incorporates a flat to prevent rotation about the tank axis. The stinger fitting provides support about the two directions normal to the tank axis while allowing axial growth of the tank during pressurization.

Table 3. shows the weight breakdown of the vessel.

TABLE 3. VESSEL WEIGHT SUMMARY

	MATERIAL	WEIGHT LB
LINER	CP-3 Ti	4.93
PORTED BOSS	CP-70 Ti	1.09
STINGER BOSS	CP-70 Ti	0.96
FILM ADHESIVE	TOUGHENED EPOXY	0.43
FIBER	T1000	9.88
EPOXY	EPON 826	4.69
	TOTAL	21.98

Materials

Torayca T1000 carbon fiber was the fiber reinforcement for the vessel. T1000 has a typical tensile strength in excess of 800 ksi, a modulus of 42 msi and a density of 0.065 lb/in^3. Table 4 shows good correlation between predicted and observed burst pressure demonstrating consistent and predictable material performance.

TABLE 4. PRESSURE VESSEL BURST DATA

TEST VESSEL	VESSEL WEIGHT LB	PREDICTED MINIMUM BURST	PREDICTED TYPICAL BURST PRESSURE	MEASURED BURST PRESSURE	PERFORMANCE FACTOR PV/W (V=4105 IN^3)
Development Design “Heavy” Wrap 0.032 in thick Liner	25.8 LB 26.0 LB	7810 PSI	8540 PSI	8500 PSI 9300 PSI	1.35×10^6 IN 1.47×10^6 IN
Development Design “Light” Wrap 0.32 in thick Liner	24.5	7152 PSI	7833 PSI	7850 PSI	1.32×10^6 IN
Qualification Unit “Light” Wrap 0.020 in thick Liner	22.0	7082 PSI	7750 PSI	7880 PSI	1.47×10^6 IN

An epoxy resin system consisting of Epon 826 epoxy resin and a proprietary curing agent formulation was used for the wet winding operation. The cured resin system has a tensile strength in excess of 10,000 psi, a modulus of 0.30 msi, a density of 0.042 lb/in^3, an elongation in excess of 7%, and a glass transition temperature of 210°F.

Film adhesive was applied to the liner prior to filament winding of the composite in order to insure that the liner was securely bonded to the composite. The adhesive has a non-liner shear stress/strain behavior. For purposes of analysis, the adhesive was idealized as a 0.007 inch thick isotropic layer with a shear yield stress of 6.1 ksi, a shear modulus of 0.13 msi, and a plastic shear modulus of 0.2 msi.

The liner consists of CP-3 titanium for the basic shell and CP-70 titanium in the boss and stinger regions. Material properties of the titanium are given in Table 1. A modified Goodman diagram was constructed from data obtained from Warlaw et al[2] for runout (> 10^6 cycles) and is shown in Figure 4. An additional modified Goodman diagram for finite life of 118,000 cycles (corresponding to the vibration requirement) was also constructed based on the procedures recommended by Shigley[3]. This modified Goodman diagram is also shown in Figure 4.

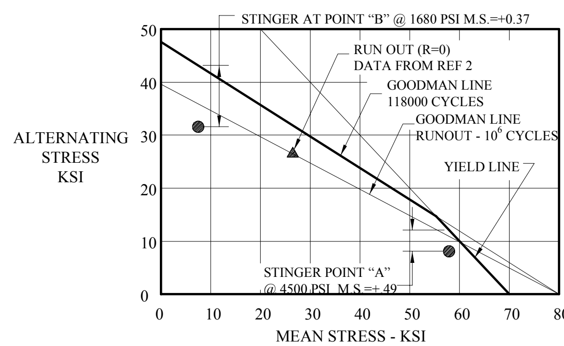

FIGURE 4. ESTIMATED FINITE FATIGUE LIFE OF CP-70 TITANIUM

ANALYSIS SUMMARY

Analysis of the pressure vessel consisted of 1) a geometric and material nonlinear axisymmetric finite element analysis for pressure loading, 2) a pressure stiffened shell finite element analysis for vibration response, and 3) a conventional bending and fatigue analysis of the stinger and port attachments.

Figure 5 shows the axisymmetric model used to estimate the response of the vessel to pressure loading. The finite element program used was ABAQUS[4] V4.9.

