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ASME SECTION VIII RECERTIFICATION OF A 33,000 GALLON VACUUM-JACKETED LH_2 STORAGE VESSEL FOR DENSIFIED HYDROGEN TESTING AT NASA KENNEDY SPACE CENTER

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ABSTRACT

The Ground Operations Demonstration Unit for Liquid Hydrogen (GODU-LH2) has been developed at NASA Kennedy Space Center in Florida. GODU-LH2 has three main objectives: zero-loss storage and transfer, liquefaction, and densification of liquid hydrogen. A cryogenic refrigerator has been integrated into an existing, previously certified, 33,000 gallon vacuum-jacketed storage vessel built by Minnesota Valley Engineering in 1991 for the Titan program. The dewar has an inner diameter of 9.5' and a length of 71.5'; original design temperature and pressure ranges are -423°F to 100°F and 0 to 95 psig respectively. During densification operations the liquid temperature will be decreased below the normal boiling point by the refrigerator, and consequently the pressure inside the inner vessel will be sub-atmospheric. These new operational conditions rendered the original certification invalid, so an effort was undertaken to recertify the tank to the new pressure and temperature requirements (-12.7 to 95 psig and -433°F to 100°F respectively) per ASME Boiler and Pressure Vessel Code, Section VIII, Division 1. This paper will discuss the unique design, analysis and implementation issues encountered during the vessel recertification process.

NOMENCLATURE

AI	Authorized Inspector
ANSI	American National Standards Institute
ASME	American Society of Mechanical Engineers
BPVC	ASME Boiler & Pressure Vessel Code
ESC	Engineering Services Contract (NASA KSC)
FEA	Finite Element Analysis
GODU-LH2	Ground Operations Demonstration Unit for Liquid Hydrogen project.
IRAS	Integrated Refrigeration and Storage
KSC	Kennedy Space Center
LH_2	Liquid Hydrogen
LN_2	Liquid Nitrogen
MAWP	Maximum Allowable Working Pressure
MLI	Multi-Layer Insulation
NASA	National Aeronautics and Space Administration
NBP	Normal Boiling Point
NBIC	National Board Inspection Code
SSC	Stennis Space Center
‘	Feet (units)
“	Inches (units)
C_0	Original Ring Circumference
ΔC	Local Change in Circumference

D_o	Outer Diameter of Inner Vessel
I_{bp}	Bolt Pattern Moment of Inertia
I_{req}	Required Moment of Inertia of a Stiffener Section
$I_{x\text{-}sec}$	Moment of Inertia of the Chosen Stiffener Section
P	External Design Pressure for the Inner Vessel
ΔP	Pressure Difference
R_0	Radius of Inner Tank Shell
R_1	Outer Radius of Stiffening Ring at an Individual Unsupported Gap
S_0	Arc Length of Gap along Inner Tank Wall
S_1	Arc Length of Gap along Stiffening Ring
S_{max}	Maximum Allowable Unsupported Gap Arc Length
δ	Maximum Unsupported Ring Gap
$\varepsilon_{allowable}$	Maximum Allowable Ring Strain
ε_{local}	Calculated Strain at an Individual Unsupported Gap
ε_{ring}	Calculated Strain of a Total Ring Assembly
θ	Angle of an Individual Unsupported Ring Gap
θ_{max}	Maximum Allowable Angle of an Individual Unsupported Ring Gap

INTRODUCTION

Use of liquid hydrogen as rocket fuel dates back to the infancy of space exploration; utilized on a large scale by the Atlas Centaur upper stage (which is still in service), and the Apollo Saturn V second and third stages in the 1960's. More recently, the Space Shuttle relied on roughly 380,000 gallons of LH_2 at lift-off to make it to orbit, and the United Launch Alliance's Delta IV utilizes a common core booster powered by the cryogen.

Extremely low density and high specific impulse makes LH_2 a superior propellant, however, possessing the second lowest normal boiling point (NBP) of all the common cryogens (-423°F) also makes utilizing this commodity exceedingly difficult and costly. Over the duration of the Space Shuttle program NASA lost approximately 50% of the hydrogen purchased because of continuous heat leak into ground and flight vessels, transient chill down of warm cryogenic equipment, liquid bleeds, and vent losses [1]. A key goal of the GODU-LH2 project at Kennedy Space Center is to develop technologies and operational methods capable of eliminating such losses—which can translate into untold savings over the life of a launch program—and to explore new possibilities for propellant conditioning.

Central to the GODU-LH2 project is the concept of Integrated Refrigeration and Storage (IRAS), which allows for energy (i.e. heat) to be removed directly from the LH_2 by a cryogenic refrigerator, and affords three unique capabilities: (1) if the tank heat-leak and refrigeration power are balanced, "zero boil-off" can be achieved, and the liquid level can be maintained indefinitely; (2) gaseous hydrogen can be introduced into the vessel and liquefied to fill the tank, as opposed to using liquid tanker trucks; and (3) if the refrigeration power is greater than the tank heat-leak the liquid can be cooled below its NBP; this is referred to as densification, and will be the primary mode of interest in this publication.

