

Article

Boil-Off Gas Generation Characteristics of Liquid Hydrogen in a Type-C Tank Under Sloshing Conditions: Effects of Filling Ratio, Excitation Parameters, and Baffles

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Abstract

Liquid hydrogen (LH₂) is a promising energy carrier for maritime transportation owing to its high storage density and carbon-free utilization. However, its cryogenic storage temperature inevitably causes boil-off gas (BOG) generation, even with advanced insulation, and ship-induced sloshing can further disturb interfacial heat and mass transfer. This study numerically investigates BOG generation in a Type-C LH₂ tank under sloshing conditions. A transient two-phase CFD model was developed using the volume of fluid method to track the liquid–vapor interface and the Lee phase-change model to calculate evaporation and condensation. Sloshing excitation was imposed through mesh motion, and parametric simulations were performed for different filling ratios, frequencies, amplitudes, and baffle configurations. The results show that the 50% filling condition produced the lowest boil-off rate, while amplitude had a stronger influence than frequency by increasing free-surface deformation, internal velocity, and convective heat transfer. Baffles reduced liquid motion but could increase BOG through additional conductive heat-transfer paths. These findings support BOG prediction and baffle design for Type-C LH₂ tanks.

Keywords: liquid hydrogen; type-C tank; boil-off gas; sloshing; lee phase-change model; baffle

1. Introduction

The transition toward carbon-neutral energy systems has accelerated the development of alternative fuels and corresponding storage and transportation technologies. In the maritime sector, increasingly stringent environmental requirements have encouraged the adoption of low- and zero-carbon energy carriers that can replace conventional fossil fuels in long-distance transportation. Among the available candidates, hydrogen has been widely regarded as a promising energy carrier because it can be produced from renewable energy sources and does not emit carbon dioxide during use [1]. In particular, liquefied hydrogen (LH₂) has received increasing attention for large-scale transportation because liquefaction can significantly increase the volumetric storage density compared with compressed gaseous hydrogen [1,2]. Therefore, the development of reliable LH₂ storage tanks and cargo containment systems is becoming an important technical issue for future hydrogen supply chains.



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Despite these advantages, the storage and transportation of LH2 involve considerable thermal and safety challenges. Hydrogen must be stored at approximately 20 K, which produces a large temperature difference between the cryogenic fluid and the surrounding environment [1,3]. Even when high-performance insulation systems are applied, heat ingress into the tank cannot be completely avoided. The heat transferred through the tank wall and insulation system causes partial evaporation of the liquid phase and generates boil-off gas (BOG). Excessive BOG generation may increase the internal tank pressure, reduce storage efficiency, and impose additional operational burdens, such as reliquefaction, fuel utilization, or venting [3]. Accordingly, accurate prediction of BOG generation under various operating and motion conditions is essential for the safe and efficient design of LH2 storage and transportation systems.

The thermo-fluid behavior inside an LH2 tank is governed by complex interactions among heat ingress, two-phase flow, interfacial heat and mass transfer, and phase change. In a partially filled cryogenic tank, liquid hydrogen and vapor hydrogen coexist, and the temperature field may become non-uniform owing to natural convection, thermal stratification, and interfacial transport. Liu et al. [4] numerically investigated thermal stratification in a non-isothermal LH2 tank under sloshing excitation and showed that the temperature near the liquid–vapor interface and the tank bottom could become higher than that in the central liquid region. These results indicate that the internal thermal response of an LH2 tank cannot be evaluated solely from the total amount of external heat ingress, because local flow structures, phase-change behavior, and interfacial heat transfer strongly affect pressure and evaporation characteristics.

In practical transportation environments, LH2 tanks are not subjected only to static thermal conditions. Ship motions, vehicle acceleration, braking, turning, and wave-induced disturbances can generate sloshing of the liquid phase inside the tank. Sloshing changes the shape and location of the free surface, increases the interaction between the liquid and vapor phases, and enhances convective heat transfer near the tank wall. Qiu et al. [5] analyzed the effect of filling level on the hydrodynamic characteristics of sloshing under the first natural frequency and reported that the pressure response varied significantly with the filling ratio. Zhu et al. [6] studied the thermodynamic coupling characteristics of LH2 sloshing in a horizontal tank for heavy-duty trucks and confirmed that large-amplitude sloshing strengthens convective heat transfer, increases the liquid temperature, and decreases the vapor temperature. Li et al. [7] also examined the thermal dynamic behavior of LH2 in a trailer storage tank using a numerical model and reported that sloshing can increase both the evaporation rate and the tank pressure while weakening thermal stratification. Kartuzova and Kassemi [8] validated a CFD model for a small-scale tank subjected to high-level lateral acceleration and showed that lateral sloshing can significantly influence heat transfer and pressure variation.

The effect of sloshing on BOG generation has also been investigated in cryogenic tanks for liquefied natural gas (LNG), which provides useful reference data for LH2 tank analysis. Wu and Ju [9] developed a numerical model considering phase change and external heat leakage to study BOG generation in a Type-C independent LNG tank under sinusoidal sloshing excitation. Their results showed that sloshing excitation significantly affects the internal temperature distribution, vapor–liquid interface motion, and BOG generation characteristics. Grotle and Æsøy [10,11] analyzed the thermal response of marine LNG fuel tanks under sloshing-induced mixing and demonstrated that liquid motion can strongly modify the thermodynamic state of cryogenic tanks. Jeon et al. [12] experimentally and numerically investigated the influence of sloshing motion on BOG and boil-off rate (BOR) in a cryogenic liquid tank and showed that the evaporation behavior can vary depending on the motion condition. However, LNG-based results cannot be directly

extended to LH2 tanks because LH2 has a much lower boiling temperature, lower liquid density, and different liquid-to-vapor density ratio. Beyond maritime transportation, liquid hydrogen is also attracting increasing attention in hydrogen-powered aviation because of its high gravimetric energy density and potential for low-emission propulsion. In hydrogen aeroengines, cryogenic fuel storage, waste-heat recovery, and thermal management are closely coupled, and recent studies have investigated closed Brayton-cycle systems for cooling and waste-heat recovery. Therefore, accurate prediction of LH2 thermal behavior and BOG generation is important not only for marine tanks but also for broader hydrogen-powered transportation systems.

In addition to thermal-fluid responses, internal structures used for sloshing suppression are important design factors in cryogenic tanks. Baffles are commonly installed to reduce free-surface deformation, dissipate liquid kinetic energy, and mitigate excessive impact loads on tank walls. Duan et al. [13] numerically investigated fluid–structure interaction in a liquid-filled tank with baffles and showed that internal baffles can improve flow stability, although large loads may be concentrated near the baffle region. Xue et al. [14] experimentally examined vertical baffles with different configurations and demonstrated their effectiveness in reducing sloshing pressure. Kang and Liu [15] analyzed the influence of baffle position during braking and turning of a tank truck and reported that the baffle arrangement strongly affects liquid motion and hydrodynamic force. For LH2 tanks, Liu and Li [16] further showed that baffles can influence pressure, temperature, velocity distribution, and phase-change behavior near the vapor–liquid interface.

Although baffles are effective in suppressing liquid motion, their function in LH2 tanks should be evaluated from both hydrodynamic and thermal perspectives. From a hydrodynamic viewpoint, baffles reduce free-surface deformation and suppress violent sloshing. From a thermal viewpoint, however, they may provide additional solid conduction paths and increase heat transfer into the cryogenic fluid. Therefore, a baffle arrangement that is effective in reducing liquid motion may still increase BOG generation if the additional thermal conduction becomes dominant. This trade-off between sloshing suppression and thermal penalty has not yet been sufficiently clarified for Type-C LH2 tanks under sloshing conditions.

Several recent studies have emphasized that the numerical analysis of LH2 tanks is a practical and cost-effective approach, because direct experiments using liquid hydrogen require strict safety measures and high operating costs [2,17]. Nevertheless, previous studies have mainly focused on either hydrodynamic loads, pressure variation, or thermal response under limited operating conditions. In particular, systematic investigation of BOG generation in a Type-C LH2 tank under combined variations in filling ratio, sloshing frequency, sloshing amplitude, baffle installation, and the number of baffles remains insufficient. This limitation is critical because the evaporation characteristics of LH2 tanks can vary significantly depending on the intensity of sloshing and the heat-transfer paths formed by internal structures.

In this study, the BOG generation characteristics of liquid hydrogen inside a Type-C tank under sloshing conditions were numerically investigated. A two-phase CFD model was established using the volume of fluid (VOF) method to track the liquid–vapor interface, and the Lee phase-change model was applied to account for evaporation and condensation. The sloshing motion was implemented using a mesh motion approach, and the thermophysical properties of hydrogen were defined as functions of temperature and pressure. The numerical model was validated through grid independence verification and comparison with available reference data before conducting the parametric analysis.

The main objective of this study is to clarify the influence of sloshing-related parameters on the thermo-fluid behavior and BOG generation of a Type-C LH2 tank. The effects

of filling ratio, sloshing frequency, and sloshing amplitude were examined by comparing the free-surface motion, temperature distribution, velocity field, vapor mass variation, and BOR. In addition, the effects of baffle installation and the number of baffles were analyzed to evaluate the balance between hydrodynamic suppression and additional heat conduction. The results of this study provide useful numerical data for BOG prediction, baffle arrangement optimization, and thermal safety assessment of Type-C LH2 tanks under maritime transportation conditions.

