RESEARCH ON THE OPTIMAL SHAPE OF HIGH-PRESSURE TANKS FOR VEHICLES WITH VERTICAL TAKE-OFF AND LANDING

Alexandru – Mihai CIȘMILIANU[1], Mihaela NĂSTASE[2], Ionuț Cosmin ONCESCU [3]

Despite the extensive researches regarding the space application high-pressure tanks (HPT), real-time searches for commercial of the shelf (COTS) solutions designed for a reusable Vertical Take-off Vertical Landing (VTVL) vehicle were never addressed up until now. A study was made to identify a feasible solution which can be transformed into a COTS option. The main objective of this paper is to put down the first stone in this direction, proposing a design methodology for a reliable and feasible HPT, having demonstrated a way-forward for optimal shape determination for a self-imposed case study and to seek a continuity in this direction.

Keywords: High Pressure vessels, Tanks, VTVL Vehicle, Vertical Take-off Vertical Landing, Designed for demonstrators, Niche, Tank shape optimization.

1. Introduction

High pressure tanks/vessels are extensively used in the aerospace industry, especially for liquid engine rockets. A primary concern of tanks used in space vehicles is that they must withstand the high pressure loads that they are subjected to and at the same time have the lowest mass possible. The design of high-pressure vessels has been an important topic which has been explored for decades. However, few research studies have been focused on identifying the optimal design of such a structure.

Previous research has established that one of the critical areas that needs to be investigated in shape optimization is the design of the head of the pressure vessel. At the head and cylinder junction, high bending moments and shear forces appear in order to compensate from the stiffness difference between the

[1] PhD Student, Polytechnic University of Bucharest, National Institute for Aerospace Research "Elie Carafoli" INCAS, Romania, e-mail: cismilianu.alexandru@incas.ro
[2] PhD Student, Polytechnic University of Bucharest, National Institute for Aerospace Research "Elie Carafoli" INCAS, Romania, e-mail: nastase.mihaela@incas.ro
[3] PhD Student, Polytechnic University of Bucharest, National Institute for Aerospace Research "Elie Carafoli" INCAS, Romania, e-mail: oncescu.ionut@incas.ro

components [1]. Middletown and Owen [2] used the finite element method and parametric optimization techniques to minimize the maximum shear stress in the design of a pressure vessel head. Instead of investigating the mechanical stress, Blachut [3] applied parametric optimization to minimize the weight by considering the limit pressure that first causes a full plastic region along the entire thickness of the vessel. Other relevant work was performed by Zhu and Boyle [4], who used the elastic compensation method to estimate the limit loads that can be used as an objective function in the shape optimization techniques for the vessels. Banichuk et al. [5] used an analytical approach to find the optimal curve profile of the head in order to maximize the ratio between the head volume and mass subjected to stress limits. Magnucki and Lewinsky [6] analytically assessed the geometrical head to generate and equivalent stress less than or equal to the cylinder stress.

Most pressure vessels are closed by convex torispherical, ellipsoidal or hemispherical heads. Axially symmetric shells under uniform pressure can be considered as structures in the membrane stress state [7]. Torispherical head closures were optimized in the work proposed by Batchelor and Taylor [8]. Muc and Gruba [9] used splines developed by genetic algorithms to optimize hemispherical heads. Yushan and Wang [10] determined the best parameters of two-arc approximate ellipsoidal heads.

The specific objective of this research was to identify the optimal shape of high-pressure vessel for a reusable Vertical Take-Off and Landing vehicle that can withstand the loading conditions of the mission and at the same time possess the lowest weight possible. This paper follows a case-study design, with in-depth analysis of the optimum head shape of the pressure vessel. The originality of this study is that it also presents a development plan from the requirements stage up to the testing, manufacturing and integration stages and the tanks are now envisioned to be designed for reusable VTVL demonstrators which need a different design mindset as the needs are different.

2. Development Methodology

A design technology roadmap is developed through identifying the critical steps and actions and translating them into a logic scheme with the scope of giving an understanding on the needed pathway which is needed to be taken in order to obtain a High-Pressure Tank COTS option.

