

Article

Extended Parametric Design of Cryogenic Liquid Hydrogen Tanks

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Abstract

The use of liquid hydrogen (LH₂) as a zero-emission energy carrier is increasingly relevant for next-generation transport systems, requiring reliable cryogenic storage solutions operating at approximately $-253\text{ }^{\circ}\text{C}$. This study presents a computational analysis of LH₂ storage tanks, focusing on the coupled thermal–structural behavior of insulated cryogenic vessels under varying design parameters. The present work expands upon our previous effort by employing the Finite Element Method (FEM) to perform a parametric investigation of these tanks. This is accomplished by examining the effect of the insulating material and the change in the storage volume of LH₂, as well as the change in the selected hydrogen boil-off rate (BOR). Results show that increasing the BOR from 0.1%/h to 1.0%/h reduces the required insulation thickness and mass by approximately 90% across all configurations, significantly altering system-level mass distribution. Substituting polyurethane foam (PUR-64) with lower-density equivalent (PUR-32) yields heat-flow reductions of up to 19.5% in two configurations, while producing an unexpected increase of approximately 10% in one case due to nonlinear thermal-gradient effects. Furthermore, increasing the hydrogen storage volume from 50 m³ to 150 m³ enhances gravimetric efficiency by 58%, 75%, and up to 104% depending on tank geometry. The results demonstrate the demand for robust numerical models, due to the nonlinear dependence of the materials involved with temperature.

Keywords: liquid hydrogen (LH₂); cryogenic storage tanks; finite element method (FEM); thermal and structural analysis; boil-off rate (BOR)



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1. Introduction

The use of hydrogen (H₂) fuel is expected to significantly influence the design and operation of next-generation transport systems. In aviation in particular, liquid hydrogen (LH₂) storage has attracted significant attention because of its potential to provide high energy density, along with low-carbon emissions. In those applications, hydrogen is most commonly stored in liquid form (LH₂) within cryogenic tanks that operate at approximately $-253\text{ }^{\circ}\text{C}$. This approach requires substantially lower storage pressures than those needed for compressed gaseous hydrogen (GH₂). On a mass basis, liquid hydrogen provides an energy density of about 2.8 times that of conventional aviation kerosene. Assuming comparable engine efficiencies and specific fuel consumption, an aircraft powered by hydrogen can theoretically achieve a considerably greater operating range than a similar

aircraft using kerosene [1–4]. However, despite its high gravimetric energy density, LH₂ exhibits a much lower volumetric energy density. As such, the precise management and incorporation of the LH₂ storage tank is of outmost importance. In addition to storage capacity considerations, LH₂ tank design must address a number of requirements, including thermal insulation, hydrogen boil-off losses, structural integrity under cryogenic conditions, and overall gravimetric efficiency. These requirements make LH₂ tank design a complex thermo-structural engineering problem.

Several studies have investigated the integration of LH₂ storage systems into both conventional and unconventional aircraft configurations. These include modified commercial aircraft, blended-wing-body concepts, Flying-V configurations, forward-swept-wing aircraft, unmanned aerial vehicles, and other hydrogen-powered aircraft concepts [5–18]. Collectively, these studies demonstrated the feasibility of LH₂-fueled aviation and highlighted the importance of tank placement, aircraft integration, weight management, and storage-system performance. However, most of these investigations focused primarily on aircraft-level design considerations, while comparatively less attention was devoted to the detailed thermo-structural behavior of the LH₂ storage tank itself. The ENABLEH2 project concluded that LH₂ aircraft fuel systems are technically feasible for 2035, with optimal large-diameter tanks achieving a 65–80% gravimetric index [19]. Key findings emphasize utilizing Al-Li alloys or CFRP, managing 24-h boil-off through re-utilization in thermal management systems, and noting that increased fuselage size raises operational weight at empty by approximately 23%. Since then, research has expanded towards aircraft integration, tank placement, structural design and thermal performance.

Beyond aircraft or aerospace integration, which remain the most research topics for popular use, LH₂ storage could be a critical enabling technology for stationary storage facilities and ground transportation systems. Recent reviews have highlighted that, despite the diversity of these applications, the key challenges governing LH₂ storage performance remain heat ingress, boil-off losses, insulation effectiveness, and the strong temperature dependence of hydrogen thermophysical properties [20]. Relevant investigations addressing the thermal and structural behavior of LH₂ storage tanks include the studies conducted by Tapeinos et al. [21–23]. The authors performed both numerical and experimental analyses of an LH₂ tank featuring a configuration of interconnected spherical compartments, consisting of polymer and composite double-wall structures separated by an insulating foam layer. The tank performance was evaluated under different combinations of temperature and pressure conditions, while the requirement for structural reinforcement in critical regions of the external surface was also examined. Using the MRT7-3 Meridian medium-range aircraft as a reference, Gomez and Smith [24] conducted an analysis using the Finite Element Method (FEM) to investigate the integration of LH₂ storage tanks. Their study examined the structural modifications required for the existing airframe and evaluated the effects of the tank's internal temperature and pressure conditions on the overall design. In their study, Bagarello et al. (2026) leveraged a layered architecture consisting of an aluminum liner (Al 2219), a polymeric insulation core (PUR), and a carbon-fiber-reinforced plastic (CFRP) outer shell to minimize system-level weight penalties while avoiding the complexity of active venting sub-systems [25]. Comprehensive reviews by Bagarello et al. [26] confirm that passively insulated cryogenic composite sandwich pressure vessels represent the most promising technology for near- to mid-term aircraft integration, eliminating the need for complex, heavy active vacuum subsystems. In addition to individual component design, the recent literature has expanded toward full systems-level conceptual architectures. For instance, Karaiskos et al. [27] coupled tank sizing with a one-dimensional distribution system model (including pipelines, booster pumps, and regulator valves) to assess the

thermal-hydraulic penalties and overall weight impact along the entire fuel path from the fuselage storage units to the engine interface.

Nevertheless, recent reviews have identified a continuing lack of studies focusing specifically on aviation-oriented LH₂ tank design, particularly regarding the influence of tank geometry on storage performance [28]. Therefore, further investigation of the effects of tank geometric configuration remains necessary to support the development of efficient hydrogen-powered aircraft. While aerospace tanks are additionally subjected to dynamic operating conditions such as acceleration, vibration, and varying gravity levels, understanding the underlying thermal behavior of LH₂ storage systems under controlled conditions constitutes a necessary first step toward the development of more advanced aerospace-specific models. As such, the present work focuses on the thermal performance of a representative LH₂ storage tank under steady conditions, providing design insights and modelling methodologies that can subsequently be extended to aircraft-integrated storage systems.

In our previous work [1], a parametric thermo-structural analysis of LH₂ storage tanks was performed with emphasis on the influence of tank geometry and LH₂ fill level. The study demonstrated the importance of accounting for temperature-dependent material properties and resulted in preliminary sizing guidelines for aircraft-integrated LH₂ storage systems.

To the authors' knowledge, no previous study has systematically quantified the combined influence of storage volume, structural material selection and prescribed boil-off-rate requirements on the thermo-structural sizing of LH₂ tanks using temperature-dependent material properties. This constitutes the primary contribution of the present work. The analysis is extended by investigating additional design parameters that are critical during the preliminary design stage, namely the structural material selection, storage volume, and prescribed boil-off-rate (BOR) requirements. Furthermore, the influence of geometric characteristics associated with different curvature levels is examined through geometry-specific governing relations. The objective is to establish a broader set of thermo-structural design guidelines capable of supporting the selection and sizing of LH₂ storage tanks for future hydrogen-powered transportation systems.

A static numerical investigation based on the FEM is performed to assess the behavior of these tanks under different design conditions. The study explores the influence of alternative structural materials, variations in hydrogen-storage volume, and changes in the selected storage configuration. Dynamic flight-related effects, such as aircraft maneuvers, vibration and fuel sloshing, are not considered in the current investigation and constitute a topic for future work. Based on the obtained results, a set of recommendations for the preliminary sizing of LH₂ tanks is developed to support design decisions during aircraft integration studies. The analysis also highlights the importance of robust numerical modeling techniques, as the material properties involved exhibit temperature-dependent nonlinear behavior. Further improvements in computational efficiency could be achieved by incorporating the proposed models into a comprehensive aircraft-design optimization framework.

2. Materials and Methods

2.1. Properties of the Tank's Materials

In this chapter, the detailed description of the LH₂ storage tank problem that has already been carried out in detail in the previous study [1], and the computational procedure on which the analyses were based, are introduced.

The tank configuration of the tank consists of the following three structural elements [1]: An inner shell that is in contact with the LH₂ (usually referred to in the lit-

erature as the “lining”) made of aluminum alloy, insulation material made of polyurethane foam (PUR), and an outer shell made of layers of composite material. These foams offer a lightweight, cost-effective passive insulation core, while an outer carbon-fiber-reinforced polymer (CFRP) shell provides an exceptional strength-to-weight ratio [26]. Placing the composite shell on the exterior side of the insulation minimizes thermal expansion coefficient (CTE) mismatches and thermally induced stresses by keeping the layer close to ambient room temperatures [26].

The materials used are considered to be homogeneous and linear elastic, as this study focused on their linear elastic region [1]. Regarding the representation of the external composite shell, it should be noted that the thermal conditions of the problem imply that this component remains at a temperature close to the assumed ambient value of 10 °C (283 K). As a result, the characterization of the composite material under cryogenic conditions is not necessary. For the modeling of the external composite shell, a composite material made of a polymer matrix with carbon fibers (Composite Fiber Reinforced Polymer—CFRP) from the company Hexcel© (Hexcel Corporation, Stamford, CT, USA), namely AS4/8552, transversely isotropic, was examined. The required thermal and mechanical properties of a single layer (ply) are included in Table 1 [21–23].

Table 1. Properties of the composite material (CFRP) of the outer tank shell [1].

Property	Value
Density, ρ	1600 kg/m ³
Thermal conductivity, k	$k_{11} = 3.972$ W/mK, $k_{22} = k_{33} = 0.3363$ W/mK
Specific heat capacity, C_p	1000 J/kg K
Coefficient of thermal expansion, α_{ii}	$\alpha_{11} = -0.2 \times 10^{-6}$ m/m/K, $\alpha_{22} = \alpha_{33} = 30 \times 10^{-6}$ m/m/K
Young’s modulus, E_{ii}	$E_{11} = 133$ GPa, $E_{22} = E_{33} = 9.13$ GPa
Poisson ratio, ν_{ij}	$\nu_{12} = \nu_{13} = 0.3$, $\nu_{23} \cong 0.3$
Shear modulus, G_{ij}	$G_{12} = G_{13} = 5.2$ GPa, $G_{23} \cong 5.2$ GPa
Ply longitudinal tensile strength, X_t	2178.2 MPa
Ply longitudinal compressive strength, Y_t	91.7 MPa
Ply transvers tensile strength, X_c	1783.5 MPa
Ply transverse compressive strength, Y_c	340.9 MPa
Ply in plane shear strength, S	129.1 MPa

The thickness of the material layer is equal to $t_{\text{ply}} = 0.00019$ m [1]. As for the material layering, a pseudo isotropic layering of 4 layers (0, 45, 90 and -45°) was used. The final stratification is [0/45/90/ -45° / -45° /90/45/0] = [0/45/90/ -45°] s, with a total thickness of 0.00152 m. The change in the thickness or the stratification of the composite shell was not studied, so the thickness and the stratification remain unchanged in the analyses that follow.

For the insulating material layer, polyurethane foam (PUR) with a density of 64 kg/m³ is considered. This particular PUR is called PUR-64, with the suffix number denoting the material density. Like its PUR-64 counterpart, PUR-32 derives its name from the fact that its density is 32 kg/m³.

The two foams are made from the same parent material, with the only difference being their density. The thermal and mechanical characteristics of the foams were obtained from the experimental investigation conducted by Sparks and Arridson [29]. Unlike the CFRP outer shell material, the materials of the inner shell and the insulation exhibit non-linearity in their characteristics. This “non-linearity” is the result of the materials’ properties’ dependence on temperature, as will be analyzed in this section.

For high-performance LH₂ storage systems, vacuum multilayer insulation (MLI), perlite insulation, hollow glass microspheres and vapor-cooled shield technologies are more widely adopted [30]. However, PUR is characterized as an effective and well-established

material in terms of its thermo-mechanical properties and its general behavior, along with its low cost [1] (Figure 1). It is commonly used as a core material in sandwich structures, featuring a thickness significantly greater than that of the outer layers. Its past applications in aerospace cryogenic tanks—most notably in the external tank of the Space Shuttle—prove its effectiveness as a reliable yet simple heat barrier, simultaneously offering highly tailorable insulation properties through spray-on deposition techniques, allowing for straightforward thickness management [25].

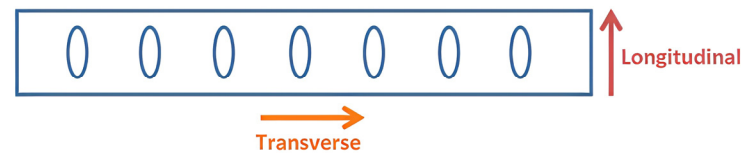


Figure 1. Sketch of a layer of polyurethane foam, with the directions of its anisotropic behavior according to [29]. The ellipses represent the bubbles of the foam. Contrary to common practice, the direction along the thickness is referred to as “longitudinal” and the direction in the plane as “transverse”.

For the analysis, the following thermos-mechanical properties are required:

- (a) The density, ρ (two different densities, 32 kg/m^3 and 64 kg/m^3) [29].
- (b) The thermal conductivity, k (see Figure 2a).
- (c) The specific heat capacity, C_p .

Sparks and Arridson [29] did not include results concerning the evolution of the specific heat capacity of materials as a function of temperature, and no relevant experimental results from other works were found. Therefore, a constant value was selected, as estimated in a relevant technical document [31].

- (d) The CTE (Figure 2b).

(e) The modulus of elasticity (Young’s modulus of elasticity), E . In the case of foams, which are materials with orthotropic characteristics, for a layer of them there is a modulus of elasticity in the directions that define the plane of the layer (characterized as transverse [29]) and a different modulus in the direction of the thickness (characterized as longitudinal [29]) (Figure 1). Therefore, the material is transversely isotropic, since it is characterized by two moduli of elasticity. In addition, there is also a difference in the modulus of elasticity in tension and compression (Figure 2c,d).

(f) Poisson’s ratio, ν . Due to the lack of results similar to those of the modulus of elasticity in general in the literature, it was chosen to maintain a constant Poisson’s ratio, at all temperatures, of $\nu = 0.3$.

(g) For the shear modulus, G , which, like the Poisson’s ratio, due to the lack of relevant results in general in the literature, it was chosen to maintain a constant shear modulus, at all temperatures, $G = 20 \text{ MPa}$.

(h) The values of the material’s strength in tensile, compressive and shear stresses, so that they can be used to calculate the Tsai-Wu criterion, which is used to evaluate the structural integrity of the foam (Figure 2e–g).

Aluminum-based tanks are characterized by their high Technology Readiness Level (TRL) [32]. Specifically, aluminum alloy Al 2219-T8 is the leading candidate for the inner liner due to its established manufacturing know-how, low cost and very low susceptibility to hydrogen embrittlement [1,26]. For the tank cases of this paper, the storage chamber is fabricated from Al 2219-T87, an alloy that has been extensively validated and thoroughly characterized for use in cryogenic environments [33]. The required physical, thermal and mechanical properties of the specific state of aluminum are the same as the one described before. The density ρ is 2840 kg/m^3 [34], while the rest are acquired through Figure 3a–f.

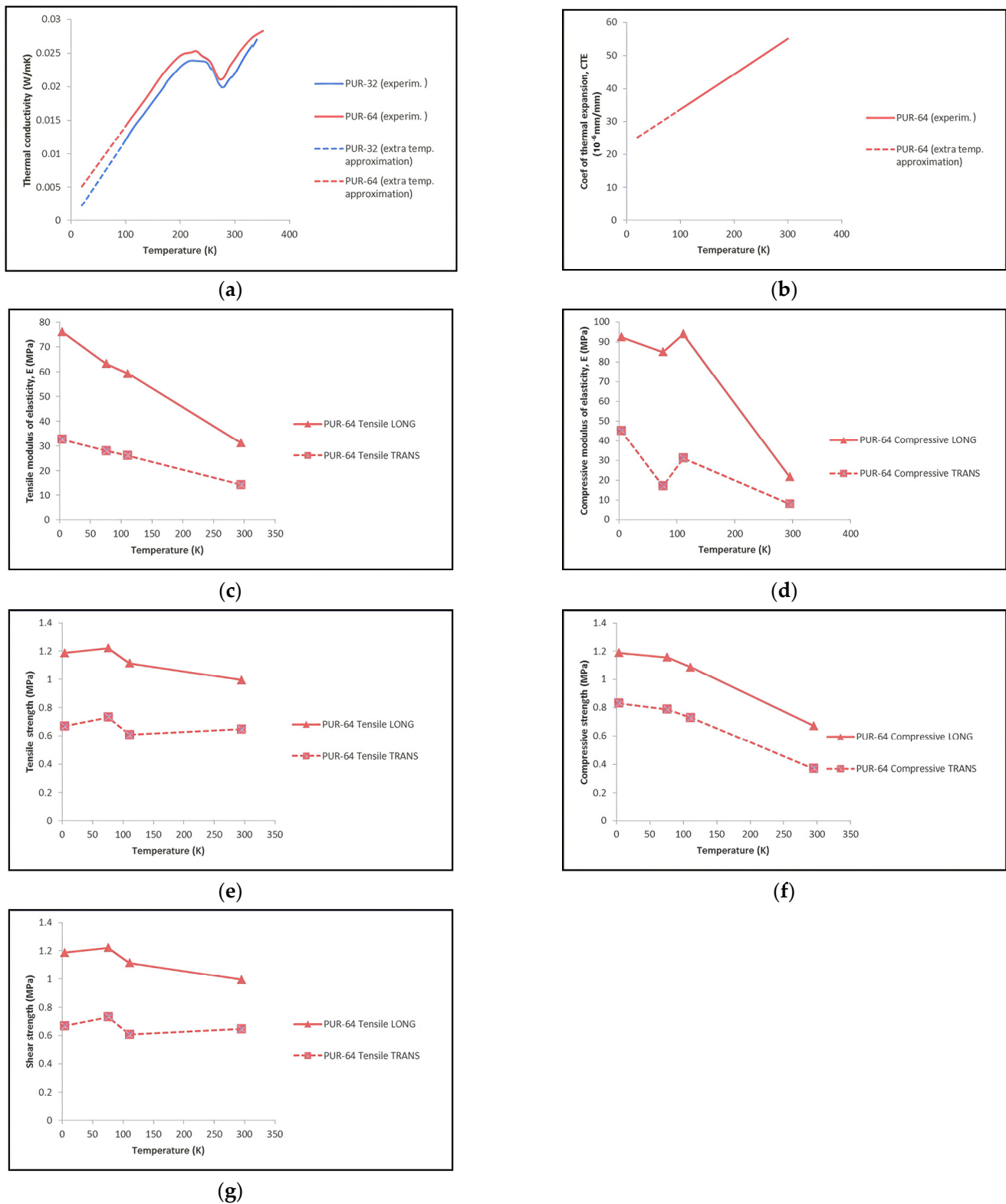


Figure 2. Temperature dependent properties of PUR-32 and PUR-64, based on experimental results by [29]. (a) Thermal conductivity in relation to temperature, (b) CTE in relation to temperature, (c) Tensile modulus of elasticity in relation to temperature, (d) Compressive modulus of elasticity in relation to temperature, (e) Tensile strength in relation to temperature, (f) Compressive strength in relation to temperature, (g) Shear strength in relation to temperature.

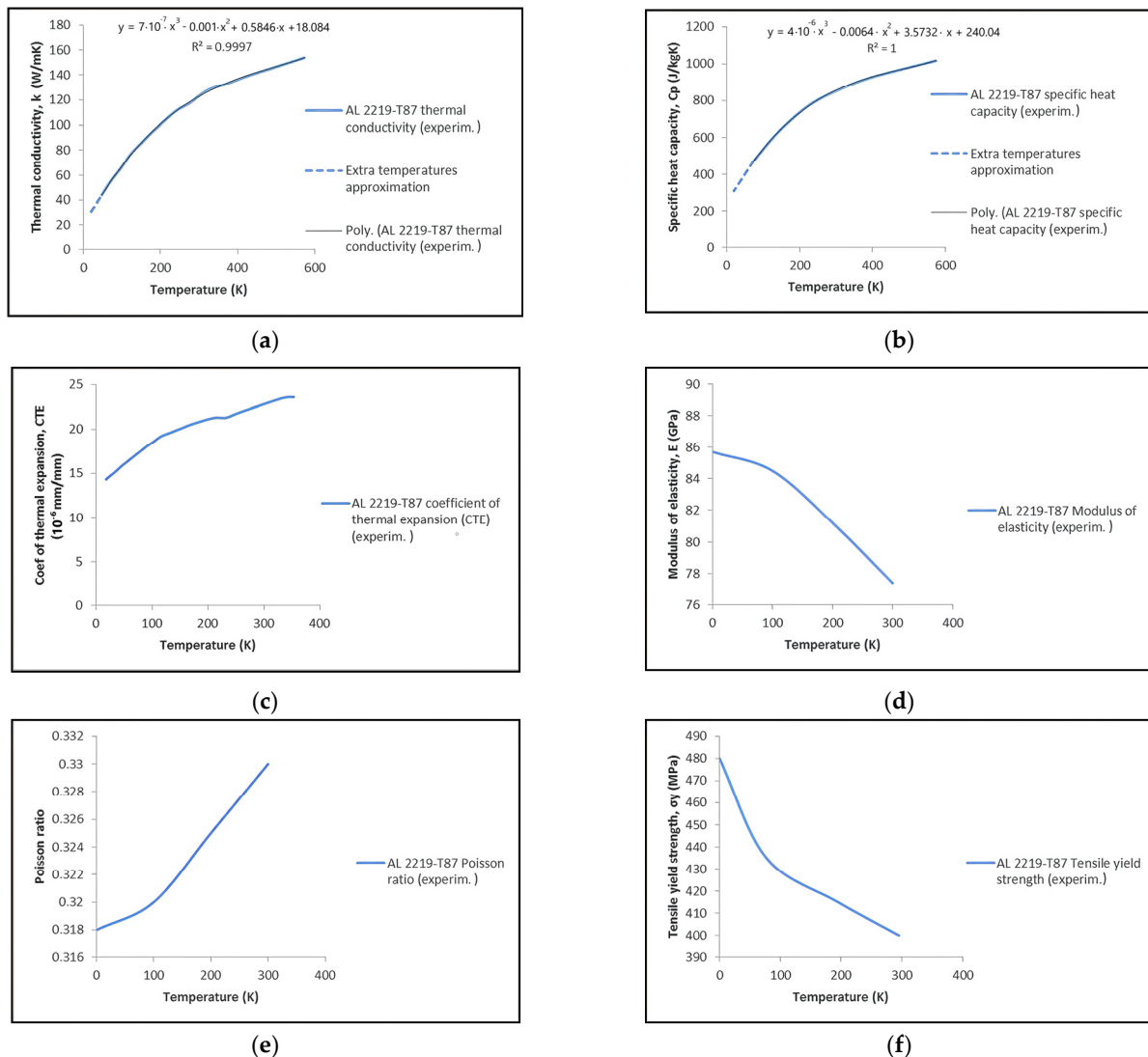


Figure 3. Temperature dependent properties of Al 2219-T87, based on experimental results by [35]. (a) Thermal conductivity in relation to temperature, (b) Specific heat capacity C_p in relation to temperature, (c) CTE in relation to temperature, (d) Modulus of elasticity E in relation to temperature, (e) Poisson ratio in relation to temperature, (f) Tensile yield of strength σ_y in relation to temperature.

Due to the expected thin thickness of the aluminum, it is acceptable that the temperature change between its surface that is in contact with the LH_2 , and the side that is in contact with the insulating material, is relatively small, so that it can be considered that the thermo-mechanical properties of the aluminum do not change with its thickness. Information on these properties was derived from the work of Simon et al. [35], who examined its behavior over a wide range of temperatures. The diagrams that follow (Figure 3a–f) present the evolution of these properties over the entire range of temperatures estimated experimentally. The temperature range of the experiments in some cases did not include the cryogenic temperature considered in this work (-253°C (20 K)). Therefore, the complete experimental value diagram was used to perform a polynomial approximation, based on which estimates were made for the values of the relevant quantities for the remaining temperature range between the lowest experimental temperature and that of -253°C (20 K) of the present study.

The tanks examined in this study are classified as non-integrated storage systems. They do not contribute to the aircraft's primary load-bearing structure and serve exclusively as LH_2 containment units operating under pressure. Steady-state thermal and pressure

loading conditions are adopted, primarily because the study is intended to support the preliminary design and sizing of LH₂ storage tanks. At this stage, the objective is to identify the fundamental thermo-structural trends associated with tank geometry, material selection, storage volume, and BOR requirements. Consequently, the tank is analyzed as an isolated storage system under controlled operating conditions. Figure 4 illustrates a cross-sectional view of the tank configuration, highlighting the regions and components considered in the present investigation.

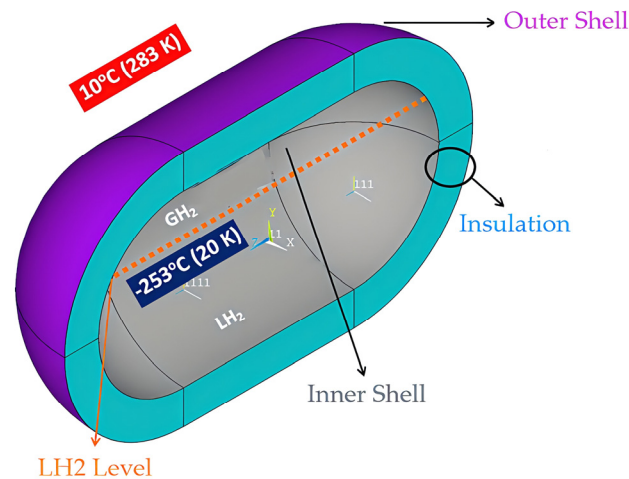


Figure 4. Sections, materials and boundary conditions of the tank.

Real aircraft operation involves transient effects arising from flight maneuvers, environmental variations, vibration, fuel sloshing, and changing acceleration fields. Studying time-dependent behavior, like transient heat transfer, vapor generation (phase change), and pressure evolution, is vital. However, the consideration of steady-state conditions enables extensive parametric investigations at a reasonable computational cost, facilitating the systematic evaluation of multiple design variables during the early stages of the design process. Furthermore, the adoption of steady-state conditions is consistent with the methodology employed in the authors' previous study [1], thereby enabling a direct assessment of the additional design parameters investigated herein.

2.2. Description of the Tank's Geometry

To generate the different tank configurations investigated in this study, the approach based on triaxially symmetric geometries proposed by Winnefeld et al. [3] was implemented. Thus, three characteristic dimensionless parameters, λ , ϕ and ψ , were derived to describe its shape and geometry (Figure 5). These parameters are associated with the dimensions a , b and c of Figure 5, which are highlighted as the semi-axes of three ellipses (a – b , a – c , b – c). Together, along with the length of the central part of the tank, l_s , provide a complete geometric description of the tank.

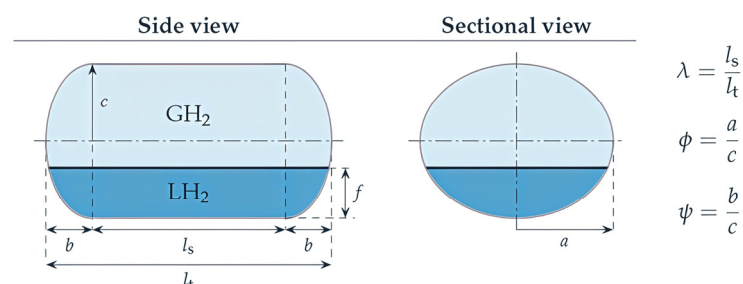


Figure 5. Geometry of the tanks, expressed through ratios of their characteristic dimensions [3].

It should be noted that the schematic description of Figure 5, for the purposes of this work, expresses the geometry of the inner aluminum shell (liner) in which the stored LH₂ is enclosed. The dimensions and the tank size ratios are connected through Equations (1)–(6) [1,3]. Dimension f describes the LH₂ level inside the tank.

$$l_t = \left[\frac{V_{total}\psi^2}{\pi\varphi} \left(\frac{(1-\lambda)^3}{6} + \frac{\lambda(1-\lambda)^2}{4} \right)^{-1} \right]^{\frac{1}{3}} \quad (1)$$

$$b = \frac{l_t - l_s}{2} \quad (2)$$

$$\lambda = \frac{l_s}{l_t} \quad (3)$$

$$\varphi = \frac{a}{c} \quad (4)$$

$$\psi = \frac{b}{c} \quad (5)$$

$$V_{LH_2} = \frac{al_s}{c} \left[(f-c)\sqrt{c^2 - (f-c)^2} + c^2 \arcsin\left(\frac{f-c}{c}\right) + c^2 \frac{\pi}{2} \right] + \frac{\pi abf^2}{3c^2} (3c-f) \quad (6)$$

Equation (6) can be used to derive dimension f iteratively, as there is no analytical solution. V_{total} is the total volume of the hydrogen storage space, while V_{LH_2} is the volume occupied only by LH₂. LH₂ tanks are never filled to 100% capacity primarily to leave a safety margin for boil-off gas (BOG). In this study, the maximum LH₂ filling percentage is of a (maximum) level of 93% [1,36]. Table 2 presents the actual LH₂ storage capacities corresponding to the three tank sizes investigated, which could facilitate an engineering interpretation of the results.

Table 2. Actual LH₂ storage capacities corresponding to the three tank sizes.

Tank Volume V_{total} [m ³]	Liquid Hydrogen Volume V_{LH_2} [m ³]	Liquid Hydrogen Mass m_{LH_2} [t]
50	46.5	3.3
100	93.0	6.6
150	139.5	9.9

2.3. Thermal and Structural Analysis

According to [1], the enclosed volume defined by the inner aluminum liner is considered as a primary design variable throughout the analyses performed in this study (Table 2). The hydrogen temperature is assumed to remain at -253 °C (20 K), while the internal pressure within the aluminum shell is maintained at 170 kPa (Figure 4). For the external surroundings, an ambient temperature of 10 °C (283 K) is adopted as a representative condition, without imposing any restrictive limitation on the analysis. The ambient temperature serves as a reference operating condition for comparison among tank configurations and is not intended to represent the full range of aircraft operating environments.

Regarding the thermal aspect of the analysis, heat transfer between the external environment and the hydrogen enclosed in the internal shell of the tank is governed by all forms of heat transfer. The fundamental transient heat transfer equation solved in the computational environment is:

$$\rho c \frac{\partial T}{\partial t} = \vec{\nabla} \cdot (k \cdot \vec{\nabla} T) + Q \quad (7)$$

Since only steady-state phenomena are examined, Equation (7) can be written as:

$$\vec{\nabla} \cdot (\mathbf{k} \cdot \vec{\nabla} T) + Q = 0 \quad (8)$$

where Q [W] is the volumetric heat generation rate and T [K] is the temperature. Equation (9) describes the thermal conductivity vector $\mathbf{k}$ [W/mK]:

$$\mathbf{k} = \begin{bmatrix} k_{11} & k_{21} & k_{31} \\ k_{12} & k_{22} & k_{32} \\ k_{13} & k_{23} & k_{33} \end{bmatrix} \quad (9)$$

In more detail, the following phenomena are examined:

- Heat transfer between the external environment (T_∞) and the composite material outer shell (T) [1] (Figure 4), via radiation, based on the Stefan–Boltzmann Law:

$$-\mathbf{k} \cdot \vec{\nabla} T \cdot \vec{n} = \epsilon \sigma (T^4 - T_\infty^4) \quad (10)$$

where σ is the Stefan–Boltzmann constant. In this case, it is important to determine the emissivity ϵ . In order to minimize the influence of this type of heat transfer, the tank is considered to be wrapped externally with a thin aluminum foil, which has a fairly small emissivity ($\epsilon = 0.07$ [37]). This choice of thin aluminum foil is noted as it does not affect the thermal and structural behavior of the tank in any other way, so in all other matters it is not taken into account, nor is it modeled as a separate component of the tank, except through the aforementioned value of the emissivity.

- Heat transfer between the composite shell and the insulating material, as well as between the insulating material and the aluminum shell [1], as described by Fourier's Law:

$$\vec{q} = -\mathbf{k} \cdot \vec{\nabla} T \quad (11)$$

- Heat transfer between the external environment and the composite shell due to convection [1]. Newton's Law of cooling is utilized:

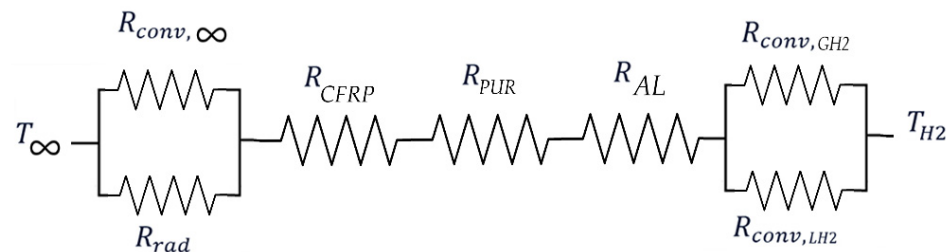
$$-\mathbf{k} \cdot \vec{\nabla} T \cdot \vec{n} = h_\infty (T - T_\infty) \quad (12)$$

In this case, the coefficient of convection h_∞ [W/m²K] refers to the undisturbed air flow of the external environment, since it is assumed that the tanks under consideration, as “non-integrated”, are stored in some internal space of the aircraft in which they are used. Therefore, the ambient air velocity can be considered for the purposes of the work as zero. Representative values reported in the literature were adopted, and specifically an indicative value equal to 10 W/m²K was taken, within the typical range of convection coefficients related to free air flow (2.5–25 W/m²K, [37]).

- Heat transfer between the aluminum shell and the enclosed hydrogen [1]. In this case, Equation (12) is used again, but with two different convection coefficients, one for LH₂ and one for GH₂. The values of the convection coefficients are shown in Table 3 [38,39], while a sketch of the thermal network of the geometry is presented in Figure 6:

Table 3. Convection coefficients values [37–39].

Model Section	Convection Coefficient h [W/m ² K]
Environment—Outer Shell (Free air flow)	10
LH ₂ —Inner Tank Environment	1.4587
GH ₂ —Inner Tank Environment	2.1827

**Figure 6.** Schematic of the thermal resistances of the tank.

The convective heat-transfer coefficients at the liquid– and gaseous–hydrogen interfaces were adopted from Dagsever’s work [38] based on experimental data from Faure [39]. These were derived for cryogenic hydrogen under comparable storage conditions, for tank sizes and geometries similar to the ones in this study. The adopted convection coefficients represent average steady-state values rather than transient local coefficients. Assuming them to be constant, isolates the influence of tank geometry, insulation material and BOR on the overall thermo-structural behavior. During actual storage, the thermodynamic state of hydrogen evolves continuously due to boil-off, pressure changes and thermal stratification, which may alter the local heat-transfer coefficients. These effects were intentionally neglected in order to focus on comparative thermo-structural trends and to simplify the computational process. As no sensitivity analysis of the hydrogen-side convection coefficients was performed, the results of this study should be interpreted within the context of these assumed thermal boundary conditions.

The BOR is defined as the percentage of cryogenic liquid that vaporizes per unit time due to heat leakage into the storage system [1]. It is a measure of storage-tank thermal performance and is directly associated with BOG generation, pressure rise, and hydrogen or cryogen losses [40]. The acceptable BOR strongly depends on the intended application and mission duration. Aviation applications, lasting several hours or even days, require substantially lower BOR values than space applications. Acceptable BOR for space applications, according to Mital et al. [41], are on the order of 1.5%, while for aviation applications they are on the order of 0.1% (a difference of one order of magnitude). GH₂ produced due to the influence of the external environment must be released through appropriate mechanisms (e.g., pressure valves) outside the tank to avoid an increase in pressure inside the tank. It is very logical to consider that a long-range or high-endurance aircraft, where the mission profile is quite time-consuming, will have a strict requirement for a low BOR, because hydrogen losses, given the operation time, will be expected to be significant. The storage system must include pressure-management provisions because hydrogen vapor generated by boil-off continuously increases tank pressure.

The thermal performance of each tank configuration was first evaluated through a steady-state thermal analysis. The insulation thickness was iteratively adjusted until the prescribed BOR was achieved. The allowable heat flow was then determined from Equation (13) [38]:

$$BOR = \frac{360000 \cdot Q}{\Delta h \cdot V_{LH_2} \cdot \rho_{LH_2}} \left[\frac{\%}{hour} \right] \quad (13)$$

where:

- Q [W] represents the overall heat transfer rate exchanged between the LH_2 contained within the tank and the surrounding environment [1];
- Δh [J/kg] denotes the latent heat of vaporization of LH_2 , which is assumed to be 461,000 J/kg [1];
- m_{LH_2} [kg] corresponds to the LH_2 mass inside the tank. It is calculated as the product of the volume V_{LH_2} and its density, ρ_{LH_2} . The latter's value is assumed to be 71 kg/m³ [1].

The structural response of the tank is influenced by two primary loading mechanisms: the internal pressure imposed on the aluminum containment shell and the thermally induced stresses associated with the temperature variations across all constituent components of the tank [1]. The Von-Mises, Tsai-Wu and Hashin failure criteria are utilized [1]. Structural phenomena such as fatigue, buckling, fracture mechanics, and interfacial stresses between tank layers are beyond the scope of this study. As such, the tanks under consideration are assumed to simply rest on the ground, not bounded or installed on a specific aircraft structure.

2.4. Computational Analysis

For the computational analysis [1], a process of specific steps is utilized. Firstly, the required insulation thickness required to achieve the heat flow dictated by the selected hydrogen BOR is calculated. This is done by conducting repeated iterative thermal analyses. The second step is to introduce the calculated temperature distribution as a boundary condition (along with the condition of the inner pressure of 170 kPa) to all the nodes of the tank and perform a structural analysis. The selected boundary conditions correspond to a representative reference operating condition rather than a worst-case scenario. Lastly, the thickness of the inner aluminum shell is calculated, through iterative analyses, until the Von Mises criterion is satisfied.

The tank components are discretized using three-dimensional (3D) 8-node finite elements. Thermal analyses are performed using the SOLID278 element, which has one degree of freedom (DOF) per node (temperature), while structural analyses employ the SOLID185 element, which has three translational DOFs per node (displacements along the x, y and z axes). Both element types are available in homogeneous and layered formulations within the APDL (ANSYS Parametric Design Language) environment [42].

Prior to the parametric analyses, a mesh convergence study was performed in our previous work [1] using a representative tank configuration ($\lambda = 0.5$, $\varphi = 1$, $\psi = 1$, internal volume 100 m³). The study investigated three in-plane mesh sizes (0.15, 0.10 and 0.05 m) combined with three through-thickness discretizations of the insulation layer (4, 6 and 8 elements). Mesh independence was assessed based on the predicted heat leak, maximum and minimum temperatures, and temperature profiles through the insulation thickness. The convergence study showed that the in-plane mesh size had a negligible influence on the thermal response, whereas the number of elements through the insulation thickness had a greater effect on the predicted heat leak and temperature distribution. Based on these results, an in-plane element size of 0.10 m together with eight elements was used through the insulation thickness, providing an appropriate balance between accuracy and computational cost. In addition, the same governing equations, finite-element formulation, material models, loading conditions, and numerical solution procedure as our previous investigation [1] are utilized. Therefore, since only the tank geometry and design parameters are varied, the previously validated mesh configuration was retained for all thermal and structural analyses.

3. Results

3.1. Use of a Different Insulating Material

For the purpose of comparing the effect of the material selected to ensure the required thermal behavior of the tanks, three arrangements are selected from the forty-five studied in the previous work [1]. The insulation material is replaced, going from the original PUR-64 to PUR-32, in order to determine its effect on the thermal behavior of the tanks. For each arrangement, the required thickness of the insulating material has been calculated as described in the previous section, in order to meet the LH₂ BOR. The three Tanks selected are represented by their distinct geometric parameters:

- (a) $\lambda = 0.25$, $\varphi = 1$, $\psi = 0.666$, with a required insulation thickness of 0.61 m;
- (b) $\lambda = 0.50$, $\varphi = 0.333$, $\psi = 0.333$, with a required insulation thickness of 0.77 m;
- (c) $\lambda = 0.75$, $\varphi = 0.666$, $\psi = 1$, with a required insulation thickness of 0.93 m.

It is recalled that in [1], the required insulation thicknesses of the tanks were calculated by taking into account the requirement to maintain the BOR at the maximum allowed value of 0.1%/h. This resulted in the determination of the maximum allowed value for the heat flow from the external environment to the hydrogen storage chamber to be 845 W. In order to carry out the comparison of the tanks, the insulation thickness calculated for the three selected tanks is maintained intact, and the heat flow from the external environment of the tank to the hydrogen chamber is calculated, using the differentiated PUR-32 insulating material. This results in Table 4:

Table 4. Change in total heat flow in the tank as a function of the insulation material, for three tank cases. An increase in the heat flow is symbolized by an ↑ arrow, while a decrease is symbolized by a ↓ arrow, in the Shift [%] column.

Tank Geometry	λ	φ	ψ	Insulation Material	Insulation Thickness [m]	Heat Flow [W]	Shift [%]
<i>a</i>	0.25	1	0.666	PUR-64	0.61	845	↑ (+10.3%)
				PUR-32		932	
<i>b</i>	0.50	0.333	0.333	PUR-64	0.77	845	↓ (−19.5%)
				PUR-32		680	
<i>c</i>	0.75	0.666	1	PUR-64	0.93	845	↓ (−19.5%)
				PUR-32		682	

Based on the study of Sparks and Arridson [29], for each value of temperature, the thermal conductivity of the insulating material PUR-32 is lower than that of PUR-64. Consequently, the PUR-32 material should be a more effective insulator compared to PUR-64. This behavior is observed in Tanks *b* and *c* since the heat flow to the hydrogen storage chamber decreases (about 19.5%). However, in Tank *a*, an increase in the heat flow (about 10%) is observed.

In order to analyze this rather unexpected result, an explanation is sought in the temperature distribution along the insulation thickness at the four characteristic points that have already been defined in the previous study [1]. These are presented in Figure 7.

It is observed that for Tank *a* (Table 4, Figure 8), the temperature distribution along the insulation's thickness, at all four points selected is characteristically different from that of the other two types of tanks *b* and *c* (Table 4, Figures 9 and 10). More specifically, for Tank *a*, a section of the thickness of the insulating material where the temperatures are almost identical may be observed. For the remaining section, the PUR-32 material develops lower temperatures than the PUR-64 material. On the contrary, for the other two tanks, with

the exception of the ends of the thickness, where the temperatures are almost identical, throughout its thickness, the PUR-32 material develops higher temperatures than the PUR-64 one. In reality, the difference in heat flow recorded is not attributed to the temperature distribution throughout the insulating material's thickness, but rather to the slope of this distribution (thermal gradient), which, along with the thermal conductivity of the material, takes part into the calculation of the heat flow. The fact that this behavior is observed only for Tank *a* suggests that multidimensional heat-transfer effects associated with its geometric configuration play a significant role. It is important to emphasize that thermal problems, where significant temperature changes are recorded (from $-253\text{ }^{\circ}\text{C}$ (20 K) to $10\text{ }^{\circ}\text{C}$ (283 K) and non-linear, temperature-dependent materials are utilized, require detailed analysis of phenomena such as those employed in the present work. Further investigation using local heat-flux distributions and temperature-gradient analysis would be required to fully explain this behavior.

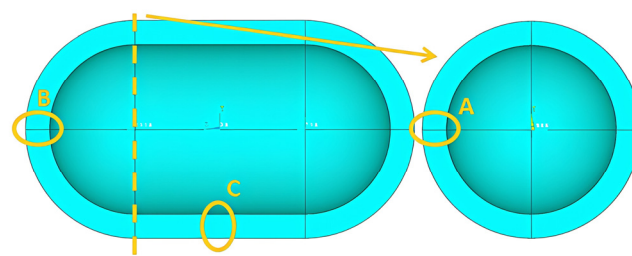


Figure 7. Cross-sections of the tank. Locations A, B, and C selected for evaluating the temperature profile across the thickness of the insulation layer [1].

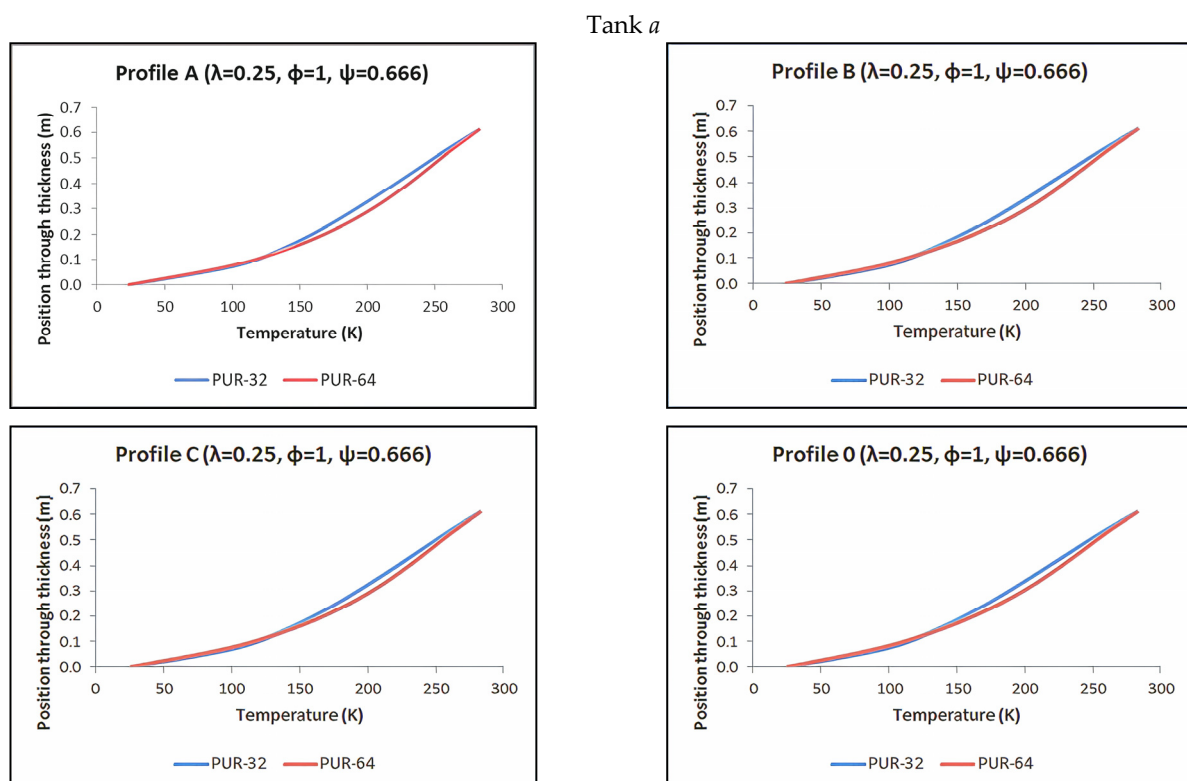
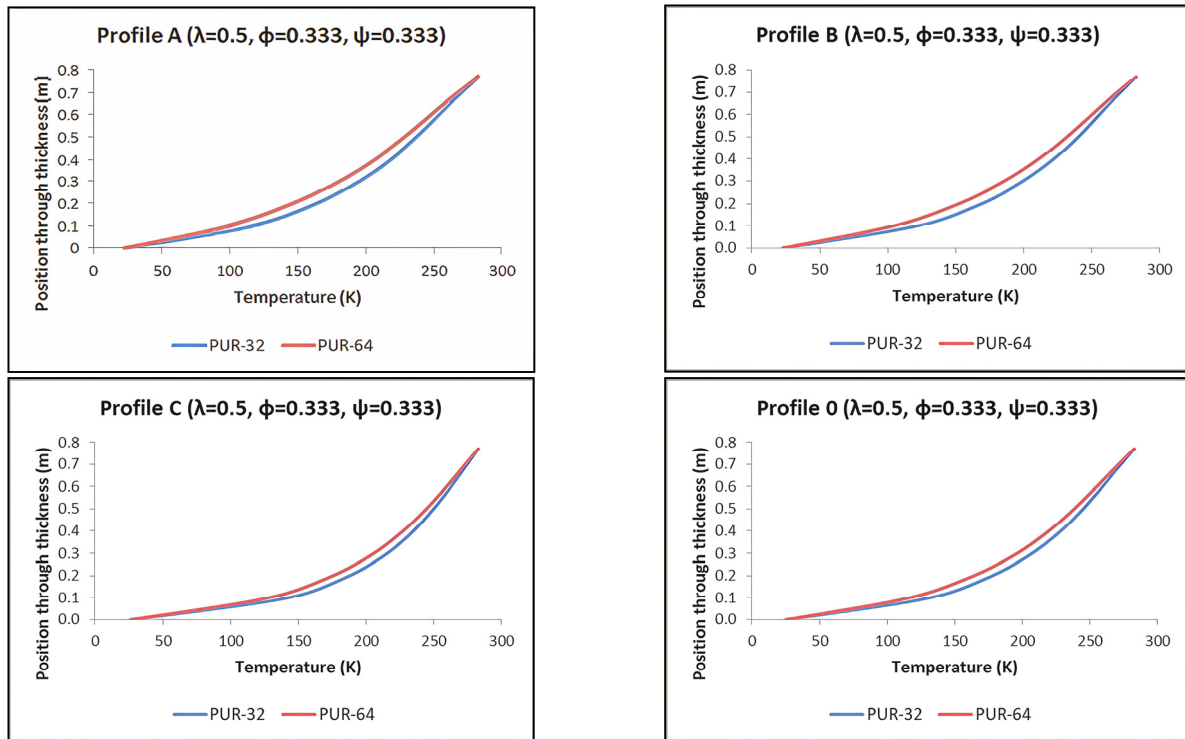
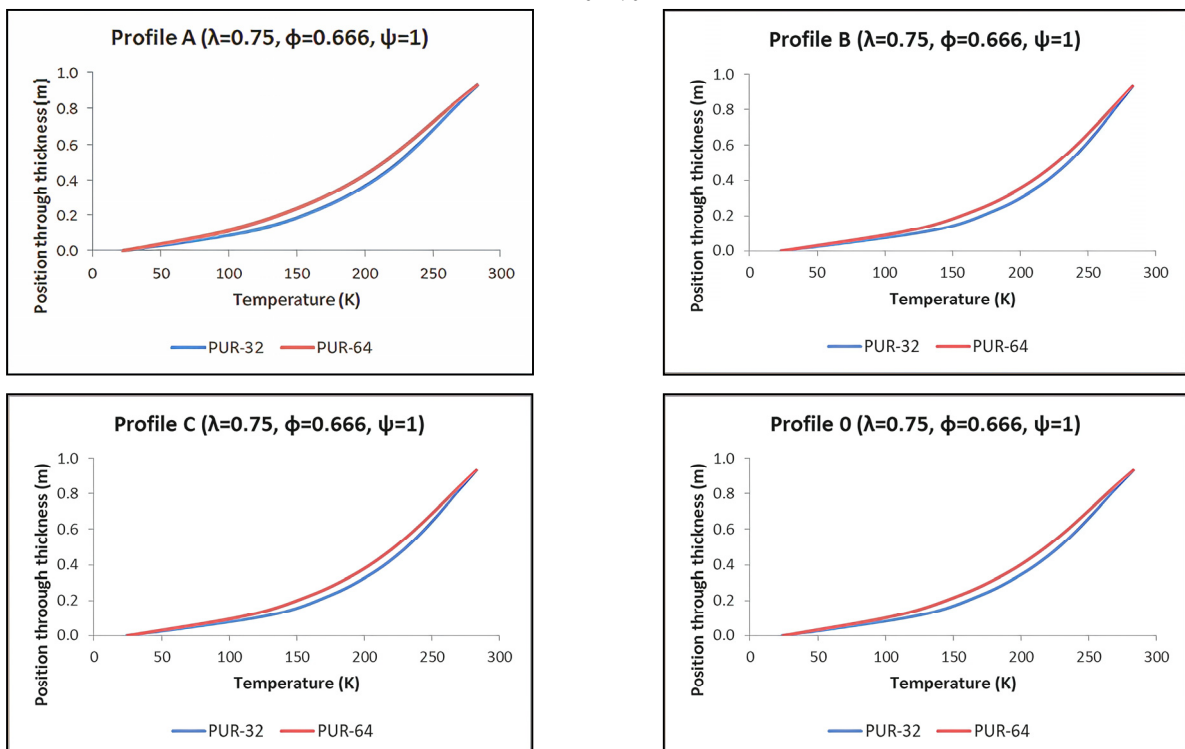


Figure 8. Temperature profile of selected areas of the insulation as a function of the material (Tank *a*).

Tank *b***Figure 9.** Temperature profile of selected areas of the insulation as a function of the material (Tank *b*).Tank *c***Figure 10.** Temperature profile of selected areas of the insulation as a function of the material (Tank *c*).

3.2. Change in the Volume of the Hydrogen Storage Chamber

In this subsection, the aim is to examine the effect of the volume of the hydrogen chamber on the thermal and structural behavior of the same three tanks (Table 4) for which the effect of changing the insulation material was examined in Section 3.1. The insulation

material used is PUR-64. Since the boil-off rate (BOR) is fixed at 0.1%/h and used as the reference criterion, any variation in the volume of stored hydrogen leads to a corresponding adjustment in the maximum permissible heat transfer between the surrounding environment and the hydrogen containment chamber, as defined by Equation (13).

The LH₂ fills up 93% of the inner tank. The remaining space is occupied by GH₂ [1]. Therefore, Equation (13) can be written as:

$$BOR = \frac{360000 \cdot Q}{\Delta h \cdot V_{LH_2} \cdot 0.93 \cdot \rho_{LH_2}} \left[\frac{\%}{hour} \right] \quad (14)$$

In order to calculate the maximum allowable heat flow for each tank case, Equation (15) can be derived from Equation (14):

$$Q_{max} = 8.45594 \cdot V_{LH_2} [W] \quad (15)$$

Three cases of hydrogen chamber volumes, 50 m³, 100 m³ and 150 m³, are studied, which compose the trio of cases for which conclusions will be drawn in this subsection. Based on Equation (15), Table 5 can be produced:

Table 5. Maximum allowable Heat Flow Q_{max} for each tank volume V_{LH_2} .

Tank Volume V_{LH_2} [m ³]	Maximum Heat Flow Q_{max} [W]
50	423
100	845
150	1268

Consequently, the following cases are examined:

(a) The overall surface area of the LH₂ storage chamber, determined by the tank dimensions lt , ls , a , b and c (Figure 11a).

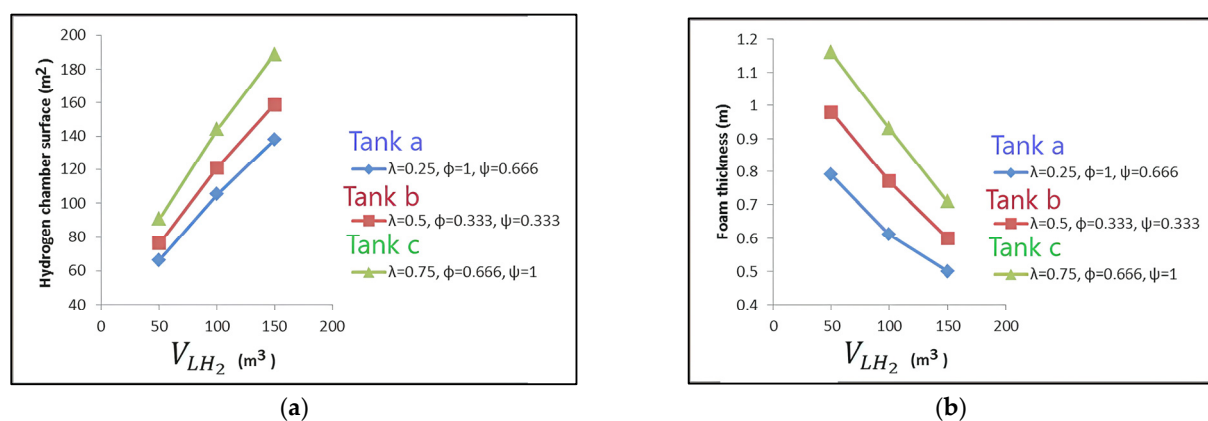


Figure 11. (a) Change in the surface area of the LH₂ storage chamber as a function of the change in the total volume of hydrogen V_{LH_2} . (b) Change in the required insulation thickness as a function of the change in the volume of hydrogen V_{LH_2} to maintain a BOR of 0.1%/h.

(b) The required insulation thickness necessary to achieve the target BOR of 0.1% (Figure 11b).

(c) The total insulation mass, evaluated from the tank geometry and the corresponding insulation volume (Figure 12a).

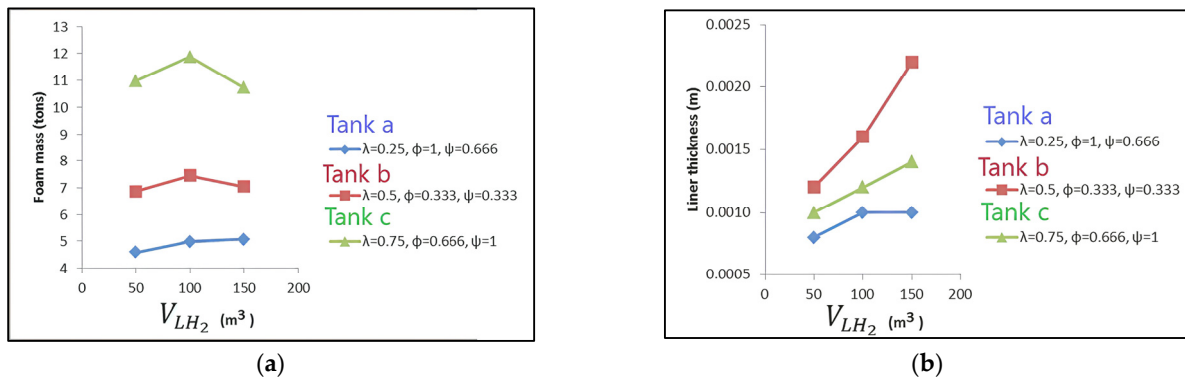


Figure 12. (a) Change in the mass of the insulation in relation to the total volume of hydrogen V_{LH_2} . (b) Change in the thickness of the aluminum shell in relation to the total volume of hydrogen V_{LH_2} to satisfy the Von Mises criterion.

(d) The minimum thickness of the aluminum liner needed to satisfy the Von Mises criterion and used for the structural assessment of the tank (Figure 12b)

(e) The overall mass of the aluminum storage shell (Figure 13a).

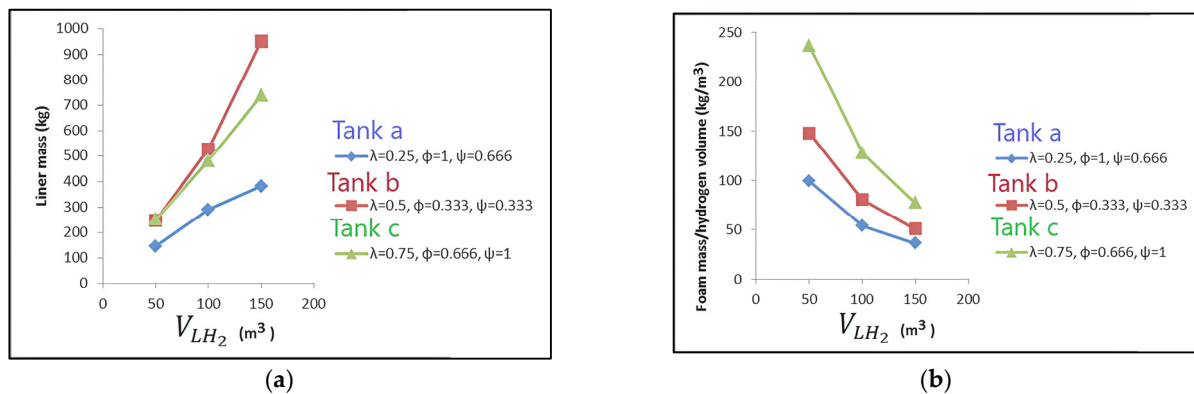


Figure 13. (a) Change in the mass of the aluminum shell as a function of the total volume of hydrogen V_{LH_2} . (b) Change in the mass of the insulation per unit of total volume of hydrogen V_{LH_2} .

(f) The ratio between the insulation mass and the total tank volume, providing a measure of insulation-related weight efficiency (Figure 13b).

(g) The gravimetric efficiency n_{grav} of the tank:

$$n_{grav} = \frac{m_{LH_2}}{m_{LH_2} + m_{tank}} = \frac{W_{LH_2}}{W_{LH_2} + W_{tank}} \quad (16)$$

which is the ratio of the total mass (or weight) of the stored LH_2 within the tank to the total mass (or weight) of the entire system [1] (Figure 14).

By studying the diagrams of the above figures, the following conclusions arise:

(a) For all three cases of tanks, increasing the volume of the hydrogen storage chamber increases the surface area required (Figure 11a).

(b) An increase in the volume of the hydrogen storage chamber, causes a reduction in the thickness of the insulation (Figure 11b), with this being a percentage of the order of 20% between the volume values of 50 m³, 100 m³ and 150 m³.

(c) For all three Tanks, the mass of the insulating material increases proportionally to the volume of the chamber as it rises from 50 m³ to 100 m³ (Figure 12a). However, when it is increased from 100 m³ to 150 m³:

i. For Tank a, the mass of the insulating material remains nearly constant.

ii. For Tank *b*, the mass of the insulating material decreases. At the maximum volume, the mass remains higher than the one corresponding to $50 \text{ m}^3 V_{\text{LH}_2}$.

iii. For Tank *c*, the mass of the insulating material decreases. At the maximum volume, the mass is lower than the one corresponding to 50 m^3 .

(d) Regarding the required thickness of the aluminum shell to meet the requirement to satisfy the Von Mises criterion (Figure 12b), three different behaviors are observed for the three different tanks:

i. For Tank *a*, an increase in the required thickness of the aluminum is observed, when increasing the volume of the hydrogen chamber from 50 m^3 to 100 m^3 . It remains unchanged when increasing the tank volume from 100 m^3 to 150 m^3 .

ii. For Tank *b*, an increase in the required aluminum thickness is observed that is proportional to the volume of the tank. The thickness has an increasing slope in the sections $50\text{--}100 \text{ m}^3$ and $100\text{--}150 \text{ m}^3$.

iii. For Tank *c*, an increase in the required aluminum thickness is observed, which appears to be linearly proportional to the volume of the tank.

(e) For the mass of insulating material per unit volume of the hydrogen chamber (Figure 13a), it is observed that for all three cases of tanks, an increase in the chamber volume implies a decrease in the mass of insulating material per unit volume. Indeed, as illustrated in the corresponding diagrams, the reduction observed for this particular parameter differs among the three tank configurations examined. However, as the storage volume increases, the values associated with the three tank types progressively converge. Consequently, it can be inferred that, as the amount of stored hydrogen increases, the variation in the required mass of insulating material becomes increasingly negligible. From a mass-efficiency perspective, this indicates that greater flexibility in selecting among the different tank geometries can be afforded.

(f) Finally, regarding the size of the barometric efficiency (gravimetric efficiency), a significant increase is observed as a function of the volume of hydrogen for all three cases of tanks (Figure 14). More specifically:

i. For Tank *a*, the barometric efficiency increased from 39% at 50 m^3 to 62.8% at 150 m^3 (58% increase).

ii. For Tank *b*, the barometric efficiency increased from 30.6% at 50 m^3 to 53.8% at 150 m^3 (75% increase).

iii. For Tank *c*, the barometric efficiency increased from 21.9% at 50 m^3 to 44.9% at 150 m^3 (104% increase).

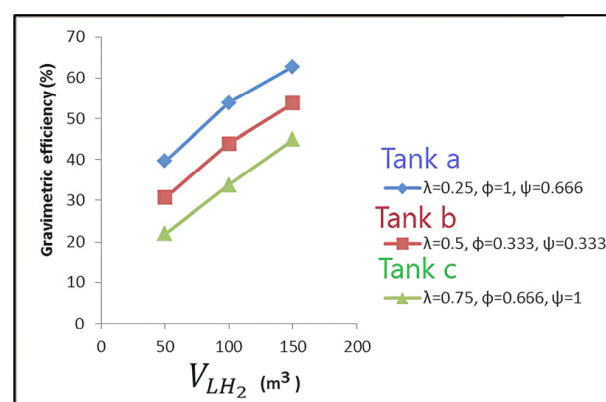


Figure 14. Variation of the gravimetric efficiency of a tank as a function of the total volume of hydrogen V_{LH_2} .

3.3. Change in the Selected Hydrogen Boil-Off Rate (LH_2 BOR)

Following Mital et al. [17], a hydrogen BOR of 0.1%/h was adopted as the reference value for all calculations performed both in the present and in the previous work [1]. This value served as the basis for evaluating the thermal behavior of the tanks and determining the required thickness of the insulating material.

In this subsection, however, two additional hydrogen boil-off rates (BORs) are considered in order to further investigate the variation in tank behavior under different thermal performance conditions. The objective is to assess whether the design of tanks with higher BORs is practically justifiable and to quantify the extent to which such an approach may influence the overall tank design characteristics.

Taking into consideration the findings of Mital et al. [17], hydrogen BORs considered acceptable for certain applications, such as space missions—where fuel utilization occurs over significantly shorter time periods compared to aviation applications—may reach values on the order of 1.5%/h.

For this reason, in addition to the reference BOR of 0.1%/h previously investigated, BORs of 0.5%/h and 1.0%/h are also investigated. The analysis of these additional BORs is conducted for the same three tank configurations investigated in the previous subsections, namely Tanks *a*, *b* and *c* (Table 4).

To draw conclusions, the following diagrams are presented:

(a) The insulation-layer thickness required to achieve target BORs of 0.1%/h, 0.5%/h, and 1%/h (Figure 15a);

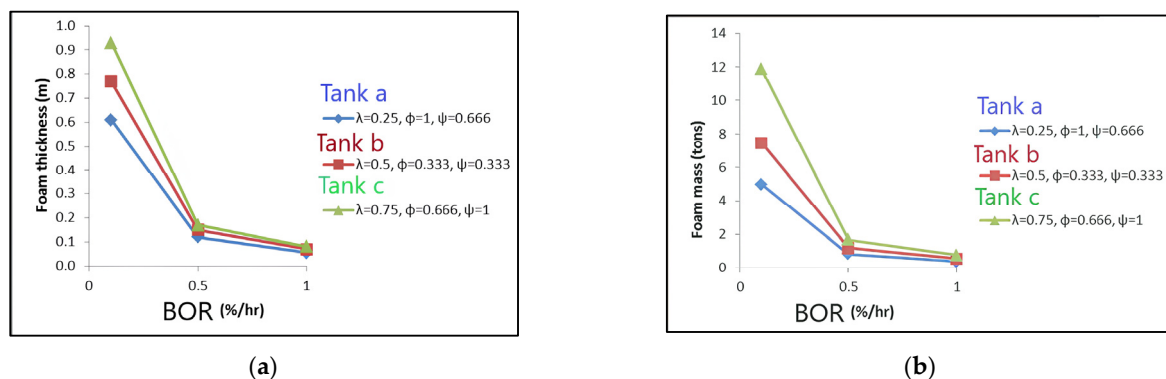


Figure 15. (a) Variation of the required insulation thickness as a function of the BOR. (b) Variation of the insulation mass as a function of the BOR.

(b) The corresponding total mass of insulation material, calculated from the tank geometry and insulation volume (Figure 15b);

(c) The minimum aluminum-shell thickness necessary to comply with the Von Mises stress criterion adopted for structural verification of the tank (Figure 16a);

(d) The resulting mass of the aluminum shell (Figure 16b);

(e) The contribution of each tank component to the overall tank mass, expressed as the percentage share of the aluminum liner, insulation layer, and composite outer shell (Figure 17a–c);

(f) The total structural mass of the tank assembly (Figure 18a);

(g) The gravimetric efficiency of the tank, as defined by Equation (4) (Figure 18b).

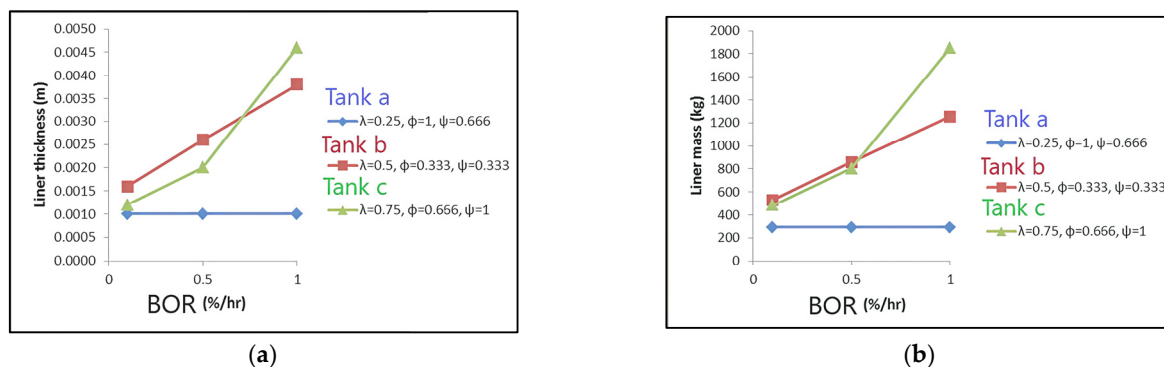


Figure 16. (a) Variation of the required aluminum shell thickness as a function of the BOR to satisfy the Von Mises criterion. (b) Variation of the mass of the aluminum shell as a function of the BOR.

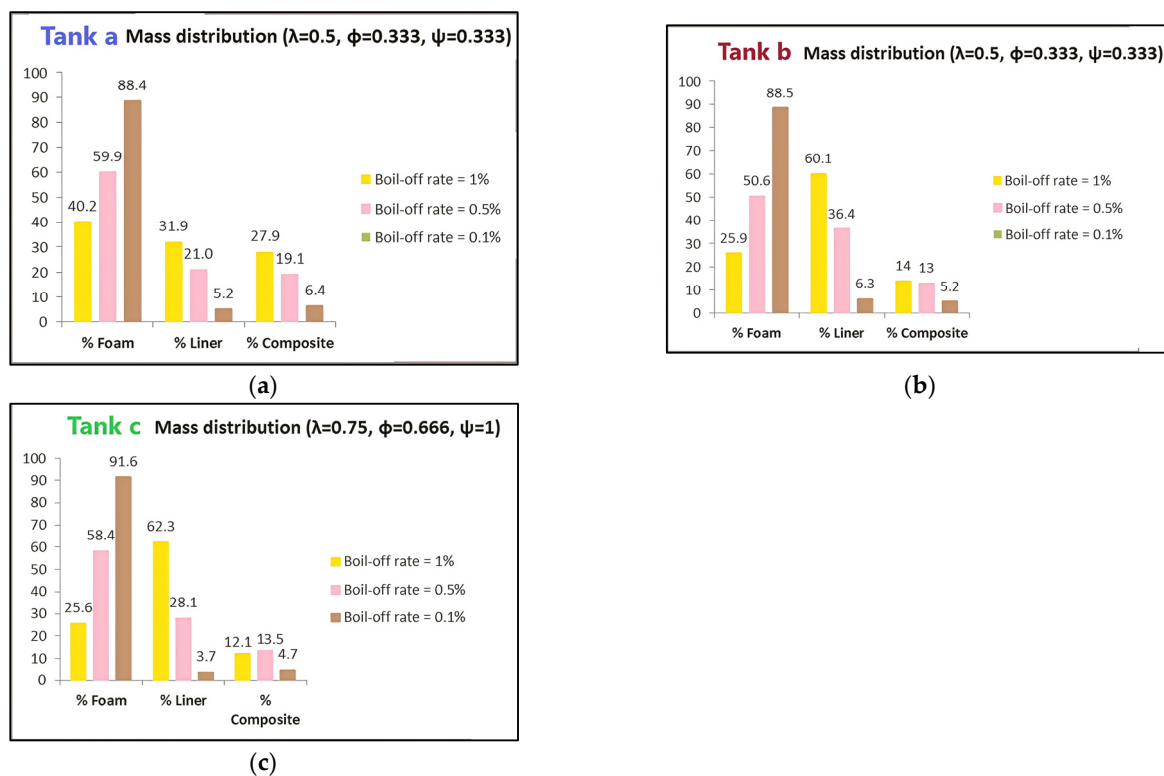


Figure 17. Percentage distribution of the masses of aluminum shell, insulation and composite shell as a function of the BOR for each Tank (a–c).

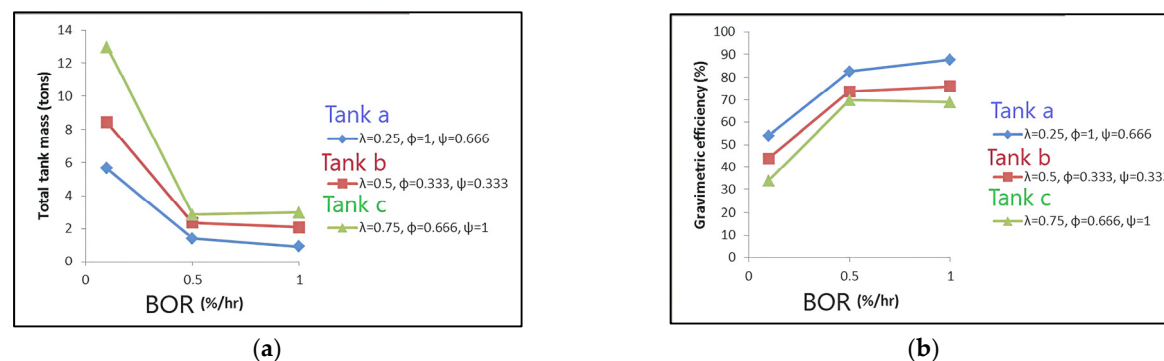


Figure 18. (a) Variation of the total mass of the tank as a function of the BOR. (b) Gravimetric efficiency as a function of the BOR.

Based on Figures 15–18, the following conclusions are drawn regarding the behavior of the tank under different selected hydrogen BOR (0.1%/h, 0.5%/h and 1%/h):

(a) For all three tank configurations examined, a significant reduction in both the required insulation thickness (Figure 15) and the corresponding insulation mass is observed as the selected hydrogen BOR increases from 0.1%/h to 0.5%/h and subsequently to 1.0%/h. More specifically, increasing the BOR from 0.1%/h to 0.5%/h results in an approximately 80% reduction in both the required insulation thickness and insulation mass for all three tank cases. A further increase in the BOR to 1.0%/h leads to an additional reduction of approximately 50% relative to the corresponding values obtained for the 0.5%/h case. Overall, when comparing the extreme cases of 0.1%/h and 1.0%/h BORs, the required insulation thickness and associated insulation mass decrease by approximately 90%.

(b) For the thickness of the aluminum shell (Figure 16), as determined by the developing Von Mises stress, three different behaviors were observed for the three tank cases analyzed, which are discussed in relation to Figures 19–21:

i. For Tank *a*, the required thickness of the aluminum shell maintains its minimum allowable thickness of 1 mm throughout all examined cases. As shown in Figure 18, changes in insulation thickness associated with different hydrogen boil-off rates have only a negligible effect on the Von Mises stress field developed within the aluminum layer. As a result, no further structural reinforcement is necessary, and the shell thickness remains unchanged.

ii. In contrast, Tank *b* exhibits an approximately linear increase in the required aluminum-shell thickness as the hydrogen boil-off rate rises. Figure 14 demonstrates a more intricate stress response, with the Von Mises stress distribution showing a pronounced sensitivity to the thickness of the insulation layer located between the aluminum liner and the external composite shell. Accordingly, modifications to the insulation thickness alter the stress state within the aluminum structure and directly affect the shell thickness required to satisfy the structural design criterion.

iii. For Tank *c*, an increasing slope of the required aluminum thickness is observed as the hydrogen BOR increases, which is due to the significant dependence of this tank shape on the thickness of the insulating material.

(c) From Figure 17, a clear trend is observed. At a hydrogen BOR of 0.1%/h, the insulating material dominates the total tank mass, accounting for approximately 90%. However, this dominance is significantly reduced at higher BORs. Specifically, its contribution decreases to approximately 50–60% for a rate of 0.5%/h and further to 25–40% for a rate of 1%/h.

(d) The total tank mass (Figure 18a) exhibits a pronounced decrease when transitioning from a BOR of 0.1%/h to 0.5%/h. In contrast, the subsequent increase from 0.5%/h to 1%/h produces no significant variation in total mass, which remains nearly constant. This suggests that the influence of the BOR on the total tank mass diminishes as the rate increases.

(e) The gravimetric efficiency of the tanks (Figure 18b), as a function of BOR, follows an inverse trend relative to the total mass, as expected. The transition from 0.1%/h to 0.5%/h is particularly critical in this regard. More specifically:

i. For Tank *a*, the gravimetric efficiency increases from 53.9% at 0.1%/h to 82.6% at 0.5%/h (an increase of 53%), followed by a further increase of 6.4%, reaching 87.8% at 1%/h.

ii. For Tank *b*, the efficiency rises from 43.9% at 0.1%/h to 73.7% at 0.5%/h (an increase of 68%), and then increases slightly by a further 3%, reaching 75.9% at 1%/h.

iii. For Tank *c*, the efficiency increases from 33.7% at 0.1%/h to 69.8% at 0.5%/h (an increase of 107%), followed by a slight decrease of 1.2%, resulting in a final efficiency of 68.9% at 1%/h.

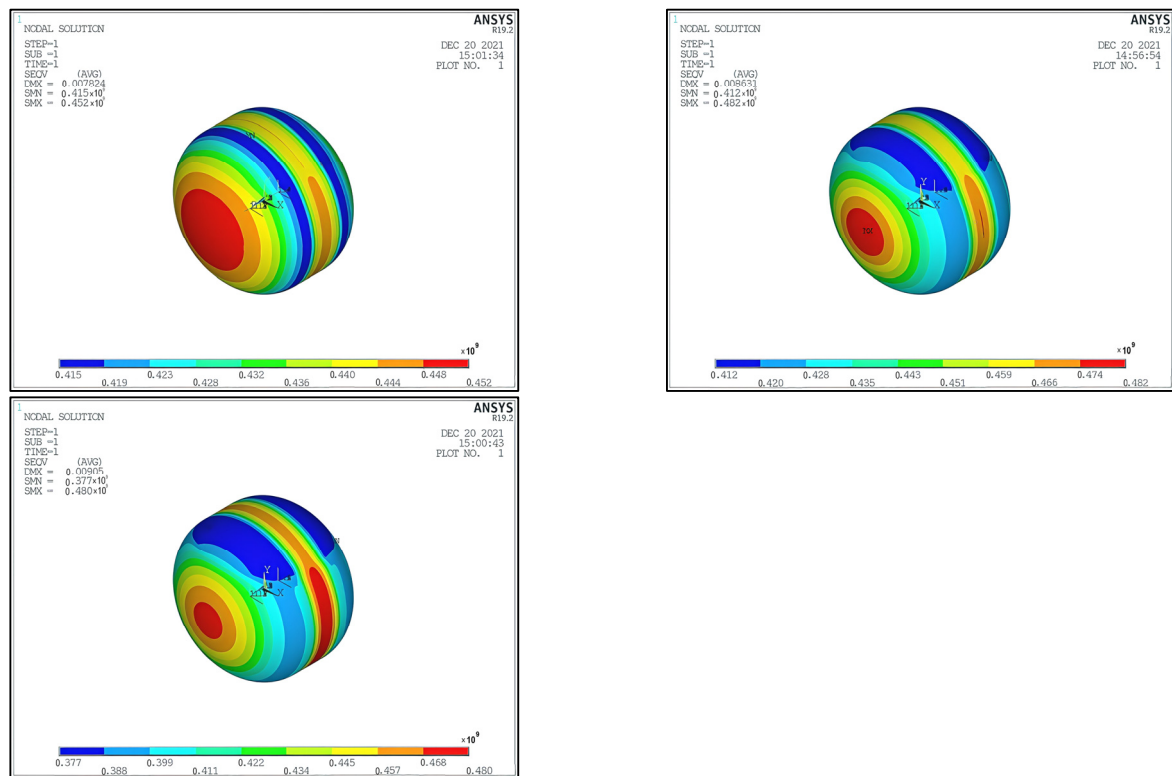


Figure 19. Von Mises stress distribution in the aluminum Tank a shell, for BOR (clockwise from top left) 0.1%/h, 0.5%/h and 1%/h.

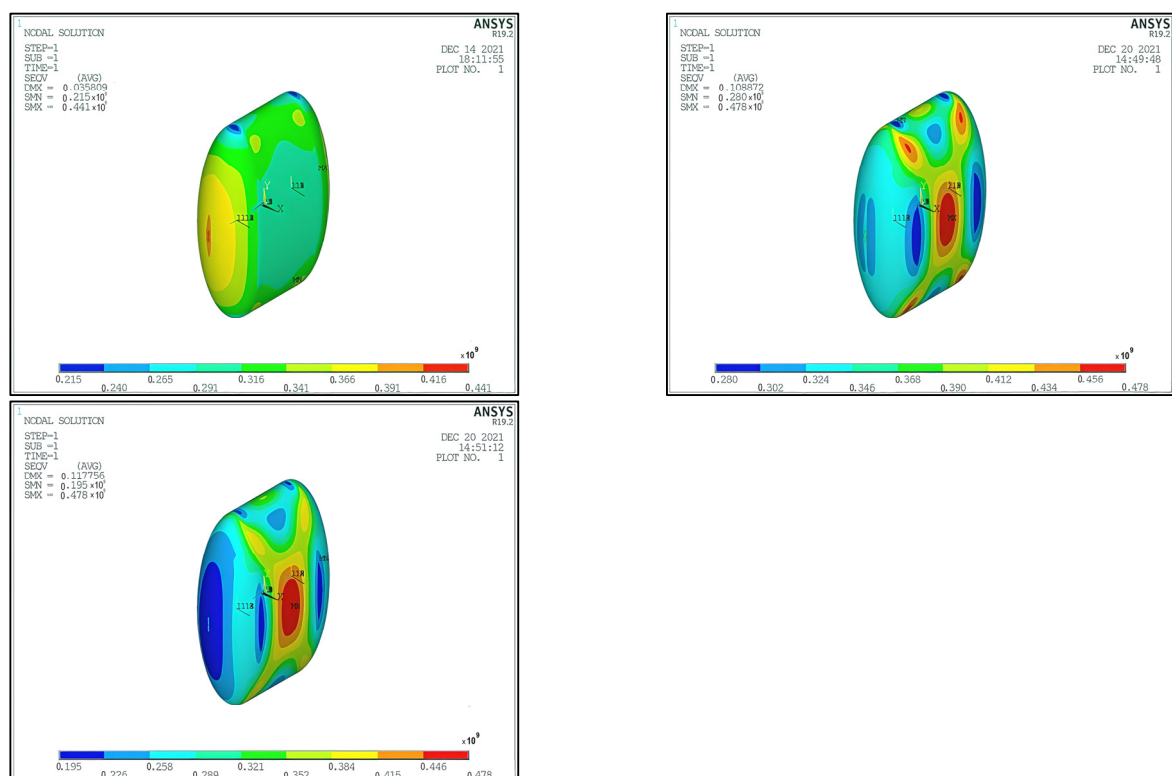


Figure 20. Von Mises stress distribution in the aluminum Tank b shell, for BOR (clockwise from top left) 0.1%/h, 0.5%/h and 1%/h.

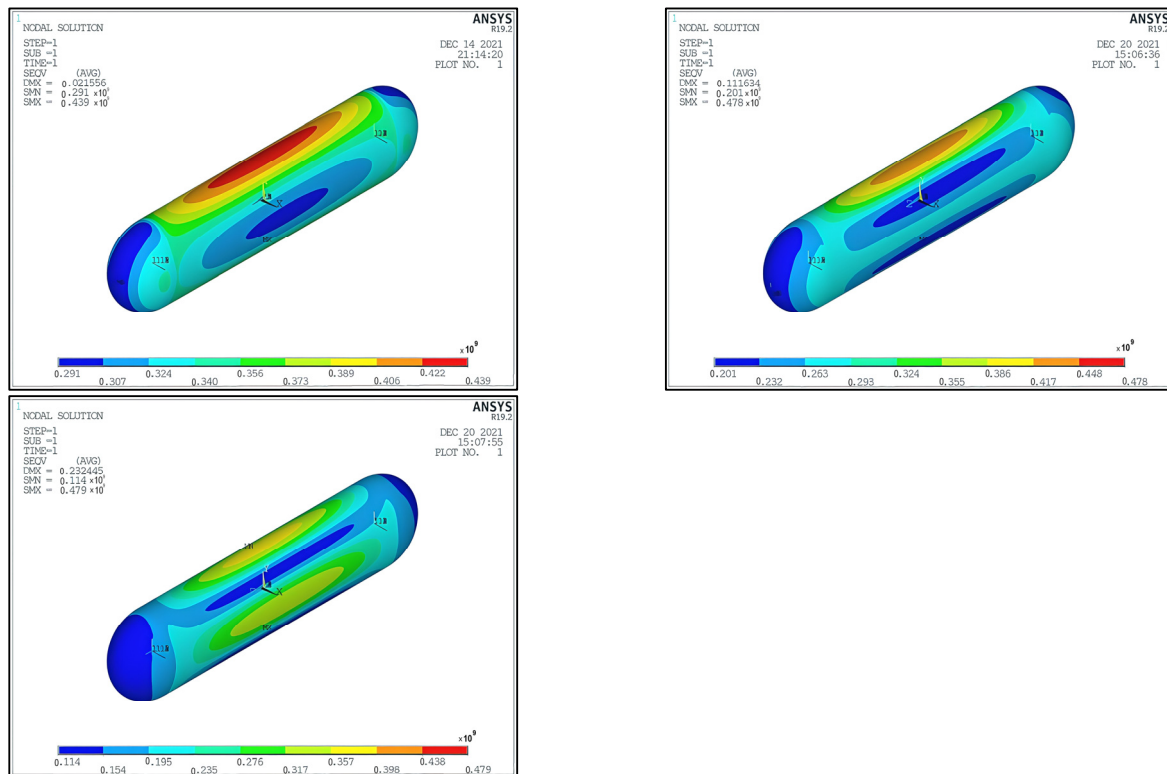


Figure 21. Von Mises stress distribution in the aluminum Tank c shell ($\lambda = 0.75$, $\varphi = 0.666$, $\psi = 1$), for BOR (clockwise from top left) 0.1%/h, 0.5%/h and 1%/h.

4. Discussion

Following the analysis on the results in Section 3, the main conclusions drawn from studying the three different Tank configurations are summarized as follows:

Section 3.1—Using a different insulating material: It is shown that the strong non-linearity of the thermal conductivity as a function of temperature of the selected insulating material (polyurethane (PUR) foam) leads to a *prima facie* paradox. In one of the three cases examined, the replacement of the insulating material with another material of the same type but with lower thermal conductivity does not lead to the expected improvement in thermal performance (i.e., a reduction in heat flow from the environment to the LH₂). Instead, the opposite effect is observed, namely an increase in heat flow. Geometry appears to influence the heat transfer. Differences in curvature may alter the temperature field and heat flow, resulting in behavior that cannot be explained by thermal conductivity alone. Although PUR-32 exhibits lower thermal conductivity than PUR-64 over part of the temperature range, the insulation is subjected to a temperature difference of approximately 260 K. Consequently, the local thermal conductivity varies continuously through the insulation thickness, modifying the temperature gradient and the effective thermal resistance. Since the temperature field also depends on the tank geometry and curvature, different configurations may exhibit different heat-transfer behavior. A more detailed investigation based on local heat-flux distributions and temperature-gradient analyses would be required to fully quantify these mechanisms and is proposed as future work.

These results indicate that simplified analytical approaches may not adequately capture the coupled thermal behavior when strongly temperature-dependent insulation properties are considered, highlighting the importance of detailed numerical analyses.

Section 3.2—Changing the volume of the hydrogen storage chamber: For a variable tank volume, the required insulation thickness decreases as the tank volume increases. This conclusion is significant from a design perspective, as it indicates that, for larger

hydrogen storage volumes, the relative impact of the insulation layer on the final tank volume becomes progressively smaller. Consequently, for a given hydrogen capacity, the use of a single large tank is generally more advantageous in terms of total construction volume than dividing the storage into multiple smaller tanks.

For example, when geometric constraints are imposed by integration into an existing aircraft design, a single large tank may be preferable to multiple smaller ones. On the other hand, in a clean-sheet aircraft design where hydrogen storage is incorporated from the outset, distributing the total hydrogen volume across multiple smaller tanks may be advantageous due to aerodynamic and integration considerations.

The analysis indicates that both the required aluminum-shell thickness and its associated mass are highly influenced by the geometric configuration of the hydrogen storage chamber. Configurations with very high or very low curvature are particularly sensitive, as they lead to localized concentrations of Von Mises stress and can significantly alter both the distribution and location of maximum stress. The insulation thickness also influences this behavior, due to the nonlinear temperature dependence of the material's elastic properties. As a result, even for identical geometric ratios (λ , ϕ , ψ), variations in tank volume lead to corresponding variations in the required aluminum shell thickness.

Section 3.3—Changing the selected hydrogen boil-off rate (BOR): The selected hydrogen BOR plays a key role in the overall behavior of the tanks due to its direct relation to heat transfer with the environment. More specifically, the BOR affects:

- i. Insulation thickness and mass: The prescribed BOR value establishes the maximum heat transfer rate permitted between the ambient surroundings and the storage tank. As expected, increasing the BOR from 0.1%/h to 0.5%/h and 1%/h leads to a rapid reduction in insulation thickness. However, a strong convergence of values is observed between 0.5%/h and 1%/h, indicating that at higher BOR levels the sensitivity of insulation design to geometric variations decreases.

- ii. Aluminum shell thickness and mass: In contrast, the aluminum shell exhibits an inverse trend. As the BOR increases, the differences between configurations become more pronounced. This is attributed to the interaction between geometry-dependent thermal fields and the resulting redistribution of Von Mises stresses governing structural sizing.

Despite these opposing trends, the total mass of the tanks decreases significantly with increasing BOR and tends to converge at higher values. This behavior is particularly important when linked to aircraft mission requirements. Long-range or endurance aircraft, which operate over extended mission durations, require low BORs to minimize hydrogen losses over time. This constraint limits feasible tank geometries (λ , ϕ , ψ) and reduces design flexibility. Conversely, short- and medium-range aircraft, with shorter mission durations and lower cumulative losses, allow for higher BORs and therefore greater flexibility in tank design, enabling more efficient integration with aerodynamic and structural constraints. It should be noted that increasing the BOR also increases hydrogen loss, venting demand, pressure-control requirements, and raises safety concerns. As such, for aircraft use, BOR should not be raised simply to improve gravimetric efficiency.

5. Conclusions

The pressing need to decarbonize the aviation sector has identified hydrogen as a primary candidate for zero-emission propulsion. LH₂ storage within the fuselage is widely recognized as a critical enabling technology for large commercial operations, as conventional wing storage is volumetrically unfeasible [26]. Comprehensive surveys confirm that passively insulated cryogenic composite sandwich pressure vessels represent the most realistic and promising technology for near- to mid-term aircraft integration [26].

This layered architecture eliminates the need for heavy, complex active vacuum subsystems, thereby maximizing system simplicity and reliability.

The gravimetric efficiency and thermal trends established in this work are highly compatible with these recent baseline investigations in the literature. For instance, Bagarello et al. [25] reported a baseline gravimetric efficiency of 36.7% for an idealized 100 m³ tank volume with a 0.5 m PUR insulation thickness. This closely aligns with our reference metrics for a 100 m³ tank operating at a reference BOR of 0.1%/h, where gravimetric efficiencies spanned from 33.7% to 53.9% depending on tank geometry. Furthermore, both studies confirm that scaling up the chamber volume (e.g., from 50 m³ to 150 m³) significantly enhances mass efficiency, showing up to a 104% increase in gravimetric metrics. Our finding that increasing the selected hydrogen BOR from 0.1%/h to 1.0%/h yields an approximate 90% reduction in insulation thickness and mass is strongly supported by these thermal sensitivities.

The thermal and mass distribution trends observed in this work are also highly compatible with the systems-level findings of Karaikos et al. [27]. Notably, the vaporization and BOR formulations implemented in their distribution model are directly derived from and validated by our underlying mathematical equations.

Overall, the results demonstrate a strong and non-trivial coupling between thermal behavior, structural response, and system-level design constraints in LH₂ storage tanks. The study shows that nonlinear material properties can lead to counterintuitive thermal performance effects, that tank volume strongly influences both insulation and structural requirements, and that geometry plays a critical role in governing stress distribution and structural sizing. In addition, the selected BOR emerges as a key system-level parameter that simultaneously affects insulation demand, structural mass distribution, and overall design flexibility.

Indeed, the benefits of modifying the parameters of the LH₂ chamber must be balanced against increased hydrogen losses, greater venting requirements, higher demands on pressure control, and additional safety considerations. Therefore, a higher BOR or LH₂ volume should not be regarded as universally advantageous. It should be noted, also, that challenges regarding the very nature of hydrogen were not investigated in this study. Those include its circulation and stratification, as well as modelling two-phase flow within the tank.

All these effects lead to complex trade-offs between thermal efficiency, structural integrity, and system integration constraints. As a result, LH₂ tank design cannot be treated as a purely thermal or structural optimization problem, but must instead be addressed as a mission-driven design problem. The optimal configuration is therefore not universal, but depends strongly on the intended aircraft mission profile, where long-range applications favor thermally constrained, low-loss configurations with reduced geometric flexibility, while short- and medium-range applications allow for more diverse and integrated design solutions. Future work could include transient thermal analyses under representative operating conditions, as well comparison with experimental data, such as temperature distributions or structural stresses. Other topics of interest include the study of variable fill levels, multi-objective optimization, pressure evolution, hydrogen stratification, and worst-case operating scenarios, using this methodology as the basis.

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Abbreviations

The following abbreviations are used in this manuscript:

H ₂	Hydrogen
LH ₂	Liquid Hydrogen
CO ₂	Carbon Dioxide
GH ₂	Gaseous Hydrogen
FEM	Finite Element Method
PUR	Polyurethane (Foam)
PUR-64	Polyurethane (Foam) with a density of 64 kg/m ³
PUR-32	Polyurethane (Foam) with a density of 32 kg/m ³
TRL	Technology Ready Level
DOF	Degree of Freedom
APDL©	ANSYS Parametric Design Language©
CFRP	Composite Fiber Reinforced Polymer
BOR	Boil-Off Rate
BOG	Boil-Off Gas
CTE	Coefficient of Thermal Expansion

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