2. Numerical Model

2.1. Lv1 Vulnerability Criterion

In this study, a transient two-phase computational fluid dynamics (CFD) model was established to investigate the thermo-fluid behavior and BOG generation characteristics of liquid hydrogen in a Type-C tank under sloshing excitation. The numerical analysis was conducted using ANSYS Fluent R22, and the governing equations were discretized based on the finite volume method (FVM). In the FVM formulation, the computational domain is divided into a finite number of control volumes, and the conservation laws are integrated over each control volume. This approach ensures local conservation of mass, momentum, and energy, which is essential for predicting the coupled thermal and hydrodynamic behavior of cryogenic two-phase flow.

The liquid and vapor phases of hydrogen were treated as immiscible phases separated by a deformable free surface. To capture the transient motion of the liquid–vapor interface, the VOF method was adopted. The phase-change process between liquid hydrogen and hydrogen vapor was modeled using the Lee model, in which evaporation and condensation are determined from the local temperature difference relative to the saturation temperature. The tank motion associated with sloshing excitation was implemented using a mesh motion technique. The thermophysical properties of hydrogen were defined as functions of local temperature and pressure based on reference data [18], and the numerical formulations were implemented according to the CFD solver theory [19]. Turbulence was modeled using the standard k - ϵ turbulence model for sloshing simulations. This model was adopted to account for turbulent mixing and enhanced momentum transport induced by the high internal velocities generated during sloshing. The same turbulence model was applied consistently to all parametric cases.

2.2. Governing Equations

The flow inside the Type-C liquid hydrogen tank was assumed to be transient, incompressible for the liquid phase, and compressible for the vapor phase through the density variation associated with local thermodynamic conditions. A single set of conservation equations was solved over the entire computational domain using the shared-field formulation of the VOF model. The continuity equation is expressed as Equation (1).

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \vec{u}) = 0 \quad (1)$$

where ρ is the mixture density and $\vec{u}$ is the velocity vector.

The momentum conservation equation is written as Equation (2).

$$\frac{\partial (\rho \vec{u})}{\partial t} + \nabla \cdot (\rho \vec{u} \vec{u}) = -\nabla p + \rho \vec{g} + \nabla \cdot \left[\mu \left(\nabla \vec{u} + \nabla \vec{u}^T \right) \right] + \vec{F}_\sigma \quad (2)$$

where p is the pressure, $\vec{g}$ is the gravitational acceleration, μ is the dynamic viscosity, and $\vec{F}_\sigma$ represents the surface tension force acting near the liquid–vapor interface. The surface

tension effect was included because the interface deformation and local curvature can affect the phase distribution during sloshing.

The energy conservation equation is given by Equation (3).

$$\frac{\partial(\rho E)}{\partial t} + \nabla \cdot [\vec{u}(\rho E + p)] = \nabla \cdot (k_{eff} \nabla T) + \dot{m}_{pq} h_{pq} \quad (3)$$

where E is the total energy, k_{eff} is the effective thermal conductivity, T is the temperature, $\dot{m}_{pq}$ is the interphase mass transfer rate, and h_{pq} is the latent heat associated with phase change. The source term on the right-hand side accounts for the energy exchange caused by evaporation and condensation.

In the VOF framework, the mixture properties are calculated from the volume-fraction-weighted contributions of the liquid and vapor phases. For a general property (ϕ), the mixture value can be expressed as Equation (4).

$$\phi = \alpha_l \phi_l + \alpha_v \phi_v \quad (4)$$

where α_l and α_v are the volume fractions of liquid hydrogen and hydrogen vapor, respectively. The subscripts l and v denote the liquid and vapor phases.

2.3. Volume of Fluid Model

The VOF method was employed to track the liquid–vapor interface during sloshing. This method is suitable for problems in which two immiscible fluids share a clearly defined interface but experience large interface deformation. In each computational cell, the phase distribution is represented by the volume fraction. A cell with $\alpha_l = 1$ is fully occupied by liquid hydrogen, a cell with $\alpha_l = 0$ is occupied by hydrogen vapor, and a cell with $0 < \alpha_l < 1$ contains the liquid–vapor interface.

The volume fraction equation for the liquid phase is expressed as Equation (5).

$$\frac{1}{\rho_l} \left[\frac{\partial(\alpha_l \rho_l)}{\partial t} \nabla \cdot (\alpha_l \rho_l \vec{u}_l) \right] = S_{\alpha_l} + \frac{\dot{m}_{vl} - \dot{m}_{lv}}{\rho_l} \quad (5)$$

where ρ_l is the liquid density, $\vec{u}_l$ is the liquid-phase velocity, S_{α_l} is the volume-fraction source term, $\dot{m}_{lv}$ is the evaporation mass transfer rate from liquid to vapor, and $\dot{m}_{vl}$ is the condensation mass transfer rate from vapor to liquid.

The liquid and vapor volume fractions satisfy the following constraint:

$$\alpha_l + \alpha_v = 1 \quad (6)$$

Because the VOF model solves a single momentum equation for the mixture, the interface motion is determined by the local velocity field and the phase-fraction transport equation. This formulation enables simultaneous prediction of free-surface deformation, interfacial heat transfer, and phase redistribution during sloshing excitation.

2.4. Phase-Change Model

The Lee phase-change model was applied to simulate the evaporation and condensation of hydrogen. In this model, phase change is determined by comparing the local phase temperature with the saturation temperature. When the liquid temperature exceeds the saturation temperature, evaporation occurs. Conversely, when the vapor temperature is lower than the saturation temperature, condensation occurs. This formulation is widely used in cryogenic tank simulations because it can be coupled with the VOF method and applied to transient two-phase flow with moving interfaces.

For evaporation from the liquid phase to the vapor phase, the mass transfer rate is defined as Equation (7).

$$\dot{m}_{lv} = \lambda_l \alpha_l \rho_l + \frac{T_l - T_{sat}}{T_{sat}}, T_l \geq T_{sat} \quad (7)$$

where λ_l is the evaporation coefficient, T_l is the liquid temperature, and T_{sat} is the saturation temperature. For condensation from the vapor phase to the liquid phase, the mass transfer rate is expressed as Equation (8).

$$\dot{m}_{vl} = \lambda_v \alpha_v \rho_v + \frac{T_{sat} - T_v}{T_{sat}}, T_v < T_{sat} \quad (8)$$

where λ_v is the condensation coefficient, ρ_v is the vapor density, and T_v is the vapor temperature. The phase-change source term in the energy equation is calculated from the interphase mass transfer rate and the latent heat of vaporization:

$$S_E = \dot{m}_{pq} h_{fg} \quad (9)$$

where h_{fg} is the latent heat of vaporization. Because the saturation temperature and latent heat of hydrogen vary with pressure and temperature, these properties were updated during the calculation using temperature- and pressure-dependent correlations based on thermophysical property data [18]. This treatment allows the numerical model to reflect local phase-change behavior under transient pressure and temperature variations.

In the present sign convention, evaporation represents mass transfer from liquid hydrogen to hydrogen vapor, whereas condensation represents mass transfer from hydrogen vapor to liquid hydrogen. The associated energy source term was treated according to the latent heat absorbed during evaporation and released during condensation. A sensitivity test was also conducted for the representative case by varying the Lee coefficients. The sensitivity results confirmed that the coefficient values affected the absolute vapor-generation amount to some extent but did not change the main BOG-generation trends observed in this study.

2.5. Mesh Motion Model for Sloshing Excitation

To reproduce the sloshing motion of liquid hydrogen inside the Type-C tank, the mesh motion model was coupled with the VOF formulation. In this approach, the computational domain moves according to the prescribed tank motion, while the governing equations are solved in the moving control volume. The conservation equation for a general scalar variable ϕ in a moving control volume can be written as Equation (10).

$$\frac{d}{dt} \int_V \rho \phi dV + \int_{\partial V} \rho \phi dV (\vec{u} - \vec{u}_g) \cdot \vec{n} dA = \int_{\partial V} \Gamma_\phi \nabla \phi \cdot \vec{n} dA + \int_V S_\phi dV \quad (10)$$

where V is the control volume, ∂V is the control-volume boundary, $\vec{u}_g$ is the mesh velocity, $\vec{n}$ is the outward unit normal vector, Γ_ϕ is the diffusion coefficient, and S_ϕ is the source term for ϕ .

The mesh motion formulation satisfies the geometric conservation requirement by accounting for the change in control-volume size during transient tank motion. The updated control-volume size at the next time step is calculated as Equation (11).

$$V^{n+1} = V^n + \frac{dV}{dt} \Delta t \quad (11)$$

where V^n and V^{n+1} are the control-volume sizes at the current and next time levels, respectively, and Δ is the time step.

The volume change rate is obtained from the mesh velocity at the control-volume boundary:

$$\frac{d}{dt} = \int_{\partial V} \vec{u}_g \cdot \vec{n} dA \quad (12)$$

The prescribed sloshing excitation was imposed as a sinusoidal tank motion. The translational displacement of the tank can be expressed as Equation (13).

$$x(t) = A \sin(2\pi ft) \quad (13)$$

where A is the excitation amplitude, f is the excitation frequency, and t is time. The corresponding tank velocity is obtained by differentiating the displacement with respect to time:

$$U(t) = 2\pi f A \cos(2\pi ft) \quad (14)$$

This mesh motion treatment enables the numerical model to capture the transient free-surface deformation, internal velocity field, and phase-change response caused by periodic sloshing excitation.