A constricted design methodology is presented below having the role to present an image of perspective onto the main pathway which is needed to be undertaken in order to achieve the end goal.

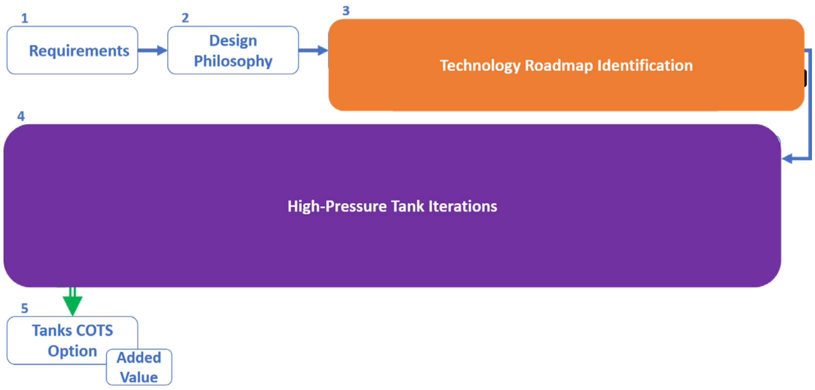

Fig. 1. Design Methodology Roadmap

The methodology starts with the definition of the considered Requirements (1), in this case, self-imposed.

Table 1

Self-imposed requirements

Name		Value
Volume [liters]		120
Working pressure [bar]		30
Compatible with media		95% nitric acid
Mounting		equatorial
Design envelope		600 L 600 l 600 H [mm]
Porting		1 top 1 bottom
Port size 5C	Top	¼ inches
	Bottom	¼ inches

Having the restrictions in mind a Design Philosophy (2) has to be put into place to understand the reasons of the desired product and this way to adapt a mindset to "design for" a demonstrator an HPT COTS option. The design philosophy is considering the critical aspects of an HPT which is needed to be designed for a reusable demonstrator and shall concede the following: fit the self-imposed requirements, offer compatible mounting (DFA -Design for Assembly), design for pressure cycling, DFC - Design for Cost, design for fast development and implementation.

Point 3 of the scheme is containing a cycle named Technology Roadmap Identification (3) followed by a cycle named High-Pressure Tank Iterations (4) and based on the outcome of this last item, an HPT COTS option (5) shall be available.

The extended version of the Design Technology Roadmap is presented below. This scheme presents the sub-phases which are needed to be made in the above-mentioned cycles.

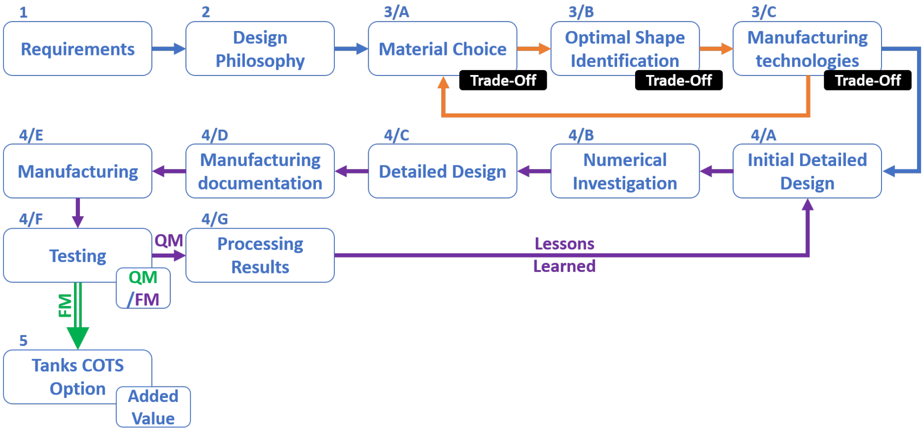

Fig. 2. Design Methodology Roadmap – Extended

The Technology Roadmap Identification (3) represents a cycle based on trade-offs which is needed to be made in order to decide the best available material (3/A) to be considered in the shape optimization (3/B) and, based on the result of the optimization, based on the available technologies (3/C), the optimum choices. Based on the optimum manufacturing technologies choices, the cycle is taken again to see what raw material is available from the initial consideration and the loop is made again until a result is achieved. At the end of the cycle, the expected result shall be a chosen material for the chosen shape and the chosen manufacturing technology.