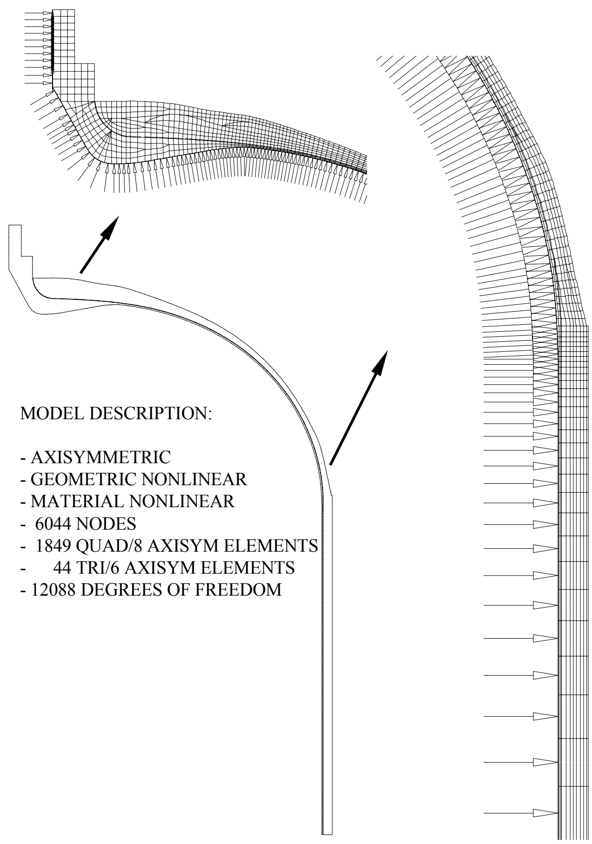

FIGURE 5. NONLINEAR AXISYMMETRIC FINITE ELEMENT MODEL

Results from the geometric/material nonlinear analysis of the vessel are shown in Figures 6 and 7 and in Table 5. Figure 6 shows the deformed shape of the vessel at proof pressure, and Table 5 shows that measured longitudinal tank deflection agreed well with analysis predictions. Figure 7 shows the change in plastic strain magnitude $\Delta\varepsilon^{PEQ}$ going from proof pressure to zero pressure. Nominal and peak values of $\Delta\varepsilon^{PEQ}$ were used in the trade study described earlier.

	PRESSURE PSI	AXIAL DISPLACEMENT INCH	
		PREDICTED	MEASURED†
PROOF	5625	0.422	0.436
ZERO	0	-0.104	-0.150
OPERATION	4500	0.342	0.355*
MIN BURST	6750	0.484	0.494*

*0.355 =0.505− 0.150 (Total less permanent set from autofrettage)
**0.494=0.644−0.150 (Total less permanent set from autofrettage)
† Qualification Vessel S/N0003Q

TABLE 5. VESSEL AXIAL DISPLACEMENTS

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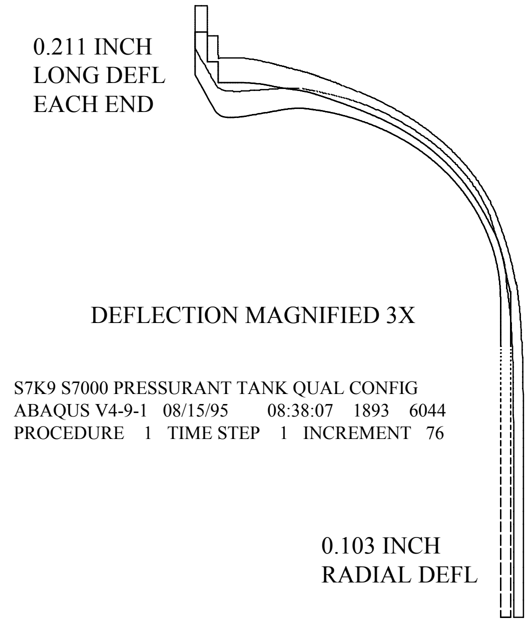