The IRAS tank developed for the GODU-LH2 project consisted of a 33,000 gallon, horizontal LH_2 storage tank originally fabricated by Minnesota Valley Engineering in 1991 for the Titan program. The vessel was utilized at launch complex 40 at the Cape Canaveral Air Force Station in Florida until completion of the program in 2005, at which point ownership was transferred to NASA. Figure 1 shows the Titan tank prior to GODU-LH2 modifications.

FIGURE 1: 33,000 GALLON LH_2 STORAGE TANK USED FOR GODU-LH2

During densification operations pressure inside the inner vessel will drop below one atmosphere due to sub-cooling of the liquid hydrogen. This condition invalidates the original ASME Boiler and Pressure Vessel Code (BPVC), Section VIII certification due to both the lower pressure and temperature. In order to recertify the tank to the new operational conditions modifications needed to be made to the inner vessel to protect against collapse were the vacuum pressure lost inside the annular space while densified. To this end a system of stiffening rings and stringers was devised, analyzed and installed inside the tank per the direction of paragraphs UG-28 through UG-30 in the 2013 edition of the BPVC [2].

The task of pulling together all the required information, providing engineering guidance, and ultimately working with the ASME Authorized Inspector (AI) to stamp the tank was contracted to GP Strategies of Columbia, MD.

ORIGINAL TANK CONSTRUCTION

Original fabrication and testing of the tank was done per the 1989 edition of the BPVC, Section VIII, Division 1 rules. It is vacuum-jacketed, with an inner vessel diameter and length of 9.5' and 71.5' respectively, and an outer shell diameter and length of 11.3' and 75.5'. The inner shell is constructed of six cylindrical sections and two 2:1 elliptical heads circumferentially welded together. Each cylindrical section is a 0.382" thick, rolled sheet of SA240 304L stainless steel, longitudinally welded. Heads are also constructed from SA240 304L. The outer shell is constructed from 0.313" thick, SA240 A-36 carbon steel, and employs sixteen stiffening rings.

Major penetrations into the inner vessel consist of three 3" fill/drain ports, one 4" vent port, and a 23" diameter man-way port located at the top of the tank. This man-way was the only means of entry into the tank for personnel and equipment, and was sealed with a vacuum plug/capacitance probe assembly (for liquid level sensing) from the manufacturer.

Per the original U-1A form the certified operational temperature and pressure ranges of the vessel were -423°F to 100°F, and 0 psig to 95 psig respectively—essentially, the tank was designed to store normal boiling point hydrogen with minimal losses, and then, when required, withstand a moderate positive pressure in order to flow out into the vehicle.

DESIGN REQUIREMENTS FOR GODU-LH2 MODIFICATIONS

A primary performance goal of the GODU-LH2 project was to densify LH_2 to the maximum extent permitted by the system. This necessarily means that the liquid temperature, and consequently the tank pressure, be decreased as low as possible; a lower bulk liquid temperature of -433°F, a 10°F decrease from NBP, was the target estimate. This translates to a density increase of roughly 7%, and a tank pressure of about 2 psia (-12.7 psig). This new temperature and pressure were used as the lower limits for the recertification effort, and defined the new design temperature and pressure ranges as -433°F to 100°F, and -12.7 psig to 95 psig respectively.

Driving the design of the internal stiffeners were two major constraints: (1) all materials, tools, etc. required to execute the modifications must fit through the 23" diameter man-way port, and (2) no welding was permitted due to safety and technical concerns, as well as to protect the delicate multi-layer insulation (MLI) wrapped around the outside of the inner vessel. These constraints meant that the stiffening rings and stringers must be modular, and assembled using only bolted joints. Also, a method must be employed to load the ring sections against the tank wall since welding was not an option.

STIFFENER DETERMINATION PER SECTION VIII

Determination of the cross-section, placement, and total number of stiffening rings required was accomplish via the methodology found in the 2013 BPVC, paragraphs UG-28 and UG-29. The flow chart in figure 2 summarizes this iterative process.

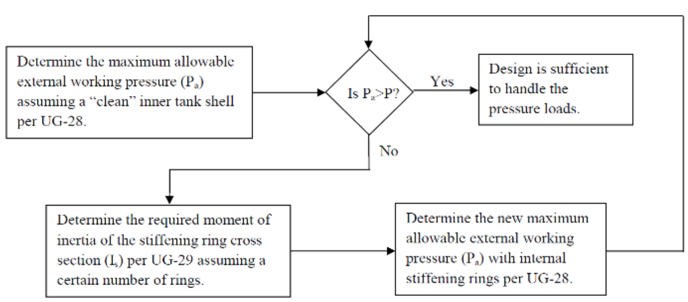

FIGURE 2: STIFFENER DESIGN METHODOLOGY PER SECTION VIII

Initially, the maximum allowable external working pressure of the inner tank in its original, unmodified configuration was calculated, and found to be only 2.1 psig—significantly lower than the 12.7 psig required. Following the above process, various stiffener cross sections and quantities were explored until a satisfactory option arose: nine total rings, evenly spaced at 81.7" intervals, with a C5x6.7 channel cross-section. This cross-section satisfied the required moment of inertia with considerable margin (I_{req}=5.95 in^4, $I_{x\text{-}sec}$=7.49 in^4), while keeping the total weight per ring to a minimum.