The High-Pressure Tanks Iterations (4) is a cycle which in the extended version, consists of a set of steps which are needed to be taken until a solution is found starting with an Initial Detailed Design (4/A) which is made based on the results of the Technology Roadmap Identification cycle (3). This is then verified trough Numerical Investigation (4/B) and the outcome of this verification is translated in minor design changes made on the initial design which derive into a Detailed Design (4/C). An empirical verification is needed in order to see if the Numerical Investigation (4/B) reflects the reality, therefore, Manufacturing Documentation (4/D) is elaborated followed up by the actual Manufacturing (4/E) and Testing (4/F). The results of the tests are then processed (4/G) and the “lessons learned” are then implemented in the Initial Detailed Design (4/A) and the cycle is made again.

The testing campaigns are thought of differently depending on the state of the design. An extensive testing campaign which will lead ultimately to a destructive test is envisioned for each new design and if the outcome is accepted, the cycle is made again and the new tank is put through a reduced testing campaign named Flight Model Testing which results in a usable COTS option tank.

3. Optimal Shape Determination

A concept trade-off study has been conducted in order to determine the optimal design of the oxidizer tank in terms of weight and integration in the self-imposed system configuration. The structure of the tank generally consists of two hemispherical domes or a cylinder part and head (dome) part. It shall also include mounting parts to ensure the functional interfaces (fluid connections and mechanical interfaces) of the pressurized tank with the system.

The input for the trade-off study consists of the required volume capacity and working internal pressure, material properties, geometrical relationships of the selected heads according to the analogue standards and an incremental outside diameter in order to achieve the required volume capacity with respect to the geometrical constraints of the vehicle structure. Standard design codes have been selected for the head shapes trade-off in order to benefit from the advantage that standard head shapes present a higher availability on the current market.

The first step of the study was to compute the thickness of the cylinder part considering the internal pressure load as a function of the external diameter. For every incremental external diameter, the heads' shape, thickness and inside volume were determined. To achieve the volume capacity required, the cylinder volume is computed as a difference between the total volume and the heads volume. Having the geometry of the tanks for every external diameter, the length and the mass are determined. The trade-off flow is represented in the figure below.

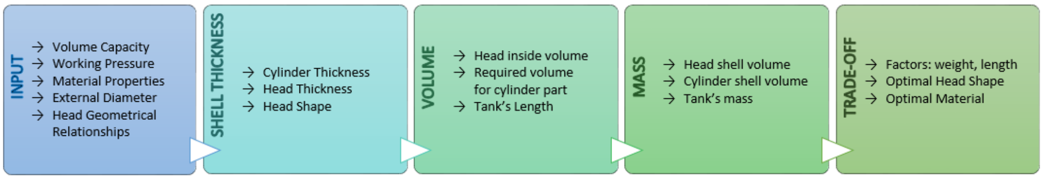

Fig. 3. Design trade-off flow

The shell of the tanks comes in various configurations such as sphere, ellipsoid, cylinder or these combinations. The shape of the tanks' shell has been selected considering the capacity, weight, manufacturability and the geometrical restraints of the available system. In this study, the following head shapes have been investigated:

a) Torispherical heads according to DIN28011 [11];
b) Semi-elliptical heads according to DIN 28013 [12];
c) Elliptical heads 1.9:1 ratio and 2:1 ratio according to NFE 81-103/ ASME code [13];
d) Hemispherical/ Spherical heads.

