Theoretical and Numerical Analysis of Freezing Risk During LNG Evaporation Process

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Abstract: The liquid natural gas (LNG) boiling process concerns most LNG applications due to a need for regasification. Depending on the pressure, the equilibrium temperature of LNG is 112–160 K. The low boiling temperature of LNG makes the vaporization process challenging because of a large temperature difference between the heating medium and LNG. A significant risk included in the regasification process is related to the possibility of solid phase formation (freezing of the heating fluid). A solid phase formation can lead to an increase in pressure loss, deterioration in heat transfer, or even to the destruction of the heat exchanger. This prompts the need for a better understanding of the heat transfer during the regasification process to help avoid a solid phase formation. The present research is focused on the investigation of the mutual interactions between several parameters, which play a significant role in the regasification process. The research is based on a zero-dimensional (0D) model, which was validated through the comparison with a state-of-the-art Computational Fluid Dynamics (CFD) model. This made fast calculations and the study of the risk of freezing for a wide range of parameter space possible, including the LNG boiling regime. The boiling regime of LNG was shown to be a key factor in determining the risk of freezing.

Keywords: LNG vaporization; regasification; freezing risk; heat transfer

1. Introduction

Natural gas (NG) is recognized as the cleanest available fossil fuel. The combustion of NG results in a much lower emission of pollutants compared to any other type of fuel (coal, lignite, and crude oil-based fuels). The specific volume of liquid natural gas (LNG) is approximately 600 times lower than it is in its gaseous state. This makes storage much more effective and permits long-distance transportation.

NG and LNG have become increasingly important on the energy market and in energy conversion policies. It is one of the fuels with the fastest growth in consumption over the past few years. The consumption rate is also predicted to increase by an average of 1.3% per year, reaching 5.2 trillion m³ in 2040 [1–3].

The increasing demand for a reduction in greenhouse gas emissions continues to generate a growing interest in the usage of LNG as a possible fuel for power generation, industry and transportation [4,5].

One application of LNG for marine transportation may be a potential answer to the emission-related demands of The International Marine Organization. The organization has established Emission Controlled Areas (ECA) where the emission of nitrogen oxides, sulfur oxides, and particulates must be reduced. In July 2017, Marine Environment Protection Committee 71 adopted amendments to regulation 13 of the MARPOL Annex VI, introducing new limits on NOx emissions from ship exhausts for the ECA area, including the Baltic Sea and the North Sea [6]. These emissions can be substantially...
reduced by using LNG, which is recognized as an environmentally friendly and ecological fuel for both present and future methods of ship propulsion [7–9].

The last stage of the LNG technological cycle is vaporization and a temperature increase to ambient levels. This is a key issue in LNG import terminals and the fuel system of LNG fed vehicles. Properly designed LNG vaporization can improve the efficiency of an overall cycle by using the cold exergy of LNG [10–13]. This can be a source of cooling power [14–16], used in the air separation process [17,18], in the freeze desalination process [19–23], or to improve the capacity of Adsorbed Natural Gas (ANG) tanks [2].

All the abovementioned technologies stress the importance of an efficient and properly conducted evaporation process for LNG. Conventional vaporizers mainly use ambient environments, available waste heat, or heat from NG combustion as heat sources. This results in high temperature differences between LNG and the heating medium and can potentially cause a very intense heat transfer.

Depending on pressure, the equilibrium temperature of LNG is 112–160 K. The low boiling temperature of LNG makes the vaporization process prone to the risk of a solid phase formation in the heating fluid. For a range of important applications, the heating fluid is water or a water–glycol mixture, characterized by freezing temperatures that are considerably higher than the boiling temperature of LNG. Solid phase formation can lead to an increase in hydraulic pressure losses, deterioration in heat transfer, or even to the destruction of the heat exchanger and any accompanying device.