Using this new stiffener configuration, the maximum allowable external working pressure of the inner tank was increased to 18.5 psig. Therefore, the chosen ring section satisfied the requirements of Section VIII.

STIFFENING RING DESIGN

Once the fundamental stiffener cross-section was established, the issues of modularity and ring-to-wall loading were addressed.

It was necessary to break the individual rings into sections in order to fit through the man-way port. These sections would then be placed into their proper positions inside the tank and bolted together with joint splices to form a continuous ring.

Considering the total weight of an unbroken ring (≈185 lb), and the impracticality of lifting/maneuvering heavy segments by hand inside the tank, it was decided that each ring should be split into three equal segments. This kept the weight of each segment within a manageable range, but did not split the rings up into unreasonable numbers of individual pieces.

Each segment had an outer arc length of 118.9" (outer diameter=114"), which allowed a 1/2" gap to exist between each of the three segments when positioned. This was necessary for two reasons: Ease of placement, since all work had to be performed by hand; and to allow the individual sections to flex during loading in order to take up any significant gaps that might exist between the ring and inner wall.

The 27 total ring segments were mechanically formed (bent) to the correct radius (57") by B & H International, Bakersfield, CA, and shipped to Kennedy Space Center (KSC) for final preparation and assembly. Since the segments received were longer than required, each was trimmed in-house, leaving short sections that were subsequently used as joint splices for each ring assembly.

Per the assumptions used in the Section VIII stiffener section analysis, namely that each ring is continuous and has a constant moment of inertia around its circumference, the three joints on each ring must also carry a moment of inertia equal to or greater than that of the ring section. This included the splice section as well as the bolt pattern used; also, the fasteners must be sufficiently strong to deal with the shear loads. Since the trimmed sections were used as the joint splices, a continuous moment of inertia was achieved around the entire ring circumference; therefore, the only engineering tasks were to determine an adequate bolt pattern, and fastener set.

The moment of inertia of a bolt pattern was calculated via Eq. (1):

$$I_{bp} = \sum_{i=1}^{N} A_i d_i^2 \qquad (1)$$

Where N = number of bolts in the pattern, A_i = the cross-sectional area of the ith bolt, and d_i = the distance from the bolt pattern center of rotation to the ith bolt. Numerous bolt patterns and fastener sizes were analyzed until an adequate configuration emerged that satisfied the required moment of inertia and space constraints. Figure 3 shows the final joint splice design.

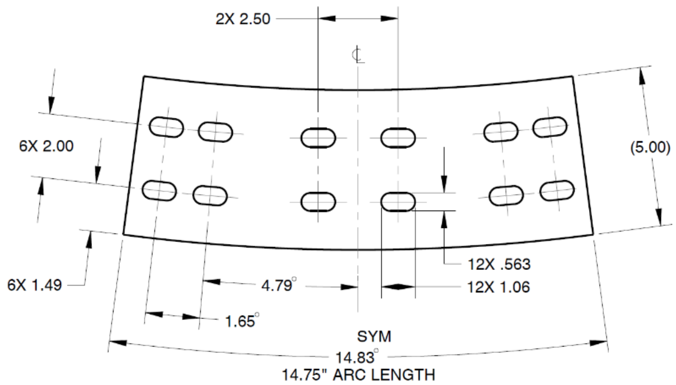

FIGURE 3: FINAL JOINT SPLICE DESIGN (C5x6.7 C-CHANNEL SECTION)

This design incorporated 1/2-13 stainless steel fasteners and possessed a bolt pattern moment of inertia of 6.16 in^4 as calculated using Eq. (1); this is greater than the 5.95 in^4 required, hence the design was deemed sufficient. Bolt shear and tear-out calculations were also conducted based upon calculated hoop stresses and finite element results, and were found to be well within the strength limits of the fasteners.

Supplementing the C-channel splice described above was a 304L stainless steel plate that aligned with the four holes in the middle of the splice, effectively "sandwiching" the adjoining ring segments. This plate measured 5" x 3.5" x 0.25" thick, and provided additional strength to the overall joint.

As stated previously, a means of loading the ring segments against the inner wall was necessary since welding was not an option. This was achieved by incorporating a "thrust bolt" at each joint to force the three ring segments outward. Each of the 5/8-18 x 11" long threaded rods spanned the three joints on the ring assemblies, and nuts were torqued outwards, acting on welded plates at the ends of each ring segment.