2.6. Boil-Off Rate Evaluation

The boil-off rate (BOR) was calculated from the increase in vapor mass during the simulation. Since a certain amount of vapor hydrogen initially exists in the ullage region, the generated BOG mass was defined as the difference between the vapor mass at time (t) and the initial vapor mass. The vapor mass was obtained by integrating the vapor volume fraction and vapor density over the computational domain:

$$m_v(t) = \int_V \alpha_V \rho_V dV \quad (15)$$

The generated BOG mass is therefore expressed as Equation (16).

$$\Delta m_v(t) = m_v(t) - m_v(0) \quad (16)$$

The BOR represents the percentage of the initial liquid mass evaporated per day. Accordingly, the generated vapor mass during the simulation time Δt was converted to a daily evaporation rate by multiplying by $\frac{86400}{\Delta t}$, where 86,400 is the number of seconds in one day. The BOR was then calculated as Equation (17).

$$BOR = \frac{\Delta m_v}{m_{l,0}} \times \frac{86400}{\Delta t} \times 100 \quad (17)$$

where $m_{l,0}$ is the initial mass of liquid hydrogen. This definition enables direct comparison of BOG generation characteristics under different filling ratios, sloshing frequencies, sloshing amplitudes, and baffle configurations. The time step Δt is 0.01 s. To reduce uncertainty associated with short-time extrapolation, the representative case was simulated over multiple sloshing cycles. After the initial transient response, the tank pressure, mean temperature, and net phase-change rate showed quasi-periodic behavior. Therefore, the BOR was evaluated using the cycle-averaged vapor generation rate obtained after the quasi-periodic state was reached.

The numerical model described above was used to analyze the coupled effects of free-surface motion, heat ingress, phase change, and internal structural arrangement on the BOG generation characteristics of the Type-C liquid hydrogen tank.

3. Computational Setup and Validation

3.1. Geometry of the Type-C LH2 Tank

The numerical analysis was performed for a Type-C liquefied hydrogen (LH2) tank intended for maritime transportation. The tank geometry consisted of a cylindrical body and two hemispherical end sections in Figure 1. As summarized in Table 1, the length of the cylindrical section was 10,175 mm, and the major axis of the ends was 1200 mm. The minor axis of the head section was 640 mm. In the baseline condition, the tank was filled with liquid hydrogen to 50% of the internal volume.

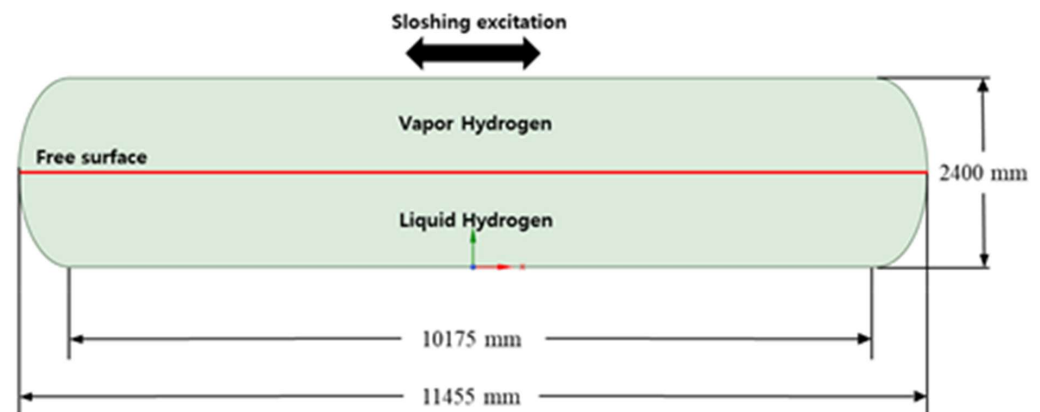


Figure 1. Schematic of the Type-C liquid hydrogen tank analysis model.

Table 1. Main dimensions of the Type-C LH2 tank.

Parameter	Value
Cylindrical tank length	10,175 mm
Head major axis	1200 mm
Head minor axis	640 mm
Baseline filling ratio	50%

To reduce computational cost while maintaining the main characteristics of the longitudinal sloshing motion, a two-dimensional model was constructed along the central longitudinal plane of the tank. This approach enables efficient parametric calculations for filling ratio, excitation frequency, excitation amplitude, and baffle configuration. The upper region of the tank was initialized as hydrogen vapor, and the lower region was initialized as liquid hydrogen. The vapor–liquid interface was defined according to the prescribed filling ratio. Accordingly, the present two-dimensional results should be interpreted as comparative trends for longitudinal sloshing-induced BOG generation rather than as final quantitative BOR predictions or direct baffle-design criteria. Fully three-dimensional simulations are required to resolve the actual mass distribution, heat-transfer paths, and geometric effects of the Type-C tank.

3.2. Thermophysical Properties of Hydrogen

The liquid phase was treated using the Boussinesq approximation because the density variation of liquid hydrogen is relatively small within the cryogenic temperature range considered in this study. The liquid density was calculated as Equation (18).

$$\rho = \rho_0[1 - \beta(T - T_0)] \quad (18)$$

where ρ is the liquid density, ρ_0 is the reference density, β is the thermal expansion coefficient, T is the local temperature, and T_0 is the reference temperature. The reference

temperature, reference density, and thermal expansion coefficient were set to $T_0 = 20.369$ K, $\rho_0 = 70.848$ kg/m³, and $\beta = 0.016684$ K⁻¹, respectively. The Boussinesq approximation was applied only to the liquid phase because the liquid hydrogen temperature remained close to the saturation temperature within a narrow cryogenic range in the present simulations. Under these conditions, density variation mainly contributes to buoyancy effects, whereas pressure-induced compressibility of the liquid phase is expected to be limited. However, for highly pressurized LH2 tanks or cases involving larger temperature variations, a real-fluid equation of state should be considered in future work.

The vapor phase was modeled as a compressible gas using the ideal-gas equation of state. Since the thermal conductivity, specific heat, viscosity, and density of hydrogen vary with temperature and pressure, these properties were not treated as constants. Instead, temperature- and pressure-dependent thermophysical properties were implemented through user-defined functions (UDFs) based on the NIST database [18]. The representative material properties at 20.369 K and 1 bar are summarized in Table 2. The ideal-gas assumption was applied to the vapor phase because the tank pressure remained close to atmospheric pressure in the present simulations. However, real-gas effects may influence local vapor density and phase-change behavior near saturation, especially under higher-pressure conditions. Therefore, the present results should be interpreted as comparative trends, and a real-gas equation of state will be considered in future work for more accurate prediction of pressurized LH2 tanks.

Table 2. Material properties of hydrogen at 20.369 K and 1 bar.

Fluid	Phase	Density (kg/m ³)	Specific Heat (kJ/kg·K)	Thermal Conductivity (mW/m·K)	Viscosity (μPa·s)
Hydrogen	Liquid	70.848	9.773	103.710	13.490
	Vapor	1.332	12.037	17.425	0.996

3.3. Mesh Generation and Grid Independence Test

A structured mesh was generated for the two-dimensional Type-C tank domain. The mesh was refined near the vapor–liquid interface and wall regions where strong gradients in volume fraction, velocity, and temperature were expected. A grid independence test was conducted to determine an appropriate mesh resolution that could provide stable numerical accuracy without excessive computational cost. The grid sensitivity was evaluated using the vapor mass at ($t = 10$) s as the comparison parameter. Five different mesh systems were considered, with the total mesh number ranging from 10,536 to 264,681 in Figure 2. As the mesh number increased, the calculated hydrogen vapor mass showed a convergent trend. In particular, when the result obtained using 105,150 cells was compared with those obtained using 159,103 and 264,681 cells, the relative differences in vapor mass were sufficiently small. Therefore, the mesh system with 105,150 cells was selected for the subsequent simulations as a compromise between accuracy and computational efficiency. For the final mesh system, an element size of 10 mm was applied. The selected mesh consisted of 58,263 elements and 57,761 nodes.

3.4. Numerical Model Validation

The main conditions of the reference experiment used for validation, including the working fluid, tank geometry, filling ratio, excitation parameters, initial conditions, and thermal boundary conditions, are summarized in the previous sections. Although the reference case is not identical to the present LH2 Type-C tank, it provides a relevant benchmark for validating the transient sloshing behavior and thermal response of the numerical model. Before applying the numerical model to LH2 sloshing simulations, the

reliability of the CFD model was examined by comparing the present numerical results with previously reported experimental data [10] in Figure 3. The validation focused on both the transient pressure response and the qualitative sloshing flow pattern.