PREDICTED VS MEASURED
FIGURE 6. VESSEL DEFORMED GEOMETRY AT PROOF PRESSURE

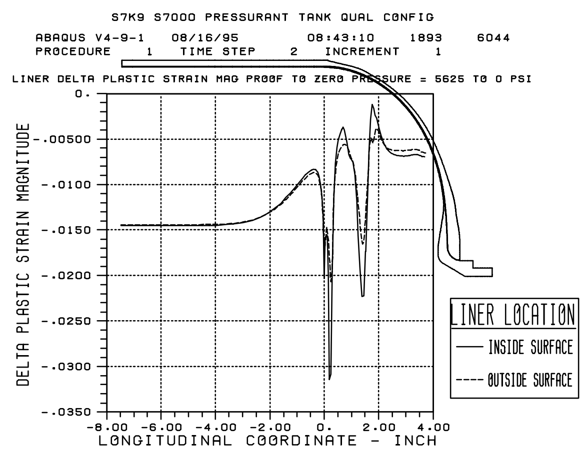

FIGURE 7 . LINER DELTA PLASTIC STRAIN MAGNITUDE PROOF PRESSURE TO ZERO PRESSURE

Results from the pressure stiffened shell finite element analysis is shown in Figure 8. The vessel was modeled with four noded quadrilateral and three noded triangular orthotropic shell elements representative of the composite wrap layup and liner. The elastic/plastic nature of the liner was not modeled because the vessel vibrational strain magnitudes were expected to be small and linear about a stationary strain state at operating pressure.

The total mass of the vessel was 29.7 lb including 6.7 lb of helium at 4500 psi. The additional mass of the helium was added as non-structural mass to the shell wall of the model. The first predicted longitudinal mode is at 415 Hz, this compares reasonably with the qualification test result of 492 Hz from the 0.5 g sine sweep and 470 Hz from the low level random vibration survey. The first lateral mode is predicted at 684 Hz; the corresponding test values from the qualification sine and random surveys were 560 Hz and 510 Hz respectively. Results from the vibration modes analysis were used in a shock response analysis to predict peak response to the random vibration environment. Given the observed damping and the permissible input notching of the performance specification, the peak displacements, accelerations, reaction forces, and strains computed were consistent with the 50g quasi-static design criteria discussed earlier.

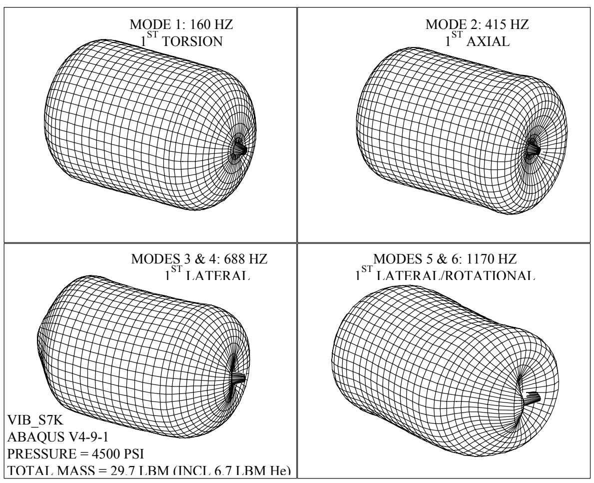

FIGURE 8. PRESSURE VESSEL VIBRATION MODES @ 4500 PSI

Lastly, a conventional bending and fatigue analysis of the stinger and port attachments was performed. Figures 9 and 10 show results for the stinger end of the vessel which was the most critical. The significant loading condition for the attachments is the 50 G equivalent quasi-static fatigue (118,000 cycles induced by random vibration discussed earlier) superimposed on the static operation pressure (4500 psi) and on the static sub-operational pressure (1680 psi). The sub-operational pressure is included because the stinger cantilever distance (dimensions "C" and "D" in Figures 9 and 10) is greater at lower pressure since the tank longitudinal expansion is smaller. Figure 9 shows a minimum margin of safety of +.48 at point "B" at 4500 psi for 118,000 cycles. Figure 10 shows a minimum margin of safety of +.37 margin point "B" at 1680 psi for 118,000 cycles. Note that the margin of safety is factored on the 50 G alternating fatigue load and not on the constant internal pressure. Figure 4 shows these critical margins on the modified Goodman diagram. Note that the stress levels are much too high for a material such as 6061-T6 aluminum. At point "A", the mean Von-Mises stress σ_m = 57.9 ksi was taken from the finite element analysis and the alternating Von-Mises stress σ_a = 8.1 ksi / 8.5 ksi was computed by superimposing an alternating meridional stress = 12.1 ksi / 12.8 ksi on the multiaxial stress state. Mean and alternating Von-Mises stress were computed in a similar fashion at point "B".