Accommodating other system design requirements—including the need for drain holes at the lowest point in the ring, stringer placement holes, and attach points for the refrigeration heat exchanger [3]—ultimately resulted in ring assemblies containing three unique segments.

Figure 4 depicts a typical, as-installed, bolted joint assembly, while figure 5 shows the overall stiffening ring assembly.

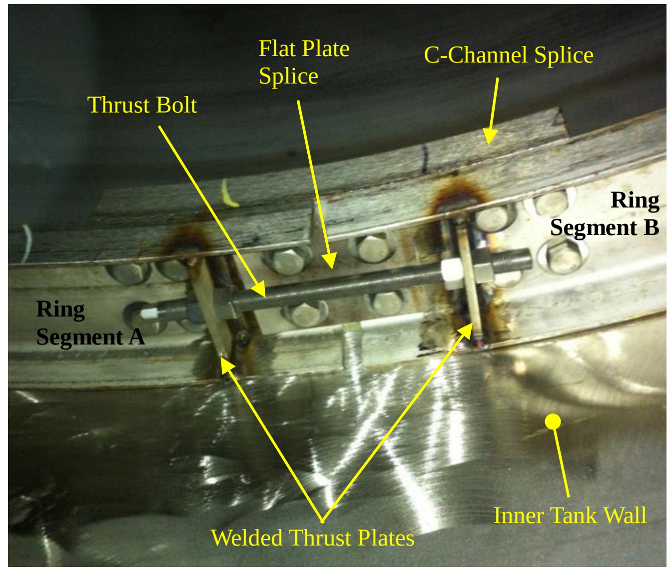

FIGURE 4: FINAL BOLTED JOINT ASSEMBLY (3 PER RING)

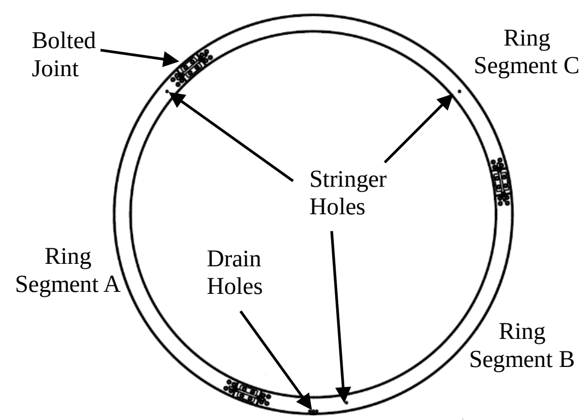

FIGURE 5: STIFFENING RING ASSEMBLY

STRINGER DESIGN

In order to satisfy the requirements put forth in paragraph UG-29 for non-welded internal stiffening rings, an adequate means of support was needed to ensure proper ring alignment was maintained were the failure scenario realized. Additionally, these supports would protect against potential ring tipping during transient chill-down of the inner tank, when large temperature differences between the top and bottom of the vessel could create unforeseeable thermal contraction issues, and potential loosening of the rings from the tank wall.

A system of 15 longitudinal stringers was devised that tied sets of stiffening rings together, effectively eliminating the chance that any one ring could tip were it to become loose. These stringers must also be allowed to contract length-wise without imparting large loads to their associated rings during chill-down from ambient temperature. To accomplish this, a telescoping feature was designed into the stringers, which allowed each end to be hard-fastened to a ring web, yet still allow the support to shrink when brought down to cryogenic temperatures.

Each stringer unit was constructed from three, 304L stainless steel pipe sections, one 1/2" schedule-10 piece and two 3/8" schedule-40 pieces. A 14" section of 1/2" pipe was bored out to accept the 3/8" rods freely, yet still maintain a close tolerance. Then one end was welded to a 3/8" rod, leaving 13" of 1/2" pipe to accept another 3/8" section; this formed the telescoping portion of the support. Each end was then threaded to accept 9/16-18 bolts, and 3/16" diameter drain holes were drilled down the length of the stringer.

One additional design feature of the stringers provided invaluable benefits beyond that of structural stability. A removable 3/16" pin was included that locked the two telescoping ends together at the minimum length (81.5"). This allowed the stringers to be used as length gauges, and helped to square the rings up with the inner tank wall during installation. Since the rings were modular, it was exceedingly difficult to ensure their perpendicularity to the tank wall prior to loading with the thrust bolts. Utilization of the locked stringers alleviated this issue to a great extent since two or more rings were then tied together at three locations, and at equal distances, which constrained each one in the vertical plane. This left clocking of the ring assemblies as the only other concern during placement; however, this was a much easier issue to solve than ensuring perpendicularity. The locking pins were removed from the stringers once each ring was secured.