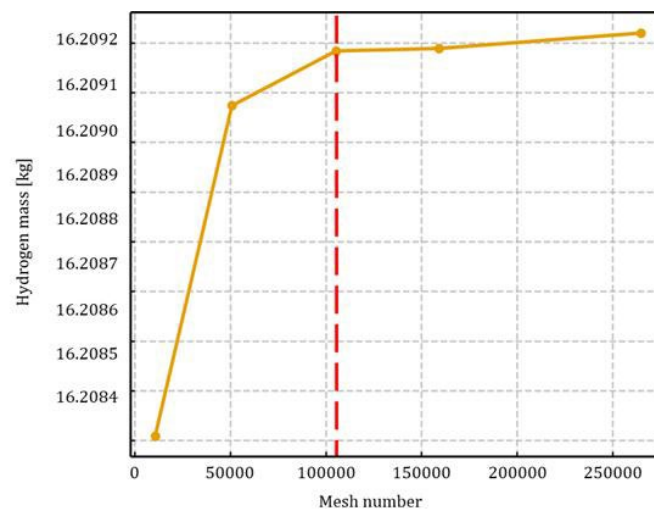


Figure 2. Comparison of vapor mass under different mesh numbers.

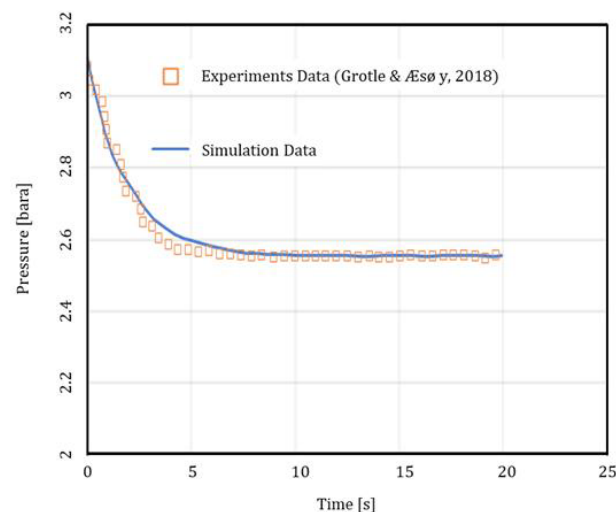


Figure 3. Comparison of tank pressure and temperature between experimental results and numerical results [10].

The predicted tank pressure showed good agreement with the reference experimental data [10] during the initial pressure variation and the subsequent quasi-steady stage. The relative difference between the numerical and experimental pressure responses was less than 5%, indicating that the present model can reproduce the coupled heat and mass transfer behavior in the tank with reasonable accuracy. In addition, the simulated free-surface motion at ($t = 5.8$) s was compared with the experimental sloshing image in Figure 4. The overall liquid motion and interface deformation were qualitatively consistent with the observed flow pattern. Based on these comparisons, the numerical model was considered suitable for analyzing the thermo-fluid behavior and BOG generation characteristics of liquid hydrogen under sloshing conditions.

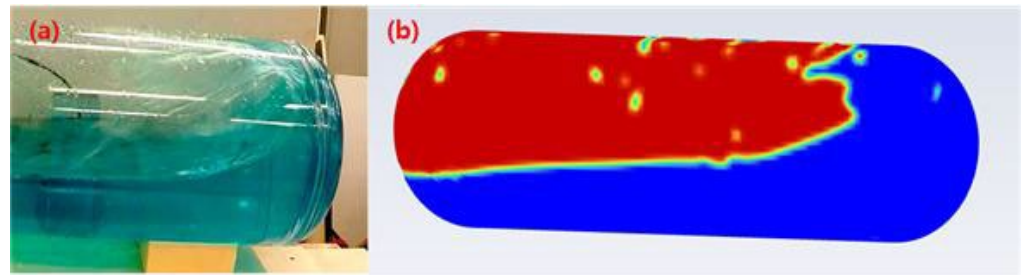


Figure 4. Sloshing flow at $t = 5.8$ s (a) experimental result [10], (b) simulation result.

3.5. Boundary and Initial Conditions

The thermal boundary condition was defined to represent external heat ingress into the Type-C LH2 tank in Figure 5. In the present model, the tank wall thickness and detailed insulation layers were not explicitly modeled. Instead, the heat transfer from the external environment to the cryogenic fluid was represented using an equivalent convective heat-transfer coefficient. The ambient temperature was set to 273.15 K, and the inner wall temperature was assumed to be 20 K based on the saturation temperature of hydrogen at atmospheric pressure. A uniform wall heat flux of 3.82 W/m^2 was applied to the tank wall. The equivalent convective heat-transfer coefficient was calculated as Equation (19).

$$h = \frac{q''}{T_e - T_{\text{wall}}} \quad (19)$$

where h is the convective heat-transfer coefficient, q'' is the applied wall heat flux, T_e is the ambient temperature, and T_{wall} is the inner wall temperature. The calculated value of h was $0.015 \text{ W/(m}^2 \cdot \text{K)}$. This uniform heat-flux condition represents an idealized thermal boundary adopted for comparative parametric analysis. In practical LH2 tanks, heat leakage can be spatially non-uniform because of insulation joints, support structures, penetrations, and local environmental conditions. Therefore, the present results should be interpreted as BOG-generation trends under simplified heat-ingress conditions, and future work should include non-uniform heat leakage and conjugate heat transfer through the tank wall and insulation system.

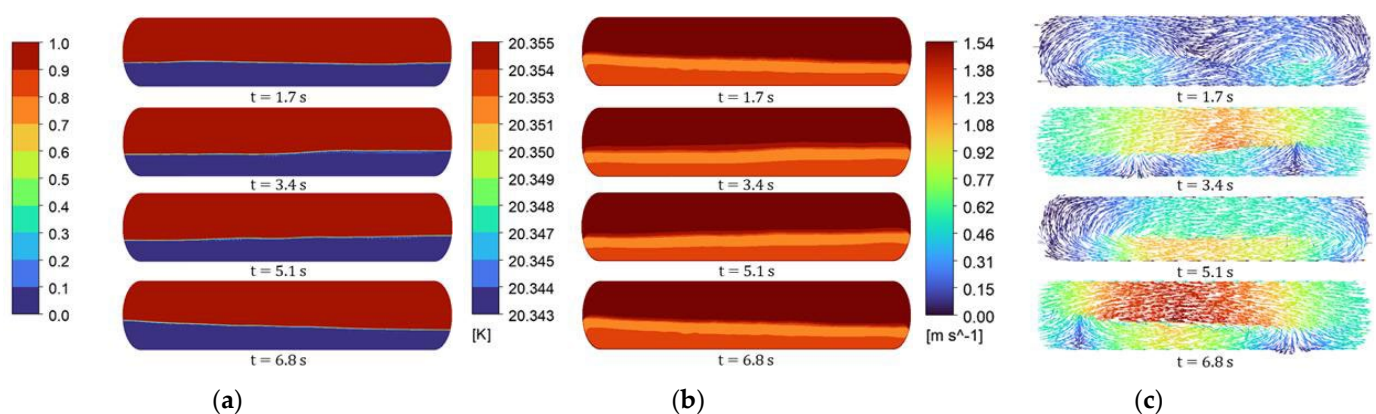


Figure 5. Thermo-fluid field characteristics under 30% filling ratio. (a) Volume fraction, (b) temperature distribution, and (c) velocity vector fields at different time steps.

The sloshing excitation was imposed in the longitudinal direction of the tank. The tank displacement and corresponding velocity imposed on the moving mesh obtained by differentiating the displacement were prescribed as a sinusoidal function in Equations (13) and (14). The initial pressure in the tank was set to 1 atm. The initial temper-

ature was defined based on the saturation condition of hydrogen. To obtain a numerically stable initial condition, a steady-state calculation was first performed under the prescribed thermal boundary condition. The resulting steady-state pressure, temperature, and phase distributions were then used as the initial condition for the transient sloshing simulation. This procedure was adopted to minimize artificial phase change at the initial time step and to ensure that the transient response was governed primarily by the imposed sloshing excitation and heat ingress.

3.6. Numerical Procedures

The SIMPLE algorithm was used for pressure–velocity coupling. The least-squares cell-based method was adopted for gradient evaluation, and the body-force-weighted scheme was used for pressure discretization. The density, momentum, and energy equations were discretized using second-order upwind schemes, whereas the turbulent kinetic energy and turbulent dissipation rate equations were discretized using first-order upwind schemes. The compressive scheme was applied for volume-fraction discretization to sharpen the liquid–vapor interface. The transient formulation was treated using the first-order implicit scheme. The convergence criteria were set to 10^{-3} for continuity, momentum, turbulent kinetic energy, and turbulent dissipation rate and 10^{-6} for energy.

In this study, the “steady-state calculation” refers to an initial static initialization step before the transient sloshing simulation. Because the tank is closed and subjected to continuous positive heat ingress, this state should not be interpreted as a true thermodynamic steady state. Instead, the static calculation was used to obtain a stable initial pressure, temperature, and phase distribution, which were then used as the initial condition for the transient sloshing simulations.

4. Results

4.1. Effect of Filling Ratio

The effect of filling ratio on the thermo-fluid behavior and BOG generation was first examined under the same sloshing excitation condition. The simulation cases for the filling-ratio analysis are summarized in Table 3. The excitation amplitude and frequency were fixed at 0.2 m and 0.15 Hz, respectively, while the filling ratio was varied as 30%, 50%, and 80%.

Table 3. Numerical simulation cases classified by filling ratio.

Amplitude, A (m)	Frequency, f (Hz)	Filling Ratio (%)
0.2	0.15	30
0.2	0.15	50
0.2	0.15	80

Figures 5–7 show the temporal variations in the liquid volume fraction, temperature field, and velocity vector field for the 30%, 50%, and 80% filling-ratio conditions, respectively. Under the 30% filling-ratio condition, as shown in Figure 8, the liquid volume was relatively small, and the deformation of the free surface remained moderate throughout the sloshing cycle. The maximum velocity was approximately 1.54 m/s. The temperature field indicates that heat accumulation occurred mainly near the tank wall and in the vapor region. Because the ullage volume was relatively large, the vapor phase was more directly affected by wall heat ingress, resulting in a higher temperature rise than that observed in the liquid region. In the 50% filling-ratio case, shown in Figure 6, the free-surface deformation was more pronounced than in the other filling conditions. The liquid moved periodically in the longitudinal direction, and the inclination of the liquid–vapor interface changed clearly

with time. The maximum velocity reached approximately 6.688 m/s, which was the highest value among the three filling-ratio cases. Local temperature variations were observed near the upper wall and head regions; however, the magnitude of the temperature change was small, on the order of 0.01 K. For the 80% filling-ratio condition, shown in Figure 7, most of the tank volume was occupied by liquid hydrogen. As a result, the free-surface motion was geometrically restricted compared with the lower filling-ratio cases. The temperature distribution was mainly characterized by localized thermal variation near the liquid surface and wall boundary regions.

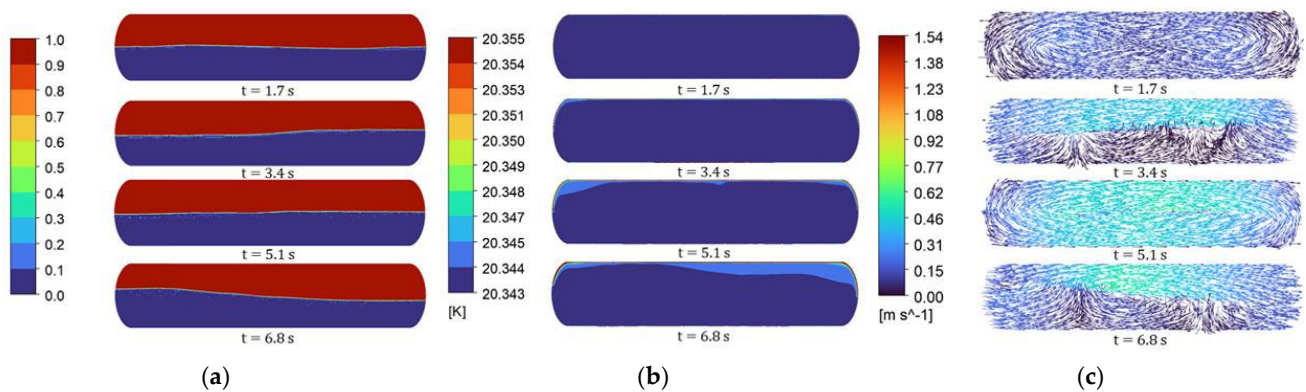


Figure 6. Thermo-fluid field characteristics under 50% filling ratio. (a) Volume fraction, (b) temperature distribution, and (c) velocity vector fields at different time steps.

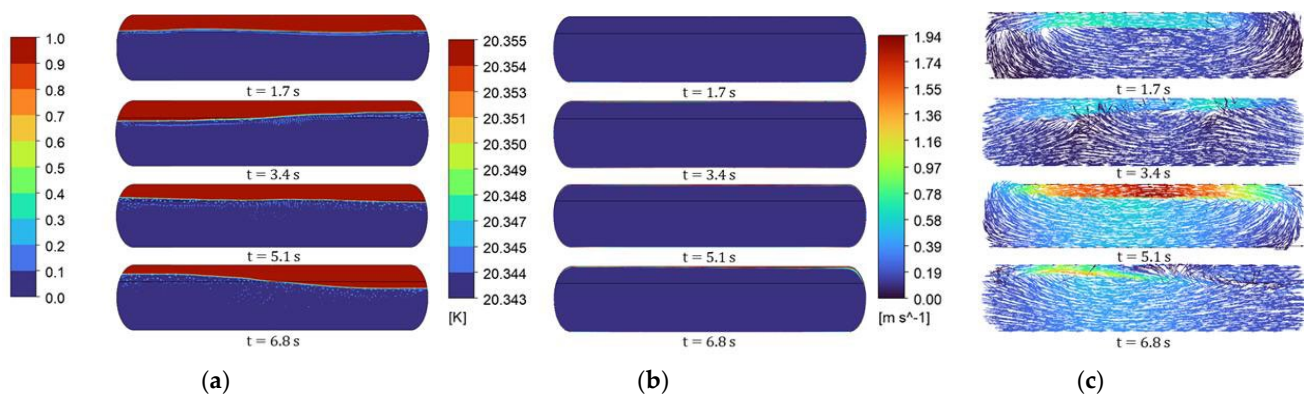


Figure 7. Thermo-fluid field characteristics under 80% filling ratio. (a) Volume fraction, (b) temperature distribution, and (c) velocity vector fields at different time steps.

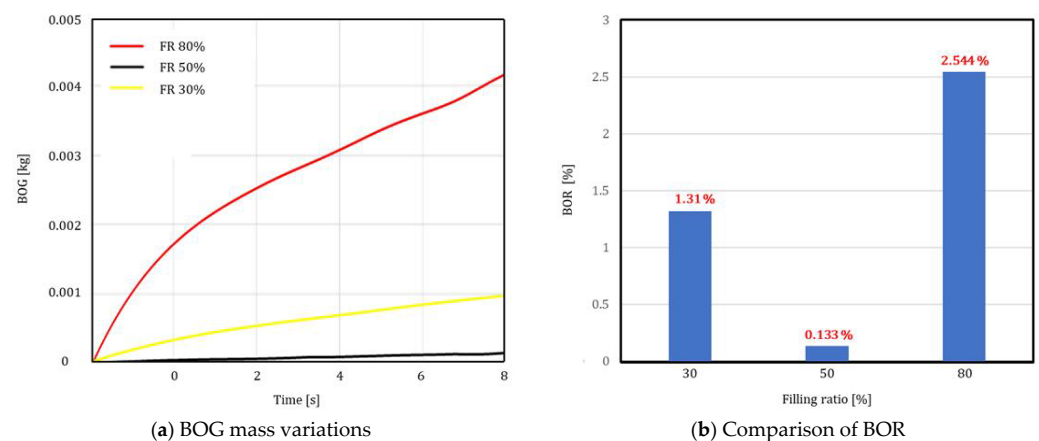


Figure 8. BOG mass and BOR variations at different filling ratio.

It should be noted that the BOR values obtained in this study are higher than typical design targets for insulated marine cryogenic tanks. Previous studies reported target BOR values of approximately 0.085%/day for large LNG cargo containment systems and approximately 0.2%/day for membrane-type LH2 cargo containment systems. Therefore, the present BOR values should be interpreted as comparative indicators under simplified heat-ingress and sloshing conditions rather than as final design BORs for fully insulated marine LH2 tanks. The relatively high BOR values, especially under large-amplitude excitation, are attributed to the idealized uniform heat-flux boundary condition and the enhanced liquid–vapor interfacial heat and mass transfer caused by severe sloshing.

Because the ullage volume was small, the vapor space experienced more rapid thermal response to heat ingress, while the large liquid volume increased the total amount of cryogenic fluid involved in the phase-change process.

The BOG mass variations for the different filling ratios are compared in Figure 8a, and the corresponding BOR values are presented in Figure 8b. The largest BOG generation was observed at the 80% filling ratio, followed by the 30% and 50% filling-ratio cases. The calculated BOR values were approximately 2.544%, 1.31%, and 0.133% for the 80%, 30%, and 50% filling-ratio cases, respectively. Thus, the minimum BOR was obtained at the 50% filling ratio. Although the 50% filling case showed relatively noticeable free-surface deformation, the BOR remained the lowest. This indicates that BOG generation is not determined solely by interface deformation but by the combined effects of vapor volume, liquid thermal capacity, wall–liquid contact area, and heat distribution. This result suggests that BOG generation is not determined solely by liquid volume or ullage volume but by the combined effect of free-surface deformation, vapor heating, and liquid thermal capacity. At a low filling ratio, the thin liquid layer is more sensitive to heat ingress, while at a high filling ratio, the reduced ullage volume and large wetted area promote vapor generation. The 50% filling-ratio condition showed the most stable balance between these competing effects.

4.2. Effect of Sloshing Frequency

The influence of sloshing frequency was investigated by fixing the filling ratio and excitation amplitude at 50% and 0.2 m, respectively. The simulation cases are listed in Table 4. Three excitation frequencies, 0.1 Hz, 0.15 Hz, and 0.2 Hz, were considered to evaluate the frequency-dependent thermal and hydrodynamic responses. The excitation frequencies were selected based on the estimated first natural sloshing frequency of the tank. Using an equivalent longitudinal sloshing model, the first natural frequency was estimated to be approximately 0.15 Hz. Therefore, the 0.15 Hz case was regarded as the near-resonance condition, whereas 0.1 Hz and 0.2 Hz were selected as lower- and higher-frequency off-resonance conditions, respectively.

Table 4. Numerical simulation cases classified by sloshing frequency.