It prompts the need for better understanding and control of heat transfer as related to LNG regasification, to avoid solid phase formation. The conducted research presents a theoretical and numerical analysis of heat transfer during the vaporization process of LNG with a particular focus on the risk of a heating medium freezing. The authors assume that overall heat transfer intensity is a function of the boiling regime of LNG, wall and ice conduction, and heating fluid convection. The most important one, from the perspective of freezing, is related to the LNG boiling regime. It depends strictly on the temperature difference between the boiling LNG and the heating fluid. The current research analyzes the LNG regasification process as a function of several parameters and it shows that the LNG boiling heat transfer coefficient is one of the main factors concerning the risk of freezing and needs to be controlled.

The presented research is based on a zero-dimensional (0D) calculation methodology of steady heat transfer between the heating medium and boiling LNG. Its development was prompted by the need for fast and reliable computations to investigate a wide range of parameter space. The proposed 0D model was checked through comparison with a state-of-the-art numerical model based on Navier–Stokes equations including heat transfer and solidification.

2. Theoretical Consideration and Zero-Dimensional Model

The simplified calculations from the presented research are based on a zero-dimensional (0D) steady heat transfer between the heating medium and boiling LNG, as shown in Figure 1.

![Figure 1. Sketch of a heat transfer direction in the considered problem.](image-url)
For the purposes of the investigation, the heating medium was assumed to be water. The heat flux density considered in Figure 1 can be expressed as:

$$\dot{q} = OHTC \cdot \Delta T_{w-LNG}$$

(1)

where $\Delta T_{w-LNG}$ is the temperature difference between water and LNG. The overall heat transfer coefficient (OHTC) depends on: (i) water convection, (ii) conduction through the partition and a possible layer of ice, (iii) and the boiling of LNG:

$$OHTC = \frac{1}{\frac{1}{h_w} + \frac{k_{wall}}{g_{wall}} + \frac{k_{ice}}{g_{ice}} + \frac{1}{r_LNG}}$$

(2)

The heat flux density can be expressed for each of heat transfer mechanism separately with corresponding temperature difference:

$$\dot{q} = h_w \Delta T_{w-wall1}$$

(3)

$$\dot{q} = \frac{k_{wall}}{g_{wall}} \Delta T_{wall1-wall2}$$

(4)

$$\dot{q} = h_{LNG} \Delta T_{wall2-LNG}$$

(5)

In case of a solidification of water a heat flux through ice needs to be also considered:

$$\dot{q} = \frac{k_{ice}}{g_{ice}} \Delta T_{if-wall1}$$

(6)

where $T_{if}$ represents the temperature at the wall–ice interface. Additionally, the Equation (3) should be substituted with:

$$\dot{q} = h_w \Delta T_{w-if}$$

(7)

The convection heat transfer coefficient of water can be expressed as:

$$h_w = \frac{N_{u_w} k_w}{D_h}$$

(8)

where the water Nusselt number $N_{u_w}$ depends on a water flow regime [24] and for laminar flow ($Re < 2300$) can be expressed as follows:

$$N_{u_l} = 6 + \frac{0.065 (D / L) R_{e_w} Pr_w}{1 + 0.04 [(D / L) R_{e_w} Pr_w]^{2/3}}$$

(9)

where $Re_w$ and $Pr_w$ correspond to the water Reynolds and Prandtl numbers, respectively. In case of a turbulent flow, for $Re_w > 10,000$ it changes to:

$$N_{u_l} = 0.023 R_{e_w}^{0.8} Pr_w^{0.3}$$

(10)

It was assumed that for a transitional flow, $2300 < Re_w < 10,000$, $N_{u_w}$ can be extrapolated.

In the present research, a thin rectangular domain is considered. It can be seen as a simplified and basic geometrical feature of the heat exchanger, for such geometry $Re_w = D_h U / v$, where $D_h = 2D$ is the hydraulic diameter of the considered geometry and is equal to twice its width. The assumed width $D$ equals 4 mm.