INSTALLATION OF STIFFENING RINGS

Installation of the stiffening rings and supporting hardware was carried out between January and August of 2014 by NASA and Contractor personnel at the Kennedy Space Center in Florida. Prior to each tank entry the volume was purged with outside air via a blower unit for roughly one hour, and then a sample was taken by a safety representative to ensure proper atmospheric composition within the confined space. Also, the permanently installed scaffolding around the tank was inspected for any signs of damage or wear. Each team member entering the vessel was required to wear a safety harness at all times, and a tri-pod with an arresting cable system was employed during entry and exit using the ladder. This system was also kept ready at all times while personal were in the tank (i.e. the cable end was tie-off inside the tank) in case an injury occurred, and help was needed to hoist the worker out. Outside purge air was also supplied while personnel were in the tank to provide adequate ventilation and cooling.

Hardware was transported to the staging platform at the top of the tank with help from a vertical scissor-lift box truck. Each of the 27 ring segments, and 15 stringers was cleaned using isopropyl alcohol and carefully lowered into the tank along with supporting hardware and tools. A portable generator provided electricity into the tank via extension cords for lighting, various power tools, and the blower unit.

Once all the required hardware was inside the vessel, measurements were made per the engineering drawings, and the tank was marked with the proper location of all nine rings. Figure 6 shows the critical dimensions and overall stiffener configuration.

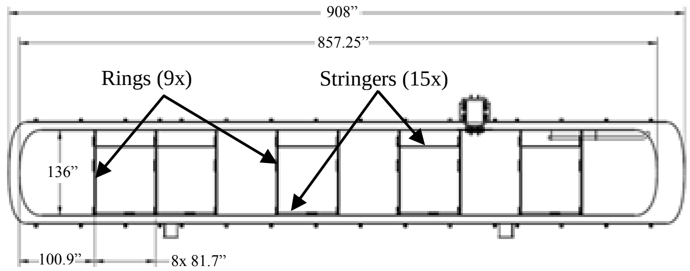

FIGURE 6: STIFFENING RING AND STRINGER CONFIGURATION

Construction of each ring assembly began with pre-fitting the joint splices at the ends of segment A (see figure 5) while it rested at the lowest point of the tank. The fasteners were left loose enough to allow segment B to be easily slid into the joint. Segment B was then lifted into position, the A-to-B joint was secured, and the two-ring assembly was rotated along the inner wall until the drain holes resided at the lowest part of the tank. Finally, segment C was lifted and slid into the C-to-B joint. This joint was secured while the rest of the assembly was held in place by personnel, and then the final A-to-C joint splices were installed. Once all three joints were secure, the ring assembly was stable enough to remain in the proper position with minimal human intervention. This allowed the other team members to install the thrust bolts and apply enough load so that the ring could not move unintentionally.

This process was repeated for each of the nine ring assemblies at the pre-determined locations. Once all were in place the stringers were installed and final adjustments were made to the orientation and position of the stiffeners. Final steps included engaging the thrust bolts to the maximum extent possible in order to load the rings against the wall and minimize any gaps, torqueing all 1/2" fasteners to 45 lb-ft, and removing the 3/16" locking pins from the stringers. Figures 7 and 8 show the inner tank during stiffening ring installation (also shown in figure 8 is the man-way opening, blower hose, ladder and unassembled heat exchanger coils).

FIGURE 7: STIFFENING RINGS PRIOR TO STRINGER INSTALLATION & FINAL ADJUSTMENT

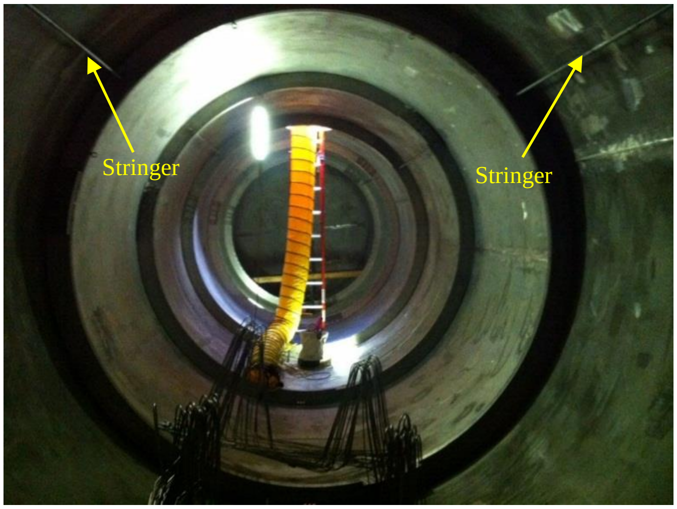

FIGURE 8: FINAL STIFFENING RING PLACEMENTS WITH STRINGERS

RING-TO-WALL GAPS

Even though, during installation the rings were loaded against the inner wall to the maximum extent possible using the thrust bolts, gaps still existed. This was caused by the inconsistencies in the roundness of both the tank—in particular at the longitudinal welds where the two ends of the rolled sheet came together—as well as the fabricated ring segments.