In summary, the analysis showed positive margins of safety for the pressure vessel. Positive margins of safety with respect to low cycle fatigue life of the liner were computed. Positive margins of safety with respect to 50 G equivalent vibration fatigue loading superimposed on constant operational (4500 psi) and sub-operational (1680 psi) pressure were computed for the end fittings.

	0 PSI	4500 PSI
DIM "C"	.95	.52
DIM "D"	1.69	1.26

POINT "A":
$\sigma_a = \sigma_{nom}$ x Fs x K_t
σ_a = 9.7 KSI x 1.0 x 1.25
σ_a = 12.1 KSI
σ_m = 57.9 KSI (MEAN VON MISES FROM FEA)
σ_a = 8.1 KSI (ALT VON MISES FROM FEA INCL σ_a = 12.1 KSI)
MARGIN OF SAFETY = +.49 (FIG 4)

POINT "B":
$\sigma_a = \sigma_{nom}$ x K_t
σ_a = 16.5 KSI x 1.0 x 1.46
σ_a = 24.1 KSI
σ_m = 20.0 KSI (MEAN VON MISES FROM FEA)
MARGIN OF SAFETY = +.48

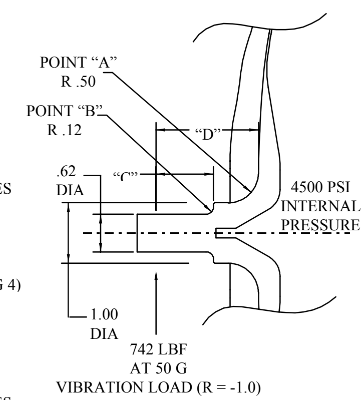

FIGURE 9. STINGER END FITTING VIBRATION FATIGUE ANALYSIS @ 4500 PSI

LINER FABRICATION

The liner is divided into two basic parts: the ported and stinger end bosses providing the mounting interfaces; and the liner membrane providing the pressurant containment barrier and the overwrap mandrel. The individual piece parts are shown in Figure 11.

The liner is constructed entirely of cp titanium except for the outlet tube stub which is 3Al-2.5V titanium. There are two grades of cp titanium, CP-3 is used on

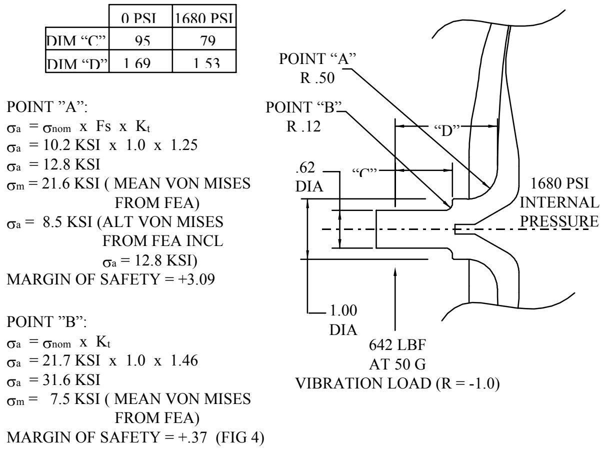

FIGURE 10. STINGER END FITTING VIBRATION FATIGUE ANALYSIS @ 1680 PSI

the central cylinder and formed domes and the higher strength grade of CP-70 is used on the bosses.

The central cylinder is fabricated from rolled and welded 0.020" sheet stock. The two end domes are hydroformed from sheet stock. The end bosses are machined from bar stock.

The liner is assembled using a variety of welding techniques and specialized tooling to minimize weld distortion. All welds are fully radiograph inspected to NAS 1514 and fracture critical penetrant inspected to MIL-STD-6866. The assembled liner is then fully annealed and leak tested prior to wrapping.

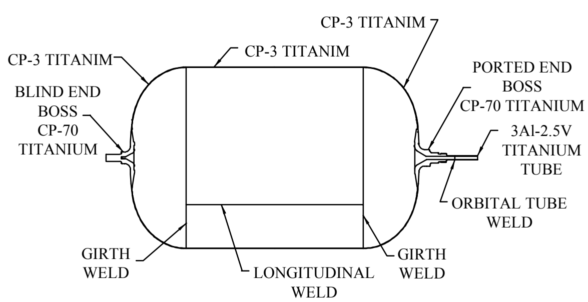

FIGURE 11. LINER CONFIGURATION

Tank Sizing and Finish Machining

Immediately after the tank is wrapped and cured, the vessel is subjected to a sizing operation (autofrettage) which substantially yields the liner. A finish machining operation is then performed on the tank bosses to establish the mounting interfaces.

QUALIFICATION PROGRAM

The qualification test program was performed to verify the tank design. Table 6 identifies the scope and sequence of the test program.

TABLE 6. QUALIFICATION TEST PROGRAM

Test Title
Examination of Product
Vol, Proof & MEOP Pressure Test
MEOP Pressure Cycle Life
Proof Pressure Cycle Life
External Leakage
Vibration Test- Sine,Random
External Leakage
Detanking Temperature
Examination of Product
Burst Pressure & Rupture Test
Data Review

Examination of Product

The tank was subjected to several visual and dimensional inspection operations prior to testing including internal boroscope inspections. No anomalies were observed.

Volume, Proof & MEOP Pressre Test

Tank volumetric capacity was performed at MEOP of 4500 psi. Tank requirement was 4105 in^3, actual recorded volume was 4280 in^3. The tank was also subjected to a proof pressure check of 5625±25 psi (test pressure was 5640 psi) and held at pressure for five minutes.

Proof & MEOP Pressre Test

The tank was subjected to a series of pressure cycles which included four proof cycles at 5625 psi and fifty MEOP cycles at 4500 psi. The tank axial growth and volumetric capacity were monitored during each pressure cycle. No anomalies were observed.

External Leakage Test

The tank was installed into a vacuum chamber and prssurized to 4500 psi with helium gas. The chamber pressure was reduced to <0.2 microns. The observed leakage was found to be <1.5 x 10^{-7}scc/sec GHe. A thermocouple was attached to the inlet boss to monitor tank temperature during the pressurization process. The maximum observed temperature was 146.7°F. However, subsequent observations revealed a 20-30°F temperature differential between the inlet boss and the liner membrane. It is highly likely that the tank was exposed to temperaures above 175°F. After the test, the tank was depressurized over a two hour period. The tank was visually inspected. No anomalies were observed.

Vibration Test- Sine and Random

The tank was subjected to a series of dynamic tests pressurized to MEOP with helium and installed into a fixture simulating spacecraft interfaces. The tank was subjected to a series of sine (to 15g's) and random (to 17 G's rms) vibration tests. All tests were successfully performed.

External Leakage & Detanking Temp.

The tank was installed into the vacuum chamber as previously described with an additional thermal barrier and additional thermocouples attached to both bosses and two on the liner. At the conclusion of the external leak test (with the chamber at vacuum), the tank was vented at 0.09 lbs/minute to evaluate the temperature drop. The lowest observed temperature was -61.3°F on the ported boss.

Examination of Product

At the conclusion of the above tests, an internal borescope inspection and external visual inspection was performed. No anomalies were observed. The tested tank weight with test fittings was 21.98 lbs (25% below specification requirement).

Burst and Rupture Test

The tank was hydrostatically burst tested to failure. The tank ruptured at 7880 psi with failure occuring in the hoop fibers with a liner separation in the parent material. The majority of the girth and longitudinal seam welds remained intact. The analytical predicted burst pressure was 7750 psi. The calculated performance factor of pressure*volume/weight (PV/W) was approx. 1.5×10^6 inches based on actuals.

CONCLUSION

The challenge of this program was to develop a high performance pressurant tank beyond conventional standards. The demonstrated peformance shows the goal has been achieved. The use of cp titanium liner provides a new material option in the COPV industry which has significant advantages over existing pressurant tanks.

The helium pressurant tank has been successfully qualified for the flight application defined. The production program is in process.

ACKNOWLEDGEMENT

This program was a significant challenge. The effort put forth by those involved at Lockheed Martin Astro Space, Pressure Systems, Inc., and Kaiser Compositek Corp. must be commended. Our sincere thanks and acknowledgement for the long hours expended to make this program a success.

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ABOUT THE AUTHORS

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