The $Pr_{water}$ number can be calculated with respect to the water temperature using the following formula [25]:

$$Pr_w = -1.01 \cdot 10^{-4} T_w^3 + 9.314 \cdot 10^{-2} T_w^2 - 28.726 T_w + 2971.8$$

(11)
The material of the wall was assumed to be stainless steel with a thickness of 1 mm. Its heat conductivity was based on the average wall temperature according to the following formula:

\[
k_{\text{wall}} = -1.8 \cdot 10^{-9} T_{\text{wall,av}}^4 + 1.717 \cdot 10^{-6} T_{\text{wall,av}}^3 \\
-6.233 \cdot 10^{-4} T_{\text{wall,av}}^2 + 1.275 \cdot 10^{-1} T_{\text{wall,av}} + 1.178
\]  

(12)

The heat conductivity of the ice layer was calculated for the average temperature of the ice, following the formula from [26]:

\[
k_{\text{ice}} = -1.42 \cdot 10^{-2} T_{\text{ice}} + 6.058
\]

(13)

The thickness of the ice layer was calculated with an assumed temperature of 273.15 K at the ice-water interface.

A key issue in the considered analysis was the determination of the boiling heat transfer coefficient of LNG \( h_{\text{LNG}} \). It was determined based on a pool boiling experimental results reported by [27–29]. To determine the boiling heat transfer coefficient, this experimental data was gathered and compared in Figure 2.

![Figure 2](image_url)

**Figure 2.** Experimental data on LNG boiling HTC curves gathered from [27–29].

Furthermore, the current analysis assumes that the temperature difference, governing the boiling heat transfer coefficient (HTC) of LNG, ranges from 50 to 150 K. This corresponds with the film boiling regime. It is important to note that this situation varies significantly from typical HVAC applications, where temperature differences are much smaller, and a nucleate boiling regime is to be expected. Consequently, in HVAC applications, the goal is to maximize the boiling heat transfer. As will be shown later, this is not always the case for LNG regasification, because a very intense boiling heat transfer can cause the heating medium to freeze.

The equations describing the boiling HTC of LNG are, therefore, estimated based on film boiling experimental data provided by [28], see the Figure 3.
The boiling HTC is reported to be influenced by several factors such as: surface abrasiveness \cite{30}, flow regime \cite{24,31,32} and boiling surface orientation \cite{33–35}. Taking into account the additional influence of a possible change in the boiling HTC, the current analysis considers three distinct cases. The first one corresponds to experimental data for pressure below 0.2 MPa and can be approximated as:

\[
h_{\text{LNG},A} = (-2.017 \times 10^{-5} \Delta T^3 + 1.569 \times 10^{-2} \Delta T^2 - 4.244 \Delta T + 676.5)
\] (14)

The second and third cases represent the possible increase of the boiling HTC by 50\% and 100\% respectively. These three cases are further depicted as A, B, and C, and can be seen as solid lines in Figure 3.

![Figure 3](image)

**Figure 3.** Experimental data of the LNG boiling HTC provided in \cite{28}. for different pressures; line A: the polynomial approximation of the results below 0.2 MPa, lines B and C represents increase of HTC by 50\% and 100\% respectively.

The presently developed 0D calculation methodology makes it possible to determine the temperature differences for each of the considered heat transfer mechanisms for a given temperature boundary condition of LNG and water. Figure 4 presents a flow chart diagram of the calculation process.

![Figure 4](image)

**Figure 4.** Flow chart diagram presenting the developed 0D model.
It consists of two iteration loops. The first loop adjusts the LNG boiling HTC based on the calculated temperature difference $\Delta T_{\text{wall}-\text{LNG}}$ as long as required convergence $\epsilon$ is achieved. The second iteration increases the thickness of the ice layer $g_{\text{ice}}$ as long as the temperature at the water-ice interface is below freezing temperatures. The second loop terminates either when the temperature at the interface is high enough or when the ice layer plugs the channel.

As stated previously, the main virtue of the proposed 0D methodology is very short computational time. Consequently, it permits the investigation of a vast range of parameter regimes. Nevertheless, there exists the potential danger that the developed 0D model could lack some important physics. This creates a need for validation against a state-of-the-art Computational Fluid Dynamics (CFD) model, which is done in the subsequent section.

3. Mathematical and Numerical Model of the Water-LNG Heat Transfer

3.1. Mathematical Model

The numerical model developed in the current work is based on an open source CFD numerical toolbox, OpenFOAM (Open Source Field Operation and Manipulation) [36]. It incorporates a heat transfer between fluid and solid into one mathematical problem as a conjugate heat transfer (CHT). In this work, numerical calculations were based on the chtMultiRegionFoam solver readily available in OpenFOAM. It exploits a finite volume discretization in conjunction with the PIMPLE algorithm which is a combination of PISO (Pressure Implicit with Splitting of Operator) and SIMPLE (Semi-Implicit Method for Pressure Linked Equations) methods. The capabilities of OpenFOAM’s CHT solver have been studied extensively and some recent investigations into this matter can be found in [37,38].

The multi-domain approach uses a Navier–Stokes-based mathematical model for each considered fluid region and a heat conduction model for solid regions. The regions are coupled by thermal boundary conditions. For each fluid region, the compressible Navier–Stokes equation is solved and for the solid regions, only the energy equation:

$$\frac{\partial (\rho h)}{\partial t} = \frac{\partial}{\partial x_j}\left(\alpha \frac{\partial h}{\partial x_j}\right)$$

(15)

where $h$ is the specific enthalpy, $\rho$ the density and $\alpha = \kappa / c_p$ is the thermal diffusivity, which is defined as the ratio between the thermal conductivity $\kappa$ and the specific heat capacity $c_p$.

The effects of solidification and the melting processes were simulated using the OpenFOAM built-in model solidificationMeltingSource. The built-in model uses an enthalpy—porosity approach and was used successfully in other investigations where the solidification and melting processes were of crucial importance [39]. It assumes that the phase change occurs at melting temperature, $T_{\text{melt}}$. The model uses an additional field $\alpha_1$ to simulate the liquid fraction and its values can be defined as:

$$\alpha_1 = \begin{cases} 
0 & \text{when } T < T_{\text{melt}} \\
1 & \text{when } T > T_{\text{melt}}
\end{cases}$$

(16)

Furthermore, an intermediate, partially solidified region is introduced. The liquid fraction inside this mushy region takes on values between 0 and 1 [39].

The presence of the solid phase in the flow field is incorporated into the model as a porosity effect, with porosity equal to liquid fraction (changing from 0 to 1). Porosity equal to zero is used in fully solidified regions, which extinguishes the velocities there. The relationship between the porosity and momentum in the mushy zone takes on the following form [39]:

$$S = \frac{(1 - \alpha_1)^2}{\alpha_1^2 + \epsilon} \cdot A_{\text{mush}} \cdot (u - u_p)$$

(17)
where \(\epsilon\) is a small number (0.001) to prevent division by zero, \(A_{mush}\) is the mushy zone constant, and \(u_p\) is the solid velocity due to the pulling of solidified material out of the domain (pull velocity). The energy associated with the phase change is added to the fluid enthalpy equation \([36]\):

\[
H = H_{ref} + \int_{T_{ref}}^{T} c_p \Delta T + \Delta H
\]  

where \(H_{ref}\) is reference enthalpy, \(T_{ref}\)—reference temperature, \(c_p\)—specific heat at constant pressure and \(\Delta H\) is the latent heat content depending on liquid fraction \(\alpha_1\) and the latent heat of solidification \(r\):

\[
\Delta H = \alpha_1 \cdot r
\]

The latent heat content can vary in range from zero (for a solid) to \(r\) (for a liquid).

### 3.2. Numerical Model

In the currently developed numerical model the computational domain was divided into five separated regions as shown in Figure 5.

**Figure 5.** Numerical geometry considered in the proposed numerical model. The LNG regions are separated from the water region by walls.

This can be perceived as a simplified geometrical feature of a plate heat exchanger (PHE) \([40,41]\), which usually consists of a large number of thin plates separated by a small gap. Consequently, the PHE channels contain alternately hot and cold fluid.

In this study, only one channel for the heating fluid (water) is considered and it is separated by walls from two LNG (cold fluid) regions. The hydraulic diameter of the water channel was set to \(D_h = 8\) mm resulting in a distance of \(x_w = 4\) mm between the walls. The thickness of the walls \(x_{ss}\) separating the water channel and LNG was set to 1 mm.

The water channel region was modeled based on Navier–Stokes equations with solidification \((17)\) and thermophysical properties that correspond to liquid water or ice, approximated by polynomial functