Thermodynamic study on condensation heat transfer and sloshing of the liquid hydrogen storage

Zhengtai Xinjia[a,b], Xiaoqing Zhang[*a,b], Chengjia Xu[a,b]
[a]State Key Laboratory of Ocean Engineering, Shanghai Jiao Tong University, Shanghai 200240, China; [b]Collaborative Innovation Center for Advanced Ship and Deep-Sea Exploration, Shanghai Jiao Tong University, Shanghai, 200240, China

ABSTRACT

Liquid hydrogen (LH_2), which has been the subject of considerable research and development, is a promising candidate for the transport and storage of hydrogen in future energy systems. Nevertheless, the harsh storage and transportation conditions pose a significant challenge. A precise thermodynamic model is of paramount importance for the accurate simulation of the storage and transportation processes of LH_2. The objective of this study is to propose a modified thermal stratified model that considers condensation heat transfer between hydrogen vapor and the inner wall. The experiment data obtained from the K-site LH_2 tank was employed to introduce a correlation equation of the Nusselt number (Nu). This equation incorporates the influences of the filling ratio (FR) and heat loss rate (HLR) on condensation heat transfer. The modified model exhibited an error rate of less than 3%. Subsequently, the modified model was employed in an investigation of sloshing excitation, with a comparison being made with the Saturn IBAS-203 LH_2 storage tank experiment data. The modified model exhibited an error range of 8%. This indicates that the model is highly accurate in simulating long-distance LH_2 transportation at sea.

Keywords: Liquid hydrogen, cryogenic storage, thermal stratified model, condensation heat transfer, sloshing

1. INTRODUCTION

Hydrogen energy stands out in the realm of clean and renewable energy sources, recognized for its low emissions and high energy conversion efficiency[1]. However, the advancement of hydrogen energy research faces significant challenges in devising viable storage and transportation methods[2]. Various approaches are being explored, such as compression, liquefaction, and the use of physical or chemical carriers for storage[3]. Liquid hydrogen (LH_2), celebrated for its high energy density and efficient storage and transportation capabilities, has become a focal point of research[4].

Due to the complexities involved in experimental studies of LH_2 storage and transportation, many researchers have turned to simulation modeling[5]. Several models have been developed to explore cryogenic liquid storage, including the saturated homogeneous model[6], the Russian model[7], the thermal multi-zone model[8], and the thermal stratified model[9], among others.

This study introduces a modified thermal stratified model specifically designed for LH_2, using a modified Nu number correlation equation to take the condensation film heat transfer during evaporation from storage tanks into account. The model was calibrated and validated using data from two liquid hydrogen experiments to ensure good accuracy and confidence.

2. MODIFIED THERMAL STRATIFIED STORAGE MODEL FOR LH_2

2.1 Model parameters and thermophysical properties of LH_2

The simulation subject of the study is the K-site tank. Constructed by NASA, the K-site tank serves as a liquid hydrogen (LH_2) storage unit for experimental purposes. It possesses a volume of 4.89 m^3, stands 2.2 m tall, features a long-to-short axis ratio of 1.2, and can withstand a critical pressure of 450 kpa[10]. The HLR for the K-site can be adjusted to 0.35, 2.00, 3.50, or 8.00 W/m^2, and the FR can be set to 29%, 49%, or 83%.

*zxq186@sjtu.edu.cn

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In simulation modeling, the tank's internal compartment is segmented into four zones: the superheated hydrogen zone (SHZ), thermal stratification zone (TSZ), boundary layer zone (BLZ), and main body zone (MBZ), as shown in Figure 1. There is a gas-liquid Interface (GLI) between the SHZ and TSZ. The modeling process incorporates several assumptions:

(1) HLR is constant. (2) The GLI is treated as a saturated film, with the assumption that heat transfer occurs only at the GLI and BLZ. (3) Within SHZ, hydrogen gas condenses, creating a condensation film (CF) along the inner wall.

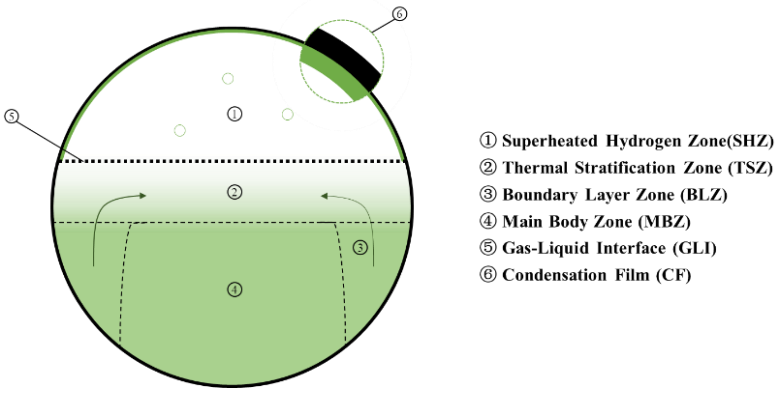

Figure 1. Schematic diagram of the four-zone storage model.

The main expressions of the model are described as follows[11]:

The total heat balance equation:

$$Q = Q_l + Q_g = \dot{q} \cdot \Delta t \tag{1}$$

Q is the total heat absorption. Q_g and Q_l are the gas phase and liquid phase heat absorption. Δt is the heat transfer time.

The equation of BLZ:

$$Q_w = h_f \cdot (T_b - T_l) \cdot S_s \tag{2}$$

Q_w is the heat transfer rates between TSZ and MBZ. T_b is the BLZ temperature. T_l is the MBZ temperature. S_s is the surface area of MBZ, which is determined by the height of MBZ. h_f is the heat transfer coefficient of the LH_2, which is set to 1.04 W/(m^2·K) to simplify the calculation in this study[10].

The equations of MBZ:

$$H_l{}^{(t+1)} = H_l{}^{(t)} + (Q_l + Q_w)/(m_l - m_e) \tag{3}$$

$$m_l^{(t+1)} = m_l^{(t)} - m_e^{(t)} \tag{4}$$

The superscript t and t+1 represent the value of the physical quantity at t and t+1 time. H_l is the enthalpy of MBZ. m_l is the mass of MBZ. m_e is the evaporation amount of MBZ

The equations of GLI:

$$Q_f = h_f \cdot (T_g - T_s) \cdot S \tag{5}$$

$$G_f = m_e \cdot (H_l - H)/\gamma + (Q_f - Q_w - Q_z)/(H_g - H_0) \tag{6}$$

Q_f is the heat transfer rates between SHZ and TSZ. Q_z is the condensation heat transfer in SHZ. T_g and T_s are the temperature of SHZ and TSZ. S is the area of GLI. G_f is the evaporation amount of GLI. γ is the latent heat of vaporization of LH_2. H is the enthalpy of the saturated liquid. H_g and H_0 is the enthalpy of SHZ and MBZ.

The equations of TSZ:

$$H_s{}^{(t+1)} = H_s{}^{(t)} + (Q_f - Q_w + Q_z)/(m_s + m_e - G_f) \tag{7}$$

$$m_s{}^{(t+1)} = m_s{}^{(t)} + m_e{}^{(t)} - G_f \tag{8}$$

H_s is the enthalpy of TSZ; m_s is the mass of TSZ.

The equations of SHZ:

$$Q_z = h_k \cdot (T_g - T_s) \cdot S_g \tag{9}$$

$$H_g{}^{(t+1)} = H_g{}^{(t)} + (Q_g - Q_f + Q_z)/(m_g + G_f) \tag{10}$$

$$m_g{}^{(t+1)} = m_g{}^{(t)} + G_f \tag{11}$$

$$\rho_g = m_g{}^{(t+1)}/V_g \tag{12}$$

$$h_k = Nu \cdot \lambda/\delta \tag{13}$$

S_g is the contact area between the gas phase and the wall. m_g is the mass of SHZ; ρ_g and V_g are the density and volume of SHZ. h_k is the condense heat transfer coefficient between the cryogenic gas and the wall. *Nu* (Nusselt number) is used to calculate the h_k, λ is the heat coefficient, and δ is the characteristic length.

2.2 A modified correlation equation of Nu number

Prior research often treats the heat exchange between the inner wall and the vapor as natural convection heat transfer, leading to h_k being overlooked. However, the temperature differential between the inner wall and the hydrogen vapor may lead to the formation of a condensation film at the vapor-wall interface. When this film forms, condensation heat transfer predominates over other modes of heat exchange between the vapor and the inner wall. To simulate this phenomenon effectively, Nu number is introduced.

