Mathematical Modelling of Thermal Stratification in a Cryogenic Propellant Tank

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Abstract. Cryogenic tanks used for space applications are filled with sub-cooled cryogenic propellants, whose liquid-vapor interface remains undisturbed for long periods of time prior to launch. During this period, substantial amount of heat leaks into the tank from external sources such as solar and ambient convective fluxes, even though the tank is well insulated. This results in thermal stratification near the liquid vapour interface. A transient, two-phase, thermodynamic model of stratification in a cryogenic tank is developed, considering propellant boundary layer flow due to natural convection close to tank wall. Continuity, momentum, energy and mass transfer equations are solved using finite difference-based formulations of SINDA/FLUINT simulator. The analytical model is validated with test results reported in literature. Subsequently, studies are carried out to investigate the effect of liquid sub-cooling in propellant tank on stratified mass and liquid temperature profile. The study shows that sub-cooling of cryogenic tank leads to significant increase in stratified mass.

1. Introduction
A tank filled with cryogenic fluid is generally used in various engineering applications like rocket propellant storage, liquefied fuel storage and air separation system, such as liquid nitrogen (LN2), liquid hydrogen (LH2), liquid oxygen (LOX) and liquefied natural gas (LNG). These storage tanks are subject to external thermal loads when placed in the ambient, which raises the temperature of the fluid filled in the tank. In cryogenic launch vehicles, the fluid inside the propellant tank passes through the turbo pump before enters to combustion chamber. When pump inlet fluid temperature becomes higher than cavitation limit or ‘limiting temperature’, pump cavitation can occur, which will lead to combustion instability of cryogenic engine. Thus, the liquid inside the tank with higher temperature than the limiting value cannot be used for combustion, making it unusable propellant mass, called stratified mass. Stratified propellant mass is considered as payload penalty for the launch vehicle. To minimise the stratified mass, highly efficient insulation are used to reduce the heat in-leak into the tank. The mass of stratified liquid is a critical parameter for design of propellant tank and efficiency of launch vehicle systems. The accurate prediction of evolution of stratified mass with time in propellant tank is very important to design the cryogenic fluid system used in launch vehicle.

Several studies on stratification in cryogenic tank have been reported in the literature. Tatom et al. [1] reported an experimental study on thermal stratification of LH2 in rocket propellants for bottom heating and side heating and suggested that bottom heating, if applied properly, may cause a considerable reduction in the stratification. Evans et al. [2] carried out the experimental study on transient effects of natural convection inside cylindrical enclosures. The study concluded that the heat transfer coefficient approaches a quasi-steady state. The numerical study carried out by Lin and Hasan [3] solved the steady state conservation equations for an axi-symmetric cylindrical enclosure and
showed the effect of system parameters like Rayleigh number, prandtl number and wall heat flux on the characteristics of thermal stratification in case of uniform sidewall heat flux. Tanyun et al. [4] performed a numerical simulation of the thermal stratification of LH2 inside the storage tank. The formulation was based on the vorticity-streamfunction method, and the surface temperature obtained was correlated with the heat flux, fluid level and time. Panzarella et al. [5] carried out numerical simulation by introducing the effect of thermal stratification in the liquid region to the effect of pressure rise on various heating conditions. A numerical simulation for the convection currents in a rectangular enclosure was reported by Sun et al. [6] with volume base finite difference method. The study concluded with the effect of increase in baffle size to the convection currents. The work was limited to laminar region only. Barsi et al. [7] carried out numerical calculation and experiment of the self-pressurization in LH2 tank. They considered the heat in the liquid is used for evaporation of liquid at interface and modeled the ullage gas phase as a lump in respect of energy and mass. Ganguli et al. [8] performed an experimental and numerical study for stratification in fluid container taking working fluid as water. The study showed the higher stratification in less time with higher Rayleigh number. The study was restricted to single phase CFD simulation. An experimental and numerical study was carried out by Ludwig et al. [9] which showed the effect of sloshing on variation in tank pressure and liquid stratification.

Though plenty of literatures on stratification are available, very few of them investigates the stratification process for different sub-cooled condition of liquid. Since cryogenic launch vehicle systems require sub-cooled propellant at inlet of its turbo-pump, determination of stratified mass under different sub-cooling levels is essential. In the present study stratification phenomenon is investigated with LN2 as working fluid. A two phase thermal stratification model is developed considering the flow of natural convection currents in liquid due to heat in-leak. The numerical model is validated with the experiment data reported in literature [9] and rise in liquid temperature with time is compared with the literature data. The model is used to calculate the variation in evolution of stratified mass in different pressurisation of 2 bar, 3 bar and 4 bar.

2. Thermal Stratification in Cryogenic Tank
Heat in-leak into the tank results in temperature difference, and consequently density difference, between the liquid adjacent to the wall and bulk liquid. This forces the liquid adjacent to wall to flow upward under the action of buoyancy, as shown in figure 1. This process results in accumulation of hot liquid near to the liquid-vapor interface and leads to a temperature gradient along vertical axis of the tank. This phenomenon is called ‘thermal stratification’. The accumulated liquid near the interface is called ‘thermal stratified liquid’.

![Figure 1. Schematic of liquid stratification phenomenon in a cryogenic tank](image)

The total propellant mass inside the tank having temperature, higher than the temperature limit for pump cavitation, is termed as stratified mass. Though pump cavitation is not a direct concern in the
present model on stratification studies using LN2, a limiting temperature less than the lowest saturation temperature out of three pressurization cases (2bar and 83.63K) is chosen. The aim of choosing a limiting temperature is to study the effect of sub-cooling on evolution of stratified mass in a cryogenic tank. Thus, an arbitrary value of 80K (less than 83.63K) is taken as the limiting temperature to define the stratified mass of LN2 in the present study.

3. Mathematical Formulation

3.1 Conservation of Mass

The net mass flow from a fluid node is equated to the rate of change of mass in the control volume as shown below.

\[ \dot{m}_{up} - \dot{m}_{down} = \frac{dm}{dt} \]  \hspace{1cm} (1)

Conservation of mass is ensured at each fluid node during each time-step of transient simulation.

3.2 Conservation of Momentum

The governing equation for flow connectors is simply a complex form of Newton’s second law. The momentum conservation equation for a fluid connector is written as below:

\[ \frac{dm}{dt} = \frac{A_c}{L} \left[ (P_{up} - P_{down}) - f \cdot \frac{1}{2 \rho A_c^2} \left( \frac{dm}{dt} \right)^2 \right] \]  \hspace{1cm} (2)

Viscous coefficient \( f \) is calculated using Churchill formulation denoted in Incropera and Dewitt [10]. Fluid nodes are connected by fluid connectors on which momentum conservation is imposed.

3.3 Conservation of Energy

The energy conservation equation is expressed on the basis of the first law of thermodynamics. The rate of increase of internal energy in the control volume is equal to the difference between the rate of energy transport into the control volume and the rate of energy transport from the control volume. The energy conservation equation based on enthalpy can thus be written as below:

\[ \frac{dU}{dt} = (H_{up} \dot{m}_{up} - H_{down} \dot{m}_{down}) + \frac{dQ}{dt} \]  \hspace{1cm} (3)

Where:

\[ \frac{dQ}{dt} = hA(T_w - T_{lu}) \]  \hspace{1cm} (4)

3.4 Equation of State

Thermodynamic variables at a particular fluid node are calculated using real fluid state equation with compressibility factor (Z), being an input from NIST database [11].

\[ P = Z \cdot \frac{m}{V} \cdot \frac{R}{M} \cdot T \]  \hspace{1cm} (5)

3.5 Boundary layer thickness and velocity

The upward flow in boundary layer is determined by using the formulations developed by Tsuji et al. [12].

\[ v_b = 0.210Gr_x^{0.125} [g\beta(T_w - T_l) u_l]^1/3 \]  \hspace{1cm} (6)

\[ \delta = \frac{3.04uGr_x^{0.250}}{[g\beta(T_w - T_l) u_l]^{1/3}} \]  \hspace{1cm} (7)

where
3.6 Heat and mass transfer

The estimation of different modes of heat transfer into the cryogenic tank and heat and mass transfer across liquid-vapor interface is considered in the model, as shown in figure 2.

\[ Gr_x = \frac{g \beta (T_w - T_l) x^3}{v_l^2} \]  

(8)

3.6.1 Heat and mass transfer across interface

The model is used to study the evolution of liquid temperature inside the tank considering conduction, convection, heat and mass transfer across the interface. Heat, from ullage to liquid, transfers through interface. Heat transfer from the interface to liquid is expressed as:

\[ \frac{d Q_{u-int}}{dt} = h_{u-int} A_{int} \Delta T_{u-int} \]  

(9)

Similarly, heat transfer from the interface to liquid is shown as

\[ \frac{d Q_{int-l}}{dt} = h_{int-l} A_{int} \Delta T_{int-l} \]  

(10)

It has been assumed that the heat transfer across liquid vapor interface is due to natural convection. Convective heat transfer from interface to liquid is expressed by the correlations [10] as below.

\[ h_{int-l} = 0.27 \frac{k_l}{D} Ra_{int-l}^{0.25} \]  

(11)

where:

\[ Ra_{int-l} = Gr \cdot Pr = \frac{g \beta (T_{int} - T_l) D^3}{v_l^2} \cdot \frac{\mu_l C_p_l}{k_l} \]  

(12)

Convective heat transfer from ullage gas to interface is shown below.

\[ h_{u-int} = 0.54 \frac{k_u}{D} Ra_{u-int}^{0.25} \]  

(13)

where:

\[ Ra_{u-int} = Gr \cdot Pr = \frac{g \beta (T_u - T_{int}) D^3}{v_u^2} \cdot \frac{\mu_u C_p_u}{k_u} \]  

(14)
Since tank ullage contains only single species of gaseous nitrogen, interface will be at saturation temperature which will result in boiling or condensation at liquid-vapor interface. The difference in heat transfer from ullage gas to interface and interface to liquid will be utilized for boiling or condensation at interface. The heat required to boil or condense the fluid at interface is expressed by the equation (15).

\[
\frac{dQ_{\text{boil}}}{dt} = \dot{m} \cdot h_f \beta
\]  

The mass transfer across the interface is estimated based on the energy conservation at interface as below.

\[
\frac{dQ_{\text{u-int}}}{dt} + \frac{dQ_{\text{boil}}}{dt} = \frac{dQ_{\text{int-l}}}{dt}
\]

\[
\dot{m} \text{ is positive for condensation and negative for vaporization of the liquid and is expressed as:}
\]

\[
\frac{dm}{dt} = \frac{[h_{\text{int-l}}A_{\text{int}}\Delta T_{\text{int-l}} - h_{\text{u-int}}A_{\text{int}}\Delta T_{\text{u-int}}]}{h_f \beta}
\]  

3.6.2 Heat transfer from wall to fluid: Tank is pressurized with gaseous nitrogen and during pressurisation mixed convection heat transfer occurs in the tank ullage as a combination of forced convection and natural convection. Heat transfer from tank wall to ullage gas during pressurisation is determined by using the formulation [12] described as below.

\[
\frac{dQ_{\text{w-u}}}{dt} = hA_w \Delta T_{\text{w-u}}
\]

where:

\[
h = N_u_x k_u/x
\]

\[
N_u_x = 0.56 R_e_d^{0.67} + 0.104 R_a_x^{0.352}
\]

\[
R_e_d = \frac{\rho_u v_p d}{\mu_u}
\]

\[
R_a_x = G_r_x \cdot P_r = \frac{g \beta (T_w - T_u)x^3}{v_u^2} \cdot \frac{\mu_u C_p u}{k_u}
\]

Heat transfer from wall to liquid is calculated using natural convection formulation reported by Tsuji et. al [13] and is described below.

\[
\frac{dQ_{\text{w-l}}}{dt} = hA_w \Delta T_{\text{w-l}}
\]

where:

\[
h = 0.110 \left[ \frac{g \beta (T_w - T_l)x^3}{v_l^2} \right]^{0.25} \cdot k_l/x
\]

3.6.3 Heat transfer from ambient: Coefficient of convective heat transfer on tank surface is calculated by Churchill and Bernstein formulation from Incropera and Dewitt [10]. Heat transfer from outside is expressed as below.

\[
\frac{dQ_{\text{a-s}}}{dt} = hA_s \Delta T_{\text{a-s}}
\]

where:
To calculate heat transfer from ambient to the tank wall, the two dimensional axis-symmetric transient heat conduction equation is solved considering the variation of thermal conductivity of the insulation with temperature. Transient heat conduction equation is given by following expression considering $k$ as a function of temperature:

$$
(0.3 + \frac{0.62 \cdot Re^{0.5} \cdot Pr^{0.33}}{1 + \left(\frac{0.4 \cdot Pr}{28200}\right)^{2/3}}) \cdot \left(1 + \left(\frac{Re}{28200}\right)^{5/8}\right) \cdot \frac{k_a}{D} 
$$

Re = \frac{\rho_a v_a D}{\mu_a} \tag{27}

$$
Pr = \frac{\mu_a C_p}{k_a} \tag{28}
$$

To calculate heat transfer from ambient to the tank wall, the two dimensional axis-symmetric transient heat conduction equation is solved considering the variation of thermal conductivity of the insulation with temperature. Transient heat conduction equation is given by following expression considering $k$ as a function of temperature:

$$
\left[\frac{1}{r} \frac{\partial}{\partial r} \left( r k(T) \frac{\partial T}{\partial r} \right) + \frac{\partial}{\partial x} \left( k(T) \frac{\partial T}{\partial x} \right)\right]_\text{sol} = \rho C_p \cdot \frac{\partial T}{\partial t} \bigg|_\text{sol} \tag{29}
$$

The comprehensive flow and thermal model considers real fluid properties from NIST database [11] to calculate the thermodynamic variables mentioned in all the above equations. All the above mentioned heat transfer modes representing the mathematical thermal model have been incorporated in the current study.

### 3.7 Solution Methodology

A two-phase thermal model is developed using SINDA/FLUINT simulator which uses finite difference based flow and thermal network formulations to solve the governing equations. Figure 3 shows a flow network of convection current inside the liquid and heat transfer from tank wall to the boundary layer. The liquid domain is discretized axially and boundary layer flow at each boundary node, depicted in figure 3, is calculated from the equations 6-8. Based on the mass conservation equation, the difference in subsequent upward flow rate at a boundary node is balanced by the flow from core to boundary layer.

**Figure 3.** Schematic of liquid stratification model for a cryogenic tank

Figure 4 shows the solution flow chart representing the dependence of each parameter on different thermodynamic equations used in the present model. Major resultant parameters from the simulation
are tank wall temperature distribution, fluid thermodynamic parameters (pressure, temperature, and
density), fluid connector mass flow rates and interface mass transfer rate. These parameters are inputs
as well as outputs for different thermodynamic equations, as discussed in earlier section. Iterations for
solution convergence are carried out for each time step of simulation, thereby forming a closed
relation between the each parameter.

The numerical simulation is carried out considering the following assumptions:

i. The turbulence in liquid and ullage gas near to the interface, due to tank pressurisation is assumed
to be very less. Heat transfer from ullage to interface and interface to liquid is considered to be
natural convection over a flat plate.

ii. The flow of natural convection currents in ullage gas is not modelled in the study. The gas
domain is discretised in axial direction only.

iii. The limiting temperature to define stratified mass is assumed to be 80K for the parametric study
at different sub-cooling levels.

iv. The tank is considered to be exposed to ambient of 5m/s wind velocity for the parametric study.

4. Results and Discussion
A thermal stratification model is developed to determine the growth of stratified layers with time. The
model is validated with the experimental results [9]. Subsequently the study is carried out to
investigate the effect of different liquid sub-cooling conditions on tank stratified mass.

4.1. Model Validation
Ludwig et al. [9] carried out an experiment on pressurization of cryogenic tank filled with LN2 at an
initial temperature and pressure of 77.7K and 1.06bar. The dimensions of the cylindrical tank taken for
the experiment are shown in Table 1.

Tank is pressurized from 1.06bar to 3.0bar for the duration of 52s using gaseous nitrogen. After 52s,
tank pressure decreases and reaches 2.46bar which is very close to the measured pressure as can be
seen from the figure 5. Figure 6 describes the liquid temperature along the tank height and its variation
with time. It can be seen from the figure 5 and 6 that initially liquid is at saturation condition of 77.7K.
Subsequently, tank is pressurised from 1.06bar to 3.0bar in 52.0s and interface temperature increases
from 77.7K to 87.9K which is the saturation temperature corresponding to the tank pressure of 3bar.
The pressure reduction after pressurization results in reduction in interface temperature to 85.7K at
200s. Figure 7 shows the rise in liquid temperature at the height of 0.445m and 0.450m from the tank bottom which illustrates a well corroboration with experimental data. It is to be noted from the figures 5, 6 and 7 that simulated results are well validated with the experimental data.

**Table 1.** Details of experimental tank dimensions and initial parameters used by Ludwig et al. [9]

| Parameter                        | Value  |
|----------------------------------|--------|
| Tank height                      | 0.65m  |
| Liquid height                    | 0.455m |
| Tank inner diameter              | 0.296m |
| Working fluid                    | Nitrogen |
| Pressurant gas                   | Nitrogen |
| Pressurant gas temperature       | 294K   |
| Initial liquid temperature       | 77.7K  |
| Initial tank Pressure            | 1.06bar |

**Figure 5.** Comparison of simulated tank pressure with measurement

**Figure 6.** Comparison of simulated liquid temperature along the tank height with measurements
4.2. Effect of liquid sub-cooling

A study is also carried out to determine the effect of liquid sub-cooling on growth of stratified layers in liquid. The tank is pressurized up to different levels with gaseous Nitrogen of 100K temperature. The tank used is of the same dimensions as of the tank used for the validation. The tank is filled with LN2 up to the height of 0.455m. Tank is insulated with PUIR foam and insulation thickness is 0.030m. The tank is exposed to ambient temperature of 300K and wind speed of 5.0m/s. Initially, liquid nitrogen is filled at the temperature of 77.7K and tank pressure is 1.0 bar. Three cases are analysed in which tank ullage is pressurized up to different levels of 2.0bar, 3.0 bar and 4.0bar to achieve the different sub-cooling conditions in the tank. The tank ullage is pressurised for the duration of 200s and pressure is kept constant after the pressurization to the respective pressurization level. It is to be noted that the temperature of liquid-vapor interface is at saturation temperatures of 83.6K, 87.9K and 91.2K for the different pressurization cases of 2.0bar, 3.0bar and 4.0bar respectively.

Figure 8. Liquid temperature along the tank height with tank pressure of 2bar

Figure 9. Liquid temperature along the tank height with tank pressure of 3bar
The thermo-physical properties of nitrogen are taken from NIST database [11]. As the pressure increases, the kinematic viscosity of liquid nitrogen increases. It can be seen from the equations (6)-(8) that the boundary layer thickness and velocity will be higher for higher pressure which will increase the stratified mass. In addition to this, the interface temperature (saturation temperature) will be higher for higher sub-cooling case which results in higher heat transfer rate from interface to liquid as seen in equation 10. This leads to higher temperature rise in liquid and consequent increase in stratified mass.

The evolution of liquid temperature along the tank height for the different sub-cooled cases is shown in figures 8, 9 and 10. The figures show the liquid temperatures at the different time instants of initial condition, 200s 300s, 400s and 500s. It can be seen from the figures that the temperature lines are closer in the case of 2.0bar pressurisation compared to 4.0bar pressurisation which illustrates the less stratification in the case of 2.0bar pressurisation compared to 4.0bar pressurisation.

Figure 10. Liquid temperature along the tank height with tank pressure of 4bar

Figure 11. Liquid temperature at 5mm below the interface for different sub-cooling pressures
The growth in liquid temperature at 5mm below the liquid-vapor interface is depicted in Figure 11. It can be seen that the liquid temperature, at 500s, reaches the temperature of 81.4K, 83.8K and 85.6K for the sub-cooling at 2.0bar, 3.0bar and 4.0bar respectively. In the current study stratified mass is the mass of LN2 which is having temperature more than 80K.

Figure 12 shows the evolution of stratified mass in liquid, after the tank pressurisation (after 200s), during which the tank pressure is kept constant at different values of 2bar, 3bar and 4bar. It can be illustrated from the figure 12 that the rate of increase in stratified mass is higher for the case of higher liquid sub-cooling (higher pressure). The stratified mass after 300s of pressurization (at 500s in figure 12) is 0.74kg, 1.23kg and 1.93kg for the case of sub-cooling at 2.0bar, 3.0bar and 4.0bar, respectively.

5. Conclusion
A two-phase thermal stratification model is developed to simulate the increase of thermal stratified mass in a cryogenic tank. Initially, the model is well validated with the experimental data available in the literature. Subsequently parametric studies are carried out to bring out the influence of different liquid sub-cooling on evolution of stratified layers and stratified mass in the tank. The study shows that the growth of thermal stratification is faster in a tank containing cryogenic liquid at higher levels of sub-cooling.

It is inferred from the study that stratified mass will be higher in the tank of higher liquid sub-cooling and higher pressure. Based on the study, it is suggested that cryogenic fluid system should be designed at minimum possible pressure to reduce the build-up of stratified mass and amount of unusable propellant in the cryogenic tank, thereby achieving payload advantage for the launch vehicle.

6. Nomenclature

| Symbol | Definition                   |
|--------|------------------------------|
| $A$    | Surface area (m$^2$)         |
| $C_p$  | Specific heat (J/Kg-K)       |
| $D$    | Tank diameter (m)            |
| $d$    | Pressurising port diameter (m)|
| $g$    | Gravity (m/s$^2$)            |
| $H$    | Enthalpy (J/kg)              |
| $h$    | Heat transfer coefficient (W/m$^2$-K)|
\( h_{lg} \) Latent heat of vaporization (J/K)
\( k \) Thermal conductivity (W/m-K)
\( m \) Resident mass (Kg)
\( \dot{m} \) Mass flow rate (kg/s)
\( M \) Molecular Weight (kg/mole)
\( Z \) Compressibility factor
\( Nu \) Nusselt number
\( Pr \) Prandtl number
\( Gr \) Grashof number
\( f \) Viscous Coefficient
\( P \) Pressure (Pa)
\( Q \) Heat (J)
\( Ra \) Rayleigh number
\( Re \) Reynolds number
\( T \) Temperature (K)
\( t \) Time (s)
\( A_c \) Cross sectional area of fluid connector (m\(^2\))
\( L \) Length of fluid connector (m)
\( \nu \) Fluid velocity (m/s)
\( x \) Tank height (m)
\( U \) Internal energy (J)
\( V \) Resident volume (m\(^3\))
\( r \) Tank radius (m)

**Greek Symbols**
\( \beta \) Coefficient of thermal expansion (K\(^{-1}\))
\( \delta \) Boundary layer thickness (m)
\( \mu \) Dynamic viscosity (Pa-s)
\( \nu \) Kinematic viscosity (m\(^2\)/s)
\( \rho \) Density (kg/m\(^3\))
\( \sigma \) Surface tension (Kg/s\(^2\))

**Subscripts**
\( a \) Air
\( p \) Pressurant port
\( b \) Boundary layer
\( int \) Liquid-vapor interface
\( l \) Liquid
\( s \) Tank outer surface
\( x \) Tank height
\( u \) Ullage gas
\( boil \) Boiling
\( cond \) Conduction
\( up \) Upstream
\( down \) Downstream
\( sol \) Solid
\( w \) Tank inner surface
7. References

[1] Tatom J W, Brown W H, Knight L H, and Coxe E F, 1964 Analysis of Thermal Stratification of LH2 in Rocket Propellant Tank, Advances in Cryogenic Engineering, vol. 9, pp. 265–272.

[2] Evans L B, and Stefany N E, 1966 An Experimental Study of Transient Heat Transfer To Liquids In Cylindrical Enclosures, Chem. Eng. Prog. Symposium Ser. Vol. 62, No. 64, pp 209-215.

[3] Lin C S, Hasan M M, 1990 Numerical Investigation of The Thermal Stratification in Cryogenic Tanks Subjected to Wall Heat Flux, AIAA-90-2375, NASA Tech. Memo. 103194.

[4] Tanyun Z, Zhongpin H, and Li S, 1996 Numerical Simulation of Thermal Stratification in Liquid Hydrogen, Advances in Cryogenic Engineering, vol. 41, pp. 155–161.

[5] Panzarella C H, Kassemi M, 2003 On The Validity of Purely Thermodynamic Descriptions of Two Phase Cryogenic Fluid Storage Tanks, J. Fluid. Mech., Vol. 484, pp. 41-68.

[6] Sun Y S, Emery A F, 1997 Effects of Wall Conduction, Internal Heat Sources and an Internal Baffle on Natural Convection Heat Transfer in a Rectangular Enclosure. Int. J. Heat Mass Transfer 40, 915–929.

[7] Barsi S, and Kassemi M, 2008, Numerical and Experimental Comparisons of the Self pressurization behavior of an LH2 tank in Normal Gravity, Cryogenics Vol. 48, pp. 122-129.

[8] Ganguli A A, Pandit A B, Joshi J B, Vijayan P K, 2011. Hydrodynamic and Heat Transfer Characteristics of a Centrally Heated Cylindrical Enclosure: CFD Simulations and Experimental Measurements. Chem.Eng. Res. Des. 89, 2024–2037.

[9] Ludwig C, Dreyer M E, and Hopfinger E J., 2013 Pressure Variations in a Cryogenic Liquid Storage Tank Subjected to Periodic Excitations. International Journal of Heat and Mass Transfer 66: 223-234.

[10] Incropera F P, and Dewitt D P, 1985, Fundamentals of Heat and Mass Transfer, (John Wiley & Sons, Inc.), Chap. 7.

[11] <webbook.nist.gov/chemistry/fluid/>

[12] Woodfield P L, Monde M and Mitsutake Y, 2006 Measurement of Averaged Heat Transfer Coefficient in High Pressure Vessel during Charging with Hydrogen, Nitrogen and Argon Gas, Journal of Thermal Science and Technology, No. 06-0272, DOI 10.1299/jtst.2.180.

[13] Tsuji, T. and Nagano, Y., 1989 Velocity and Temperature Measurements in a Natural Convection Boundary Layer along a Vertical Flat Plate, Experimental Thermal and Fluid Science, 2(2), pp.208-215.