Following a post-installation inspection by GP Strategies, it was revealed that some of the gaps were too large, hence the stiffening rings did not comply with Section VIII, and the issue must be addressed before certification could be pursued.

Multiple solutions were explored, eventually culminating in the use of stainless steel shim stacks to break up the gaps into smaller, code compliant lengths. The minimum unsupported arc length allowed by Section VIII was determined via paragraph UG-29, step 8, section c: "Any gap in that portion of a stiffening ring supporting the shell shall not exceed the length of arc given in Figure UG-29.2." Using the outer diameter of the inner tank at -433°F (113.75", to account for thermal contraction and determined via analysis), the length between stiffening rings (81.7"), and the inner tank wall thickness (0.382", from the original manufacturer U-1 form), Fig. UG-29.2 yielded a maximum unsupported arc length of $0.10D_0$, or 11.4".

Shims were placed at predetermined locations for each ring based upon the number and size of its gaps. They consisted of 2" wide x 3" long x 0.060" thick 304L stainless steel sheets, bent at 90° and slotted to accommodate a #8 bolt used to fasten it to the C-channel web. Multiple were stacked up in order to fill in the various gap widths that existed throughout the ring sets. Figure 9 depicts a typical shimmed gap configuration.

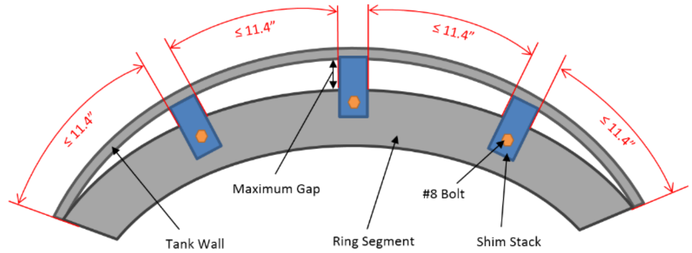

FIGURE 9: TYPICAL SHIMMED RING GAP (EXAGGERATED)

Implementation of these shims effectively satisfied the requirements put forth in UG-29; however, two other analysis tasks were performed in order to substantiate the solution: (1) determining if the remaining gap widths and corresponding arc lengths constitute a strain below that allowed by code if the failure scenario was realized; and (2) consideration of potential load concentrations at the individual gap ends due to the abrupt starts/stops introduced by the square shim stacks.

Task 1 began by determining the allowable strain ($\varepsilon_{allowable}$) via the definition of Young's modulus and the code allowable stress of 304L stainless steel at -433°F (2.93 x 10^7 psi [4] and 16,700 psi respectively). This yielded an allowable strain of 0.00057 for each ring.

Next, the total strain for each individual ring was calculated for comparison to the allowable. The method for calculating the individual ring strains consists of summing the localized changes in circumference (ΔC) of the inner tank wall (i.e. the gap arc lengths associated with each ring) and dividing by the original, total circumference. Figure 10 shows an example gap in one ring and the associated variables used to determine the localized change in circumference (θ is defined between the beginning and end of a given gap, this may be between the ends of the shim stacks as in Fig. 9, or as depicted in Fig. 10).

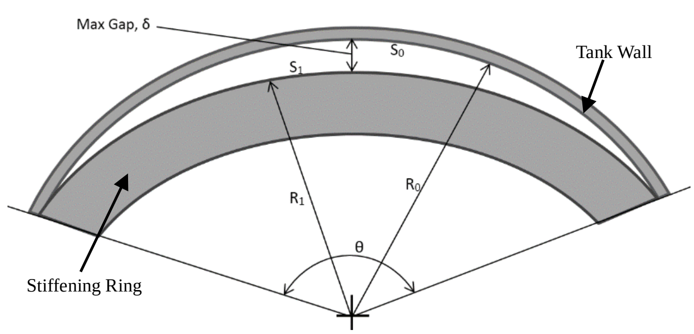

FIGURE 10: VARIABLES USED TO CALCULATE THE LOCALIZED CHANGE IN CIRCUMFRENCE

From figure 10 the individual ring strains could be calculated using Eq. 2, and the total ring strain from Eq. 3:

$$\varepsilon_{local} = \frac{\Delta C}{C_0} = \frac{S_0 - S_1}{C_0} = \frac{\frac{\theta}{180}\pi(R_0 - R_1)}{2\pi R_0} = \frac{\theta\delta}{360R_0} \qquad (2)$$

$$\varepsilon_{ring} = \sum \varepsilon_{local} \qquad (3)$$

Since the maximum arc length allowable was 11.4", S_0 and θ were effectively predefined, therefore $S_0=S_{max}$ and $\theta=\theta_{max}$. Using the mathematical definition of arc length θ_{max} was determined via Eq. 4, and found to be 11.5°.

$$\theta_{max} = \frac{180S_{max}}{\pi R_0} \qquad (4)$$

With θ_{max} known, Eq. 2 was then used to determine the maximum allowable localized strain as a function of only the gap width.

Since each ring had a unique set of gaps, the above equations had to be applied using the gap sizes found during actual inspection. All the localized strains for each ring were then summed via Eq. 3, and evaluated using the following inequality.

$$\varepsilon_{ring} < \varepsilon_{allowable} \qquad (5)$$

If Eq. 5 was violated then more shims were required to break up the gap set for the ring being analyzed. This process was repeated until each of the 9 stiffening rings were in compliance, and revealed the total number and placements of the shim stacks.

A finite element model of the inner tank, and rings with associated gaps was developed for task 2. This model was run using an external pressure load of 12.7 psig—corresponding to the maximum ΔP the inner tank would experience during the proposed failure—for both shimmed and un-shimmed gaps in order to determine if the square shims produced unwanted stress concentrations.

Two finite element models were constructed in Creo 2 and analyzed using the embedded solver Mechanica. The first had each ring modeled with its unique, unsupported gap configuration, and the other with shimmed gaps. In each, the inner tank wall was modeled using shells (2694 Tri and 4931 Quad elements total) and was dimensioned to reflect thermal contraction from ambient to -433°F. The rings were modeled as solids (2738 total Tetra elements), and constrained in all degrees of freedom. Maximum stress, strain, and radial displacement were examined in both studies. Figures 11 and 12 show stress results for the unsupported and shimmed models respectively.

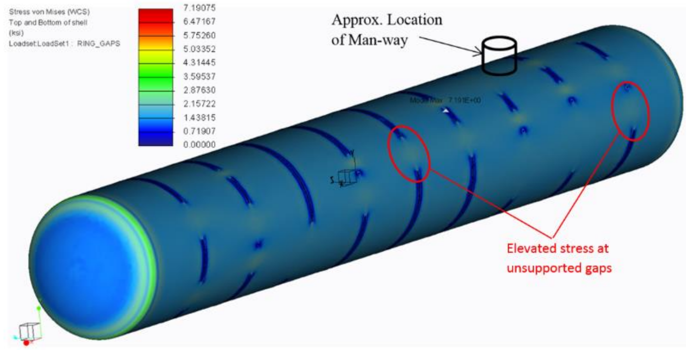

FIGURE 11: FEA VON MISES STRESS RESULTS FOR UNSUPPRTED GAPS (MAX STRESS=7.19 ksi)

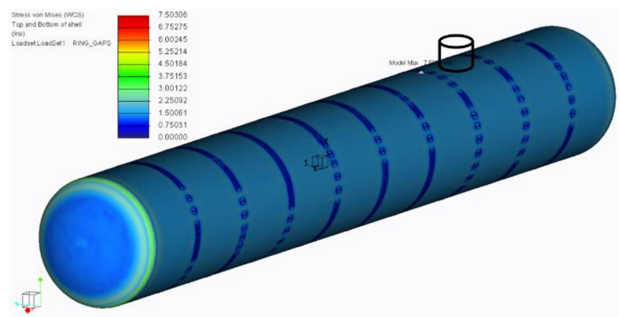

FIGURE 12: FEA VON MISES STRESS RESULTS FOR SHIMMED GAPS (MAX STRESS=7.50 ksi)

In both cases the addition of shims was found to have a negligible effect on the three design parameters. Figures 11 and 12 show that the maximum stress actually increased slightly with the addition of shim stacks, whereas the extended results showed that maximum strain and displacement were relatively unaffected. What is affected, however, is the distribution of stress over the tank shell; which is noticeably more uniform in Fig. 12.

Since the maximum stresses and strains determined via the FEA were well below that allowed by code (16,700 psi & 0.00057 respectively) the solution to use shims to break up the unsupported gaps between the ring and inner tank wall was deemed a suitable method by GP Strategies, and the Section VIII certification process could continue.

UPDATED MAN-WAY PLUG

Having no fluid or instrumentation penetrations made the original man-way plug unsuitable for GODU-LH2 operations, hence an updated one was designed and fabricated per the projects requirements. These tasks were completed at NASA Stennis Space Center (SSC) in Mississippi, and shipped to KSC as an ASME code stamped vessel per Section VIII. The design included three bayonet-style fluid connections—two for the refrigerant inlet and outlet to the internal heat exchanger, and one for a gaseous hydrogen inlet—and four, 24-wire Conax brand instrumentation feed-through's. The assembly was vacuum-jacketed to limit heat-leak, and attached via a 24", 150lb ANSI flange using Fluorogreen gasket material. Figure 13 shows the man-way plug post-installation.

FIGURE 13: UPDATED MAN-WAY PLUG ASSEMBLY POST-MODIFICATION TESTING

After the tank modifications were completed a series of three different tests were conducted in order to gain certification. First was a positive pressure test at 125% of MAWP, followed by a negative pressure test at 110% of the required pressure load on the outside of the inner vessel, and lastly a tank cold-shock was performed using liquid nitrogen. The positive pressure test was carried out using compressed nitrogen up to 119 psig (1.25 times 95 psig); while the negative pressure test was accomplished by breaking the vacuum on the annulus using gaseous nitrogen, and then pulling a vacuum on the inner tank until a ΔP of 14 psig (1.1 times 12.7 psig) was achieved across the inner tank wall. For both pressure tests GP Strategies coordinated with the Authorized Inspector to be present and officially witness the operations.

Once the pressure testing was complete, annulus vacuum was restored, and cold-shocks were carried out. Two 4,000 gallon liquid nitrogen tankers were emptied into the tank over the course of about 4 hours, effectively chilling down the bottom quarter of the inner vessel and stiffening rings. Liquid level was monitored using instrumentation rakes located inside the tank; and following chill-down, showed that roughly 5,000 gallons of LN_2 remained out of the 8,000 gallons that had been introduced.

FINAL CERTIFICATION AND TANK STAMPING

Following the installation of the internal stiffeners, acceptance of all the supporting analysis, and successful completion of the required pressure tests, the vessel was finally eligible for recertification, and application of a new R-stamp by the Authorized Inspector. Shown in figure 14 is the updated tank nameplate with the new certified operating conditions.

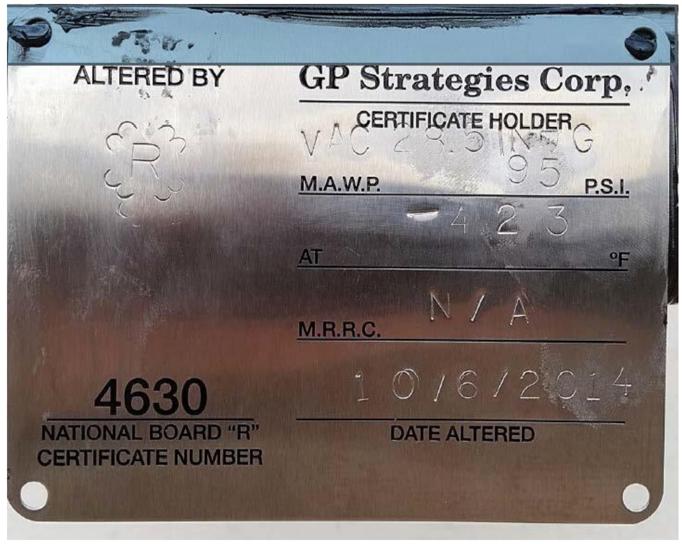

FIGURE 14: IRAS TANK CERTIFICATION PLAQUE

ISSUES WITH MINIMUM TEMPERATURE

Figure 14 reveals that the new operating pressure of the vessel is 28.5 inHg (vacuum) to 95 psig as required by the GODU-LH2 project. However, the temperature rating remains the same as the original, and is only certified to a temperature consistent with normal boiling point hydrogen, not densified as was desired. This is due to the Section VIII impact test requirements having changed between the 1989 edition, which the original tank was built to, and the current 2013 edition.

As the original impact test data was not available from the manufacturer due to the age of the tank, in order to certify the vessel to a lower temperature in accordance with Section VIII, paragraph UB-22 and UG-84, and the National Board Inspection Code (NBIC) paragraph NB-23, Charpy impact tests would have been required for all the affected base metal and weld metal materials; and at locations including the vessel heads, shell, nozzles, flanges, etc. Removing test specimens from the inner vessel was not feasible as this would have been required for all combinations of base and weld material based on material specification and heat number. Furthermore, traceability of the material specification and heat number was not available from the manufacturer due to the age of the vessel.

This resulted in the re-certification being issued at the new design pressure but not at the desired minimum temperature. During GODU-LH2 densification operations however, the liquid temperature will indeed be decreased below -423°F; therefore, approval to use the tank as intended required approval from the KSC Chief Engineer, as well as a waiver to the high level NASA requirements.

CONCLUSION

A 33,000 gallon liquid hydrogen tank has been re-certified to a new minimum pressure per the 2013 ASME Boiler and Pressure Vessel Code, Section VIII, Division 1 rules to facilitate densification testing at NASA Kennedy Space Center in Florida. Modifications to the vessel included modular, internal stiffening rings to protect against collapse while at sub-atmospheric ullage pressures; longitudinal, telescoping stringers to provide ring stability; and an updated man-way plug to allow for fluid and instrumentation feed-through's. Design, fabrication and installation of all new or updated components was done per Section VIII, and overseen by an authorized repair agency.

Placement of the new, R-stamped tank information plate was completed in October of 2014; and rated the tank to a pressure range of 0.7psia to 110 psia, (originally 14.7 psia to 110 psia). This afforded safe storage pressures of densified liquid hydrogen down to the triple-point (-434.8°F).

GODU-LH2 testing operations officially began in March 2015.

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