Amplitude, A (m)	Frequency, f (Hz)	Filling Ratio (%)
0.2	0.1	50
0.2	0.15	50
0.2	0.2	50

Figure 9 shows the liquid volume fraction, temperature distribution, and velocity vector fields for the different sloshing frequencies. As the excitation frequency increased, the liquid motion became more active and the velocity field intensified throughout the tank. In particular, the cases with 0.15 Hz and 0.2 Hz showed higher flow velocities than the

0.1 Hz case. The temperature contours also indicate that the temperature rise in the upper vapor region increased with frequency, especially near the left and right head sections.

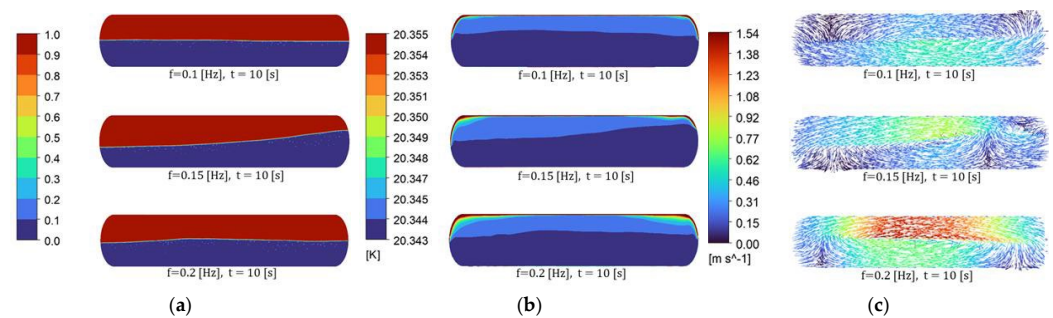


Figure 9. Thermo-fluid field characteristics according to sloshing frequencies. (a) Volume fraction, (b) temperature distribution, and (c) velocity vector fields.

The frequency dependence of BOG generation is shown in Figure 10. The lowest BOG generation was obtained at 0.1 Hz, whereas the BOG mass increased as the frequency increased to 0.15 Hz and 0.2 Hz. The BOR values calculated from the vapor mass variations are compared in Figure 11. The BOR increased from 0.095% at 0.1 Hz to 0.133% at 0.15 Hz and 0.175% at 0.2 Hz. This trend indicates that higher sloshing frequency enhances the internal flow velocity and promotes convective heat transfer between the heated wall, vapor phase, and liquid–vapor interface. However, the increase in BOR caused by frequency variation was relatively moderate compared with that caused by amplitude variation in the next chapter. This is because, under the same excitation amplitude, changing only the frequency increases the liquid velocity but does not necessarily produce a proportional increase in the free-surface deformation area.

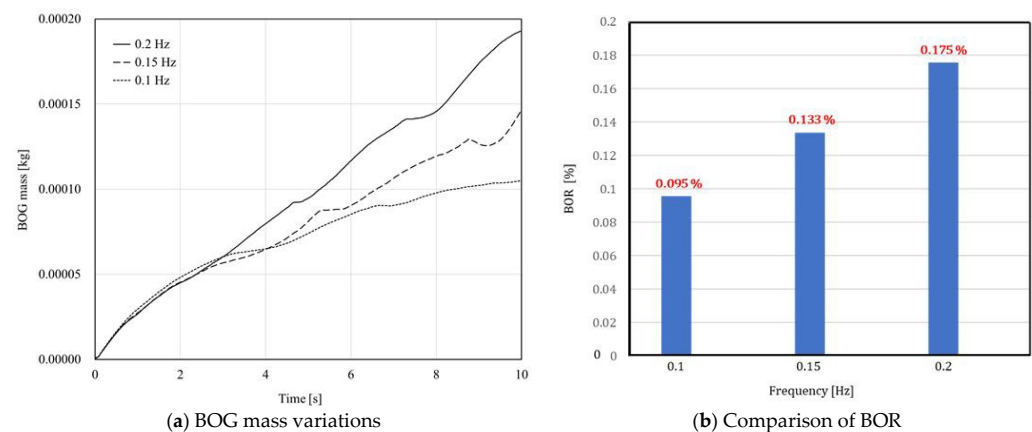


Figure 10. BOG mass and BOR variations according to sloshing frequencies.

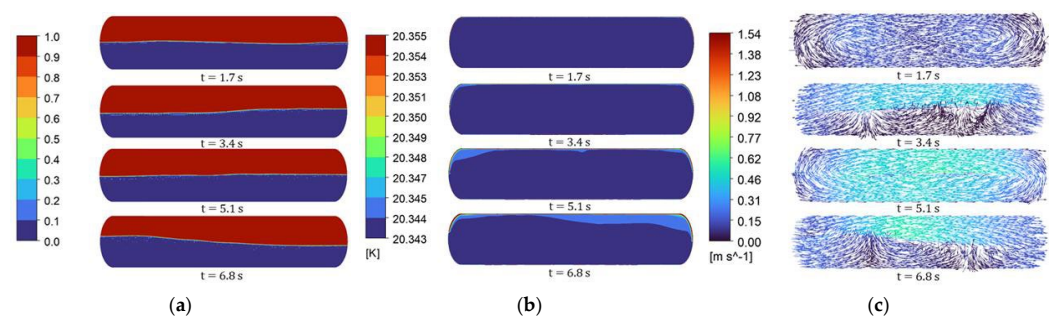


Figure 11. Thermo-fluid field characteristics under the 0.2 m amplitude condition. (a) Volume fraction, (b) temperature distribution, and (c) velocity vector fields at different time steps.

4.3. Effect of Sloshing Amplitude

The effect of excitation amplitude was examined under a fixed filling ratio of 50% and a fixed frequency of 0.15 Hz. The simulation cases are summarized in Table 5. The excitation amplitude was varied from 0.2 m to 0.6 m and 1.0 m.

Table 5. Numerical simulation cases classified by amplitude.

Amplitude, A (m)	Frequency, f (Hz)	Filling Ratio (%)
0.2	0.15	50
0.6	0.15	50
1.0	0.15	50

Figures 11–13 show the volume fraction, temperature distribution, and velocity vector fields for excitation amplitudes of 0.2 m, 0.6 m, and 1.0 m, respectively. In the 0.2 m case shown in Figure 11, the free-surface elevation was relatively small, and the liquid motion remained stable during the sloshing cycle. The maximum velocity was approximately 1.54 m/s, which was the lowest value among the amplitude cases. The temperature field showed local heat accumulation in the upper vapor region and near the head sections, but the overall thermal disturbance remained limited. When the amplitude increased to 0.6 m, as shown in Figure 12, the free-surface deformation became more pronounced. The liquid reached a higher elevation near the tank ends, and the maximum velocity increased to approximately 6.68 m/s. The temperature rises in the vapor region also became larger than that in the 0.2 m case, particularly near the upper head region. This indicates that the larger excitation amplitude intensified both liquid motion and convective heat transfer.

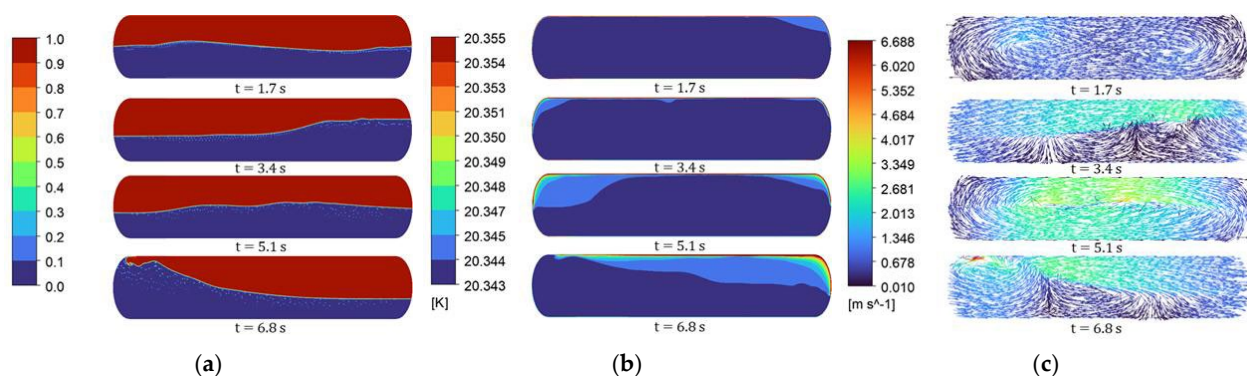


Figure 12. Thermo-fluid field characteristics under the 0.6 m amplitude condition. (a) Volume fraction, (b) temperature distribution, and (c) velocity vector fields at different time steps.

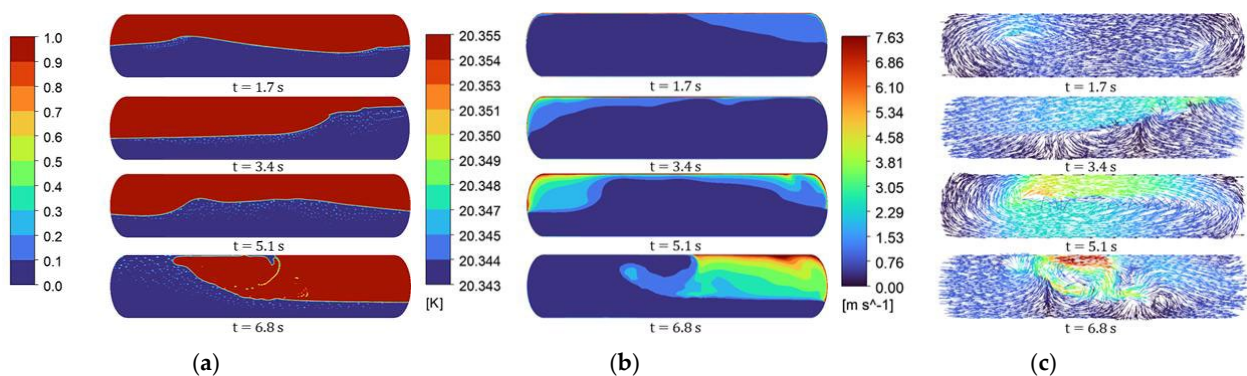


Figure 13. Thermo-fluid field characteristics under the 1.0 m amplitude condition. (a) Volume fraction, (b) temperature distribution, and (c) velocity vector fields at different time steps.

The strongest sloshing behavior was observed at the 1.0 m amplitude, as shown in Figure 13. The free surface underwent the largest deformation, and the maximum velocity reached approximately 7.63 m/s. The temperature contours showed a larger high-temperature region in the upper vapor space than in the lower-amplitude cases. As the amplitude increased, the liquid–vapor interfacial area expanded and the interaction between the cryogenic liquid and warmer vapor became more active. In addition, the higher flow velocity enhanced heat transfer from the heated tank wall to the fluid.

Figure 14a compares the BOG mass variations for the different excitation amplitudes, and Figure 14b presents the corresponding BOR values. The BOG generation was lowest at 0.2 m and increased significantly as the amplitude increased to 0.6 m and 1.0 m. The calculated BOR values were 0.133%, 1.403%, and 4.329% for amplitudes of 0.2 m, 0.6 m, and 1.0 m, respectively. Compared with the frequency effect, the amplitude effect was much more dominant. This result demonstrates that excitation amplitude is a critical parameter governing BOG generation because it directly determines the magnitude of free-surface deformation, internal velocity, and interfacial heat and mass transfer.

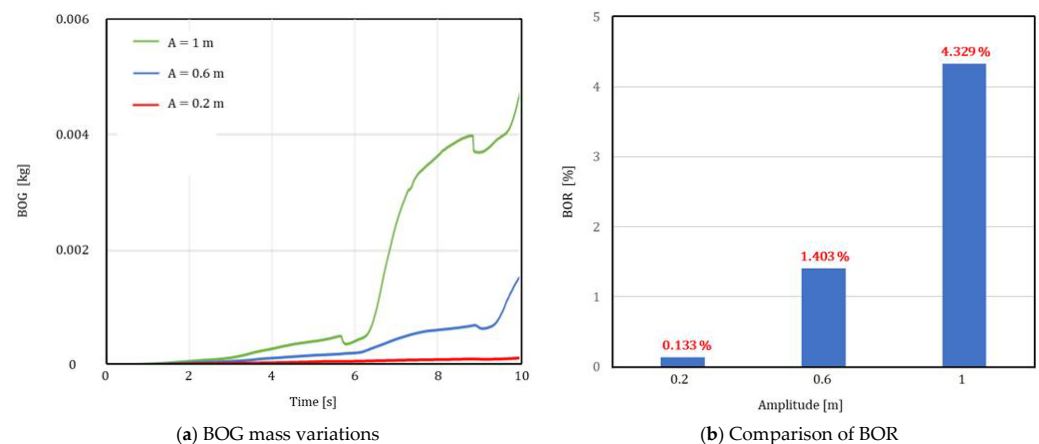


Figure 14. BOG mass and BOR variations according to sloshing amplitudes.

4.4. Effect of Baffle Installation

The influence of baffle installation was investigated by comparing the cases with and without baffles under the same sloshing condition (0.22 Hz, 50%). Figure 15 compares the liquid volume fraction distributions for the non-baffle and baffle-installed cases. In the tank without baffles, the liquid moved more freely along the longitudinal direction, and the free-surface deformation was relatively large. In contrast, when baffles were installed, the liquid motion was divided into several internal regions, and the free-surface deformation became more moderate. Figure 16 shows the temporal variation in hydrogen vapor mass for the two cases. Although the baffles suppressed the sloshing motion from a hydrodynamic perspective, the baffle-installed case showed a larger vapor mass than the non-baffle case over the entire simulation period. The difference became more distinct after approximately 7 s. The final hydrogen vapor mass was 0.016999 kg for the non-baffle case and 0.019757 kg for the baffle-installed case. This result indicates that the baffle has two competing effects. First, it suppresses free-surface motion and reduces sloshing intensity. Second, it provides additional solid structures that can conduct heat into the cryogenic fluid. In the present simulation, the thermal effect associated with the baffle structure was more influential than the hydrodynamic suppression effect, leading to a larger BOG generation in the baffle-installed case.

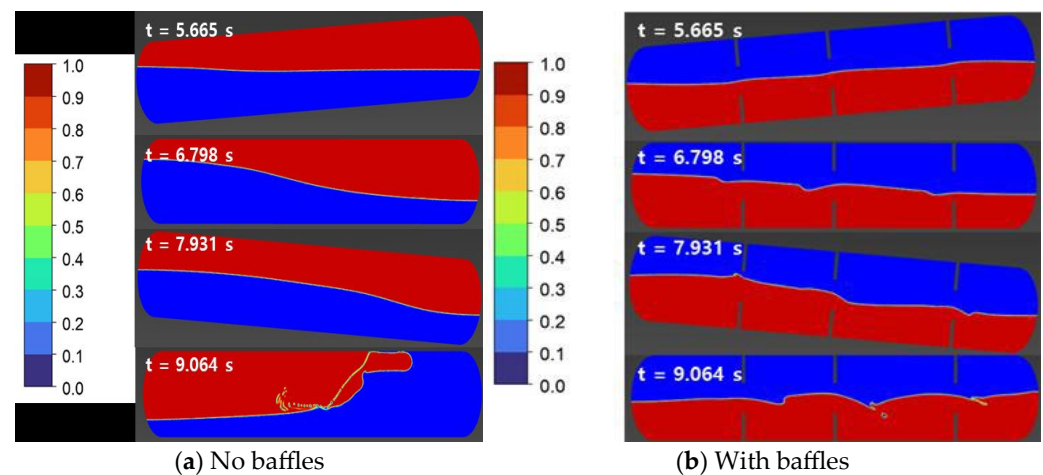


Figure 15. Comparison of volume fraction distribution with and without baffles.

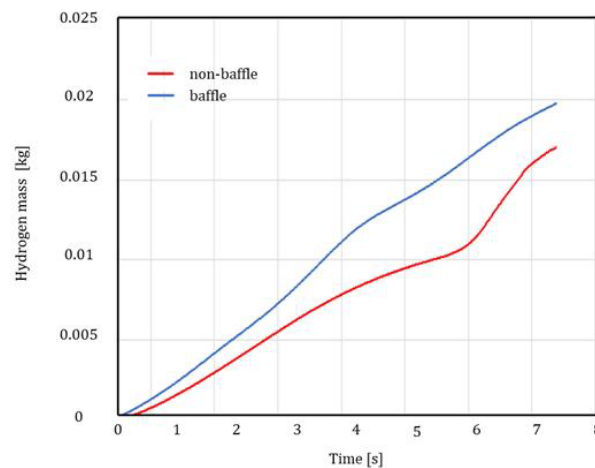


Figure 16. Hydrogen vapor mass variations for non-baffle and baffle-installed cases.

Because conjugate heat transfer through the actual steel baffle was not explicitly modeled, the increase in BOG should not be interpreted solely as a direct conductive heat-transfer effect through the baffle material. Rather, it reflects the combined influence of the simplified solid-boundary treatment, modified internal flow structure, liquid–vapor interfacial behavior, and thermal interaction near the baffle surfaces. Therefore, a more detailed model including baffle thickness, material properties, and conjugate heat transfer is required for quantitative evaluation of the thermal role of baffles.

4.5. Effect of the Number of Baffles

The effect of the number of baffles was further analyzed by comparing two-baffle and three-baffle configurations. Figure 17 shows the volume fraction and temperature distributions for the two configurations. In the two-baffle case, the free surface showed larger deformation, and the overall liquid motion was more intense.

In contrast, the three-baffle configuration divided the internal flow region into smaller sections, which reduced the free-surface deformation and stabilized the internal liquid motion. The temperature distribution also differed between the two cases. In the two-baffle configuration, a relatively strong local temperature rise was observed in the upper vapor region, particularly near the head section. In the three-baffle configuration, the reduced free-surface deformation weakened vapor-phase heat accumulation, and the overall high-temperature region became less pronounced. Although the additional baffle could increase the solid conduction path, the suppression of sloshing-induced thermal disturbance had

a greater influence in the three-baffle case. The temperature distribution also differed between the two cases. In the two-baffle configuration, a relatively strong local temperature rise was observed in the upper vapor region, particularly near the head section. In the three-baffle configuration, the reduced free-surface deformation weakened vapor-phase heat accumulation, and the overall high-temperature region became less pronounced. Although the additional baffle could increase the solid conduction path, the suppression of sloshing-induced thermal disturbance had a greater influence in the three-baffle case.

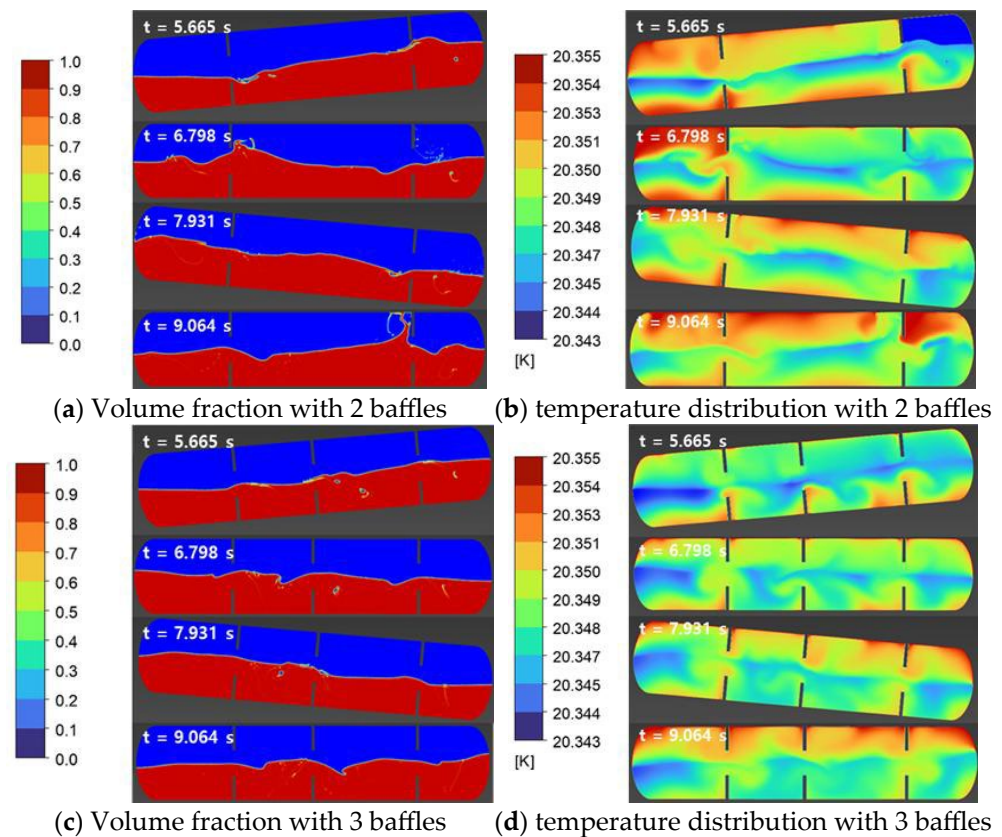


Figure 17. Comparison of volume fraction and temperature distributions in the Type-C LH2 tank with different numbers of baffles under sloshing excitation: (a,b) two-baffle configuration and (c,d) three-baffle configuration.

Figure 18 compares the hydrogen vapor mass–time histories for the two baffle configurations. The two-baffle case generated a larger amount of BOG throughout the simulation. The final vapor mass was also higher than that in the three-baffle case. This result suggests that increasing the number of baffles can reduce BOG generation when the resulting suppression of free-surface motion and vapor heating outweighs the additional heat conduction through the baffle structure. Overall, the results show that BOG generation in a Type-C LH2 tank under sloshing conditions is governed by the coupled effects of liquid motion, interfacial deformation, vapor heating, and conductive heat transfer through internal structures. Among the examined parameters, excitation amplitude had the most significant effect on BOR, while sloshing frequency produced a moderate increase in BOG generation. The baffle results further indicate that the thermal role of internal structures must be considered together with their hydrodynamic sloshing-suppression performance when designing LH2 tanks for maritime transportation.

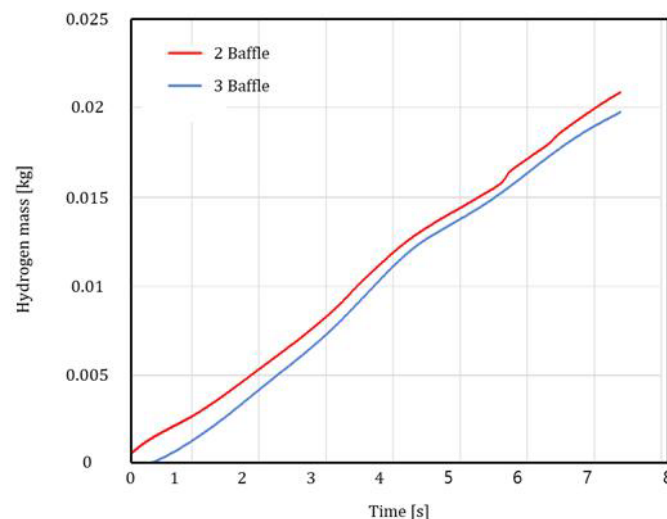


Figure 18. Time histories of hydrogen vapor mass in the Type-C LH2 tank for different baffle configurations.

5. Summary and Discussion

In this study, the thermo-fluid behavior and boil-off gas (BOG) generation characteristics of liquid hydrogen in a Type-C tank under sloshing conditions were numerically investigated. A transient two-phase CFD model was developed using the VOF method to capture liquid–vapor interface motion and the Lee phase-change model to calculate evaporation and condensation. The effects of filling ratio, sloshing frequency, sloshing amplitude, baffle installation, and the number of baffles were systematically examined. The main conclusions of this study are summarized as follows:

The filling ratio had a significant influence on the internal thermal response and BOG generation. At a low filling ratio, the relatively large ullage volume was more directly affected by external heat ingress, leading to a temperature rise in the vapor region. At a high filling ratio, the liquid occupied most of the tank volume, and the reduced ullage space and increased wetted region promoted vapor generation. Among the filling ratios considered in this study, the 50% filling condition showed the lowest BOG generation and BOR, indicating that the balance between ullage heating and liquid thermal capacity was most stable at the intermediate filling level.

Increasing the sloshing frequency from 0.1 Hz to 0.2 Hz intensified the internal velocity field and increased the temperature rise in the upper vapor region, particularly near the tank heads. Consequently, the BOG generation and BOR increased with frequency. However, the effect of frequency was relatively moderate compared with that of sloshing amplitude. This indicates that, under the same excitation amplitude, frequency variation mainly enhances the flow velocity but does not cause a proportional increase in free-surface deformation.

Sloshing amplitude was identified as the most influential excitation parameter governing the thermo-fluid behavior of the Type-C LH2 tank. As the amplitude increased from 0.2 m to 0.6 m and 1.0 m, the free-surface deformation, internal velocity, and vapor-region temperature rise all increased substantially. The maximum velocity increased from 1.54 m/s to 6.68 m/s and 7.63 m/s, respectively. The largest BOG generation was observed at the 1.0 m amplitude. This result indicates that large-amplitude sloshing increases the liquid–vapor interfacial area and enhances convective heat transfer near the tank wall, thereby accelerating evaporation.

The installation of baffles reduced the overall free-surface motion and suppressed violent sloshing patterns from a hydrodynamic viewpoint. However, the baffle-installed case generated more vapor mass than the non-baffle case. This result suggests that, for LH2

tanks, baffles should not be evaluated only as sloshing-suppression devices. The additional solid structures can provide conductive heat-transfer paths into the cryogenic fluid, and this thermal penalty may increase BOG generation even when liquid motion is reduced.

The number of baffles also affected the coupled sloshing and thermal response. Compared with the two-baffle configuration, the three-baffle configuration divided the internal flow region more effectively and reduced free-surface deformation. In the present simulation, the three-baffle case generated less BOG than the two-baffle case, indicating that additional baffles can reduce evaporation when the sloshing-suppression effect becomes larger than the increase in conductive heat transfer.

Overall, the BOG generation characteristics of a Type-C LH2 tank under sloshing conditions were strongly affected by filling ratio, excitation amplitude, and baffle configuration. In particular, sloshing amplitude and baffle arrangement were found to be dominant factors controlling evaporation behavior. The results of this study provide useful numerical data for BOG prediction, baffle arrangement optimization, and thermal safety assessment of Type-C LH2 tanks for maritime transportation.

Sloshing may have competing effects on the thermal response of cryogenic tanks. On one hand, fluid motion can disrupt thermal stratification and promote the utilization of the cold capacity of the bulk liquid, thereby reducing local vaporization and tank pressure [20]. On the other hand, severe sloshing increases the liquid–vapor interfacial area and enhances convective heat transfer, which can promote BOG generation. In the present analysis, the increase in BOG was mainly interpreted from the latter mechanism. However, the present model does not explicitly quantify the mechanical work transferred from the moving tank wall to the fluid, and future work should include a detailed energy-budget analysis considering both external heat ingress and mechanical work input.

The present two-dimensional model represents the central longitudinal plane of the Type-C tank and was adopted to perform a systematic parametric analysis with reasonable computational cost. However, actual sloshing in a cylindrical tank is inherently three-dimensional, particularly under large-amplitude excitation. Therefore, transverse circulation, spanwise vortical structures, local wave breaking, and three-dimensional turbulent mixing cannot be fully resolved in the present model. These effects may influence the instantaneous liquid–vapor interfacial area and local phase-change rate. Accordingly, the present results should be interpreted as comparative trends for longitudinal sloshing-induced BOG generation rather than as complete three-dimensional design-level predictions. Future work will extend the present model to fully three-dimensional simulations and experimental validation.

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