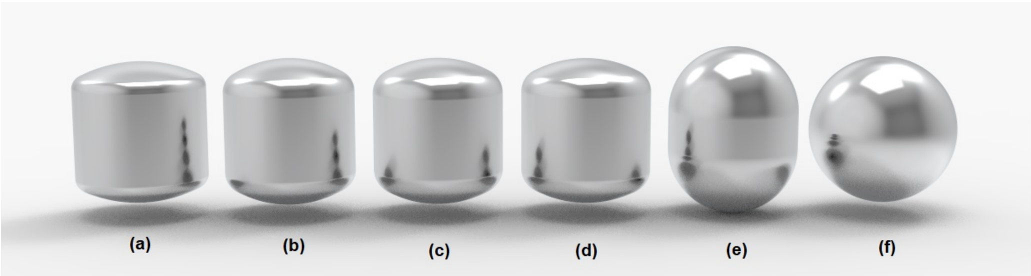

Fig. 4. Design trade-off tank configurations: (a) torispherical heads tank, (b) semi-elliptical heads tank, (c) elliptical 1.9:1 ratio heads tank, (d) elliptical 2.0:1 ratio heads tank, (e) hemispherical heads tank and (f) spherical tank

Based on the target volume, the length of the cylinder can be above or equal to "0". For example, if hemispheres shape is to be chosen as the head, and the two hemispheres are enough to achieve the target volume, the cylindrical length shall be chosen "0" obtaining a final spherical tank.

Metallic materials used for high-pressure tanks shall present specific material strength and adequate reliability. Factors that influence the mechanical properties include temperature, pressure loading time, number of cyclic loads, manufacture conditions for base material and welded parts.

For this trade-off study three (3) basic materials have been selected in order to determine the optimum weight/performance ratio for each tank of the vehicle: Stainless Steel, Al 7000 series and Ti6Al4V grade 5.

Table 2

Trade-off study material properties

Material/ Property	ρ [kg/mm^3]	E [MPa]	ν	σ_{ty} [MPa]	σ_{tu} [MPa]
AISI 316L	8.00 e-9	193000	0.25	235	560
EN AW – Al 7122	2.76 e-9	72000	0.33	410	460
Ti6Al4V, Grade 5	4.43 e-9	113800	0.31	827	896

As stated in the design trade-off flow chart, the starting point of the cylindrical tanks' sizing is the cylindrical shell thickness determination. The minimum required thickness of shell under internal pressure shall not be less than the computed by the below equations. The spherical heads configuration is treated separately since this presents a single solution for the imposed volume capacity. The thickness shall be selected in a conservative way for circumferential and longitudinal stress [14].

Cylindrical shell – longitudinal stress:

$$t = \frac{PR_e}{\sigma E + 0.4P} \quad (1)$$

Cylindrical shell – circumferential stress:

$$t = \frac{PR_e}{2\sigma E + 1.4P} \tag{2}$$

Spherical head shell:

$$t = \frac{PR_e}{2\sigma E + 1.8P} \tag{3}$$

Where:
E – joint efficiency for welded parts;
P – internal design pressure;
R_e – external radius of the shell;
σ – maximum allowed stress value;
t – minimum thickness required.

The resulting configurations have been plotted as a function of outside diameter for length and mass variables. Geometrical constraints have been imposed in order to successfully integrate the tank in the VTVL system. These constraints resulted in an area of interest in what concerns the integration in the vehicle available structure. The following figures are presenting the sizing results for the oxidizer tank according to the trade-off flow chart. The optimum design configuration has been selected in terms of mass and ergonomics.

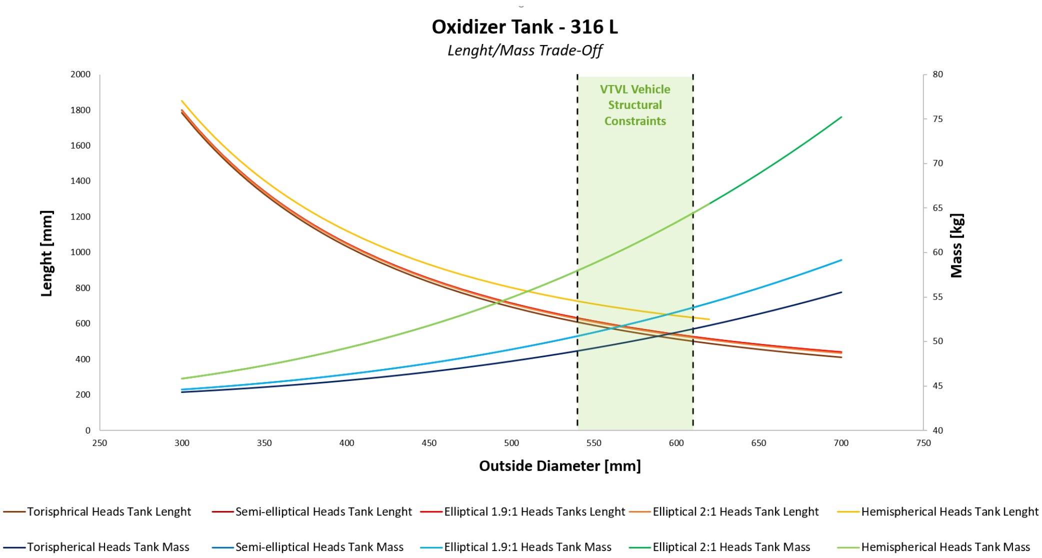

Fig. 5. Tank length/ mass trade-off – 316L

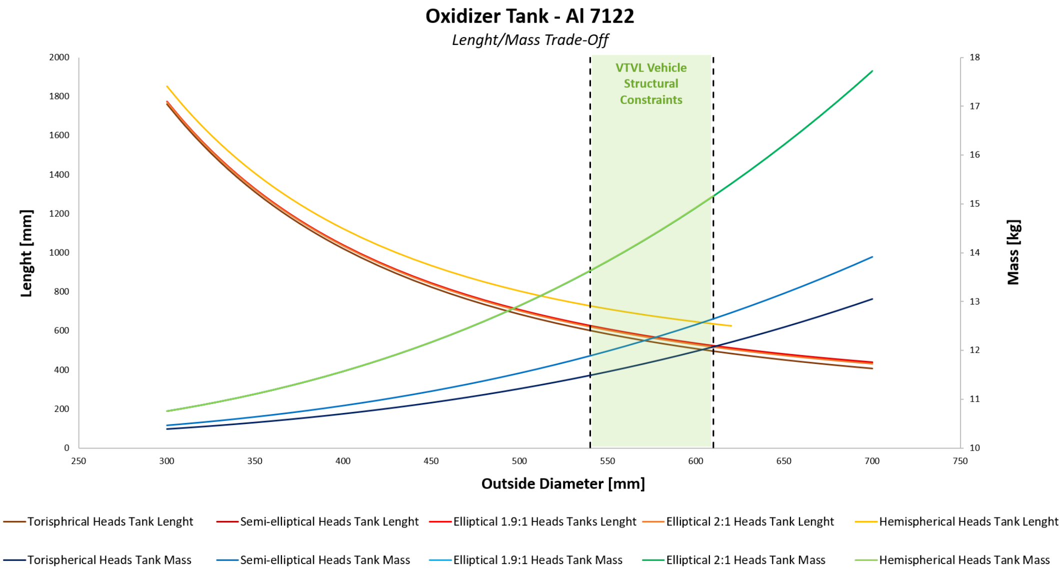

Fig. 6. Tank length/ mass trade-off – Al7122

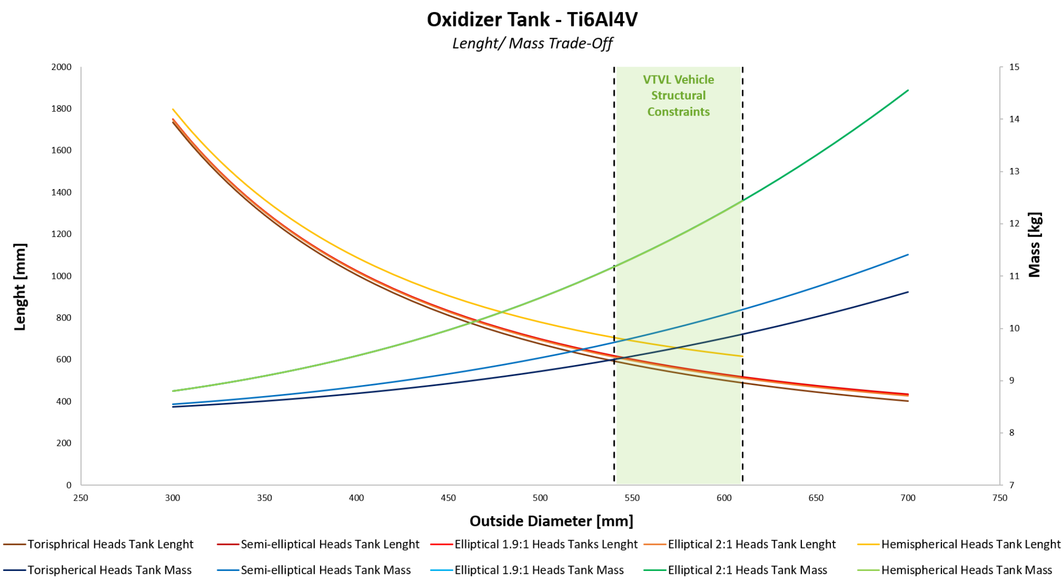

Fig. 7. Tank length/ mass trade-off – Ti6Al4V

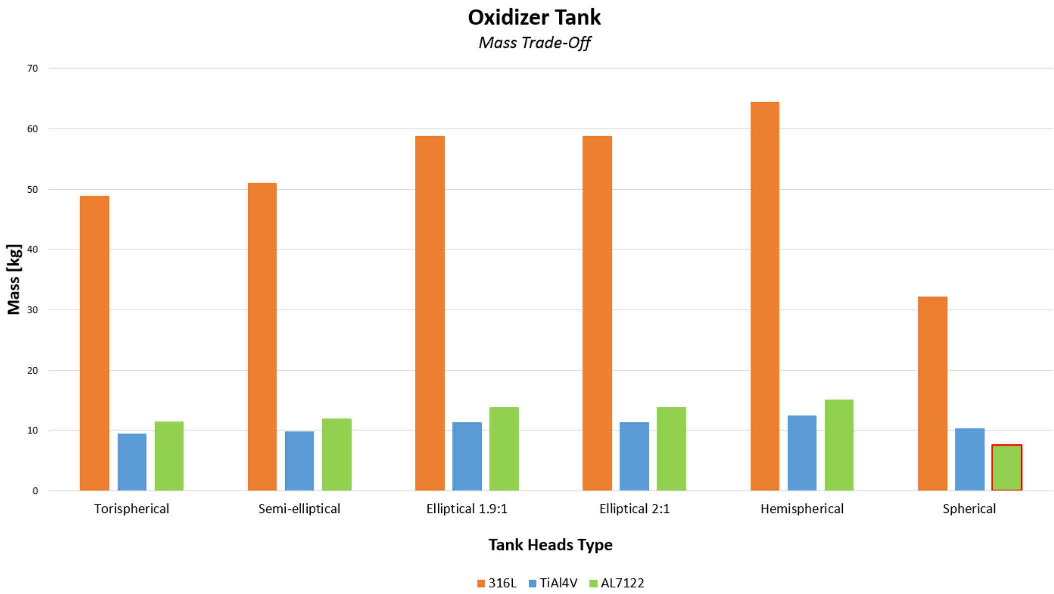

Fig. 8. Tank mass trade-off results

Fig. 8 presents the summary of the concept trade-off of the oxidizer tank. The optimal configuration is represented by a spherical tank made of Al 7122 T652 alloy.

4. Numerical Verification

This chapter presents the quasi-static analysis on the spherical oxidizer tank configuration that has resulted from the design trade-off study in order to verify and optimize the obtained analytical results.