In the domain of condensation film heat transfer, a power law correlation exists between Nu number and Ra number[12]. Building on this correlation, this study introduces an empirical Nusselt number formula that accounts for the impact of condensation heat transfer. This formula improves the precision of the thermal stratified model by incorporating the influences of HLR and FR on the condensation heat transfer process. The proposed formula is detailed in equation (14).

$$Nu = CRa^m \left(\frac{H}{\delta}\right)^n \tag{14}$$

$$Ra = Gr \cdot Pr \tag{15}$$

In equations (14) and (15), *Ra* is defined as the product of *Gr* and *Pr*. *Gr*, the Grashof number, represents the ratio of buoyancy to viscous forces within the fluid, while *Pr*, the Prandtl number, is a dimensionless number that describes the relationship between the thermal and velocity boundary layers. The height of the liquid, represented by *H*, is determined by FR. The coefficients *C*, *m*, and *n* are associated with the *Nu* number, which is derived from experimental data. These coefficients account for the combined effects of FR and HLR on condensation heat transfer. Coefficient *C* varies solely with HLR, indicating the influence of HLR. Coefficients *m* and *n* are constants, with *m* serving as the correction factor for the *Ra* and *n* representing the impact factor of FR.

Taking logarithms on both sides of equation (14), as depicted in equation (16).

$$\lg(Nu) = m\lg(Ra) + \lg(C) + n\lg\left(\frac{H}{\delta}\right) \tag{16}$$

According to equation (16), the linear correlation between *LgNu* and *LgRa* is evident under logarithmic scaling, with the coefficients *C*, *m*, and *n* derived through linear fitting of *LgNu* and *LgRa*. From a mathematical standpoint, the coefficient m signifies the slope of the fitted curve, while *C* and *n* are calculated from the intercept of the fitted curve.

As depicted in Figure 2, the red dashed double-line data represents the Nu number derived from the equation from Reference[13], depicted in equation (17).

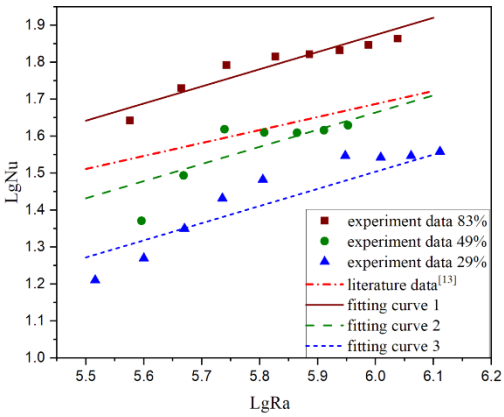

Figure 2. Fitting curve of Nu under 3.5 W/m^2 HLR.

$$Nu = 0.83569Ra^{m0.23207} \tag{17}$$

After modification, which employs three sets of fitting curves for each case, the accuracy of the Nu number calculation is substantially improved, allowing the simulation model to correctly simulate the effect of different FRs on the condensation heat transfer. Based on fitting curves 1-3, *m* can be determined to be 0.464.

Likewise, linear fits were performed for different HLR conditions to calculate the coefficient *C* and *n*. The rest coefficients are derived as follows: C=0.169, n=1.12.

Equations (18) and (19) shows the novel correlation equation for the Nu number:

$$Nu = CRa^{0.464}(\frac{H}{\delta})^{1.12} \tag{18}$$

$$C = \begin{cases} 0.169, \ HLR = 3.50W/m^2 \\ 0.234, \ HLR = 2.00W/m^2 \\ 0.557, \ HLR = 0.35W/m^2 \end{cases} \tag{19}$$

2.3 Experimental validation of the model

The utilization of the modified thermal stratified model for simulating the K-site experimental conditions, under 3.5 W/m^2 HLR and different FRs, exhibited a notable level of congruency with the experimental data. A comparative analysis between the modified model and the original one revealed that the former displayed a superior degree of alignment with the K-site experimental outcomes, as depicted in Figure 3. This improvement in correspondence can be attributed to the extended vapor-wall interface area within the modified model. As the FR diminished, the surface area of the vapor-wall interface increased, thereby accentuating the distinction between condensation heat transfer and natural convection heat transfer mechanisms. The implementation of the modified model resulted in a pressure error of less than 3% when benchmarked against the K-site experimental data, indicating a significant enhancement in accuracy and an improved trend in the calculated results. Consequently, the calculations validate the effectiveness and precision of the modified Nu correlation equation.

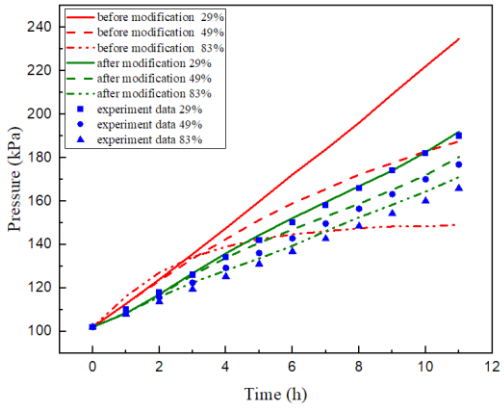

Figure 3. Effects of model modification on the LH_2 storage pressure.

3. SIMUALTION STUDY OF SLOSHING EXCITATION ON LH_2 STORAGE

On the basis of the modified thermal stratified model established in this paper, Saturn IBAS-203 LH_2 storage tank is selected to establish the liquid hydrogen sloshing storage model for the comparison and verification of simulation results[14]. The Saturn IBAS-203 LH_2 storage tank geometry is shown in Figure 4a.

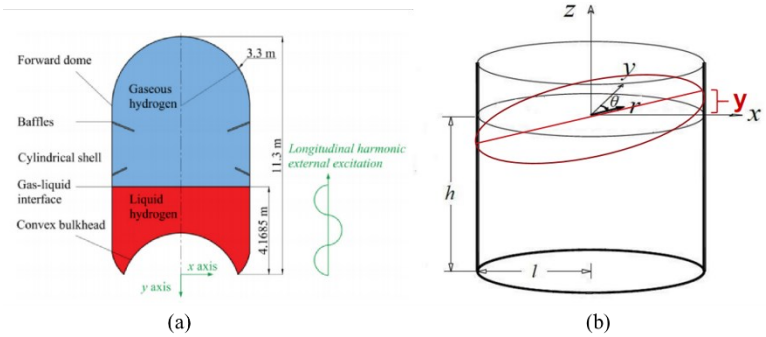

Figure 4. Liquid hydrogen sloshing tank model. (a): Saturn IBAS-203 LH_2 storage tank geometry; (b): sloshing excitation.

The effect of sloshing on the heat transfer of the tank is mainly reflected in the change of heat transfer area. The liquid surface area will change periodically with the sloshing. In this study, the simulation of sloshing conditions during sea transport is achieved by applying sine and cosine function excitation to the liquid level height at the edge of the tank, as showed in equation (20).

$$y = Asin(2\pi ft) \tag{20}$$

y is the displacement of the sloshing excitation; A is the amplitude of the sloshing excitation; and f is the excitation frequency.

For the estimation of Re and Pr numbers inside the storage tank, the convective heat transfer coefficient equation is shown as equation (21)[15].

$$Re = 0.921(RePr)^{0.5} \tag{21}$$

Referring to the Saturn IBAS-203 LH_2 storage tank experimental data, set the initial pressure as 85 kPa, initial temperature as 20 K. The sloshing frequency is fixed at 1 Hz, while the sloshing amplitude is respectively set at 0.01 m, 0.05 m and 0.1 m.

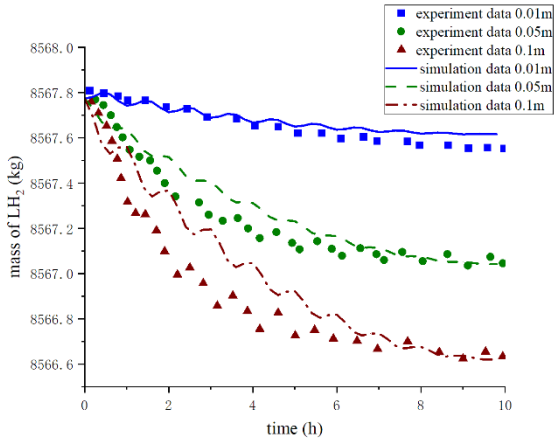

Figure 5. The impact of sloshing on LH_2 storage and transportation.

As depicted in Figure 5, the simulation results of three conditions are in good agreement with the overall trend of the experiment data, and the error range is within 8%. It can be seen that the accuracy of the established liquid hydrogen sloshing storage model is within the tolerance range, and the simulation results are correct and reliable, which can better simulate and calculate the evaporation characteristics of the actual tank under sloshing conditions. The sloshing model can be used for the subsequent further research and analysis of the effects of sloshing excitation factors on liquid hydrogen storage.

4. CONCLUSION

This study investigates the thermodynamic characteristics of LH_2 cryogenic storage with the utilization of the modified thermal stratified model.

(1) To adapt to different LH_2 storage conditions, a modified thermal stratified model has been proposed. This model includes a modified correlation equation for the Nu number, focusing on condensation heat transfer. The modified Nu number equation contains several heat transfer coefficients, taking FR and HLR into account.

(2) The modified model is used to simulation the K-site storage of LH_2. Modelling of thermal stratification and condensation heat transfer significantly improves simulation accuracy. The modified model demonstrates an error restricted within 3%.

(3) Sloshing excitation is applied to the modified model. Error of the calculated results under sloshing excitation are within 8% compared with the Saturn IBAS-203 LH_2 storage experiment data.

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