The NASTRAN finite element model employed is shown in Fig. 9. The tank shell, brackets and support structure were meshed with 19758 CQUAD4 shell elements. The interfaces tank/ brackets and brackets/ support structure were ensured by 80 CBUSH bar elements. The welded joint was modelled using MPC elements.

Mechanical loads on the tank are derived from the difference between the pressure within the tank and the ambient conditions, fuel weight and vehicle acceleration loads. Consequently, five load cases were examined as summarized in the table below. The pressure was applied using PLOAD data type, the inertial accelerations were applied with GRAVITY data type and the fluid weight was included as Non-Structural Mass (NSM). To complete the boundary conditions, the support structure was clamped at the interface (SPC 1,2,3) as shown in Fig. 9.

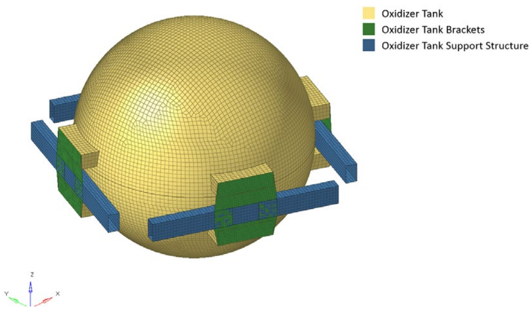

Fig. 9. Oxidizer tank FEM model

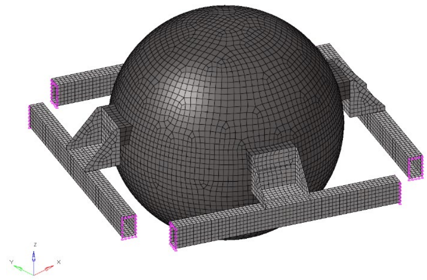

Fig. 10. Oxidizer tank boundary condition

Table 3

Load Cases

Load Case	X direction	Y direction	XY direction	Z direction	Maximum Design Pressure (MDP)
LC_01	2.5 g	-	-	-	30 bar
LC_02	-	2.5g	-	-	
LC_03	-	-	-	2.5 g	
LC_04	-	-	-	-2.5g	
LC_05	-	-	2.5 g	-	

The results were post-processed with MSC. PATRAN. An envelope of all load cases with the von Mises stress ($\sigma_{\text{von Mises}}$) was extracted from FEM. The margin of safety as follows:

Yield load margin of safety

$$MOS\_yield = \frac{\sigma_{ty}}{\sigma_{von\_mises} \cdot FOSY} - 1 \quad (4)$$

Ultimate load margin of safety

$$MOS\_ultimate = \frac{\sigma_{tu}}{\sigma_{von\_mises} \cdot FOSU} - 1 \quad (5)$$

Where:
FOSY=1.5, yield design factor of safety;
FOSU=2.0, ultimate design factor of safety.

Table 4 shows the quasi-static analysis results, demonstrating that the spherical tank configuration can withstand the operational loads. The critical zone described by the maximum von Mises stress is the welded joint influence area of the oxidizer tank shell (Fig. 10). These results confirm that the sizing optimization and optimal shape determination performed in the design trade-off study provide preliminary reliable data in the HPT development.

Table 4

Quasi-static Analysis Results

Part	Material Type	Critical Load Case	$\sigma_{vonMises}$ [MPa]	σ_{ty} [MPa]	σ_{tu} [MPa]	MOSy	MOSu
Tank Shell	EN AW 7122 T652	LC_02	199.65	410	460	0.369	0.152
Tank Bracket	EN AW 7075 T651	LC_05	50.993	460	530	5.014	4.197
Tank Support Structure	EN AW 7075 T651	LC_04	33.638	360	460	6.135	5.838

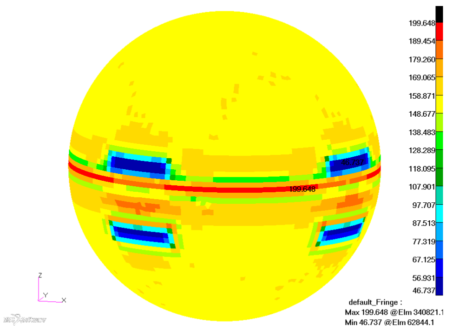

Fig. 10. Max von Mises Stress = 199.648 MPa, LC_02

5. Conclusions

A Design Methodology Roadmap is proposed where a firm product is developed and tested until a commercial of the shelf option of a high-pressure tank is developed. Great added value aside for the actual COTS option is that having this roadmap run, a supply chain, manufacturer, testing facility are identified, and connections are made with them which translates into commercial and logistical roadmap readiness and perks like trustworthy subcontractors, low-lead times. Also, trough this cycle being made, the trust of the numerical investigations can be enhanced through the "lessons learned" reducing, trough exercise, the future High-Pressure Tanks Iterations cycle.

The optimal configuration of the HPT that meet the design and performance requirements has been identified throughout the design trade-off study. The resulted spherical HPT has been evaluated by finite element structural analysis, confirming that the tank can withstand the imposed loads and that the design methodology proposed in this paper can serve as an effective guide to select an optimal configuration prior to detailed analysis and testing program.

Future research will investigate the optimum design of the shape of a filament-wound composite pressure vessel (COPV). Composite vessels represent an interesting research area because of their high strength-to-weight ratio, much higher than that of a metallic pressure vessel.

Another direction of the future work shall be taken in identifying trough numerical investigation an accurate service life for the studied configurations and search for a way to make them customizable in this regard. This is a stringent high value point of interest as the reusable VTVL demonstrators needs an extensive usage of the tanks.

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R E F E R E N C E S

[1]. *R C Carbonari, PA Munoz Rojasm, EQ Andrade, GH Paulino.,* Design of pressure vessels using shape optimization, an integrated approach, International Journal of Pressure Vessels and Piping, 2011, 198-212

[2]. *Middleton J, Owen DRJ.* Automated design optimization to minimize shearing stress in axisymmetric pressure-vessels. Nuclear Engineering and Design 1977; 44(3):357-66.

[3]. *Blachut J.* Minimum weight of internally pressurized domes subject to plastic load failure. Thin-walled Structures 1997; 27(2):127e46.

[4]. *Zhu L, Boyle JT.* Optimal shapes for axisymmetric pressure vessels: a brief overview. Journal of Pressure Vessel Technology. Nov 2000, 122(4): 443-449 (7 pages)

[5]. *Banichuk NV, Ragnedda F, Serra M.* Optimization of mass effectiveness of axisymmetric pressure vessels. Structural and Multidisciplinary Optimization 2008; 35(5):453-9.

[6]. *Magnucki K, Lewinski J. Fully* stressed head of a pressure vessel. Thin-walled Structures 2000;38(2):167-78.

[7]. *Kruzelecki J, Proszowski R.,* Shape optimization of thin-walled pressure vessel end closures. Structural and Multidisciplinary Optimization (2012), 46:739-754.

[8]. *Batchelor MJ, Taylor TE,* Equivalent torispherical pressure vessel heads. Int J Press Vessel Pip 7(3):229–244, 1995

[9]. *Muc A, Gruba W.,* Optimization of dome heads using splines conjugated with genetic algorithms. Proceedings of the Second Conference on Thin-Walled Vessels, pp 149–156, 2001

[10]. *Yushan Z, Wang ZR* Stress analysis of two-arc approximate ellipsoidal pressure vessel heads and parameter optimization. Int J Press Vessel Pip 67:199–202, 1996

[11]. *DIN 28011:1993-01* Torispherical heads, 1993-01

[12]. *DIN 28013:1993-01* Ellipsoidal heads, 1993-01

[13]. *NFE 81 103: 1997* Dished ends – Ellipsoidal dished ends – Dimensions, 1997

[14]. *ASME* BPVC Section VIII-Rules for Construction of Pressure Vessels Division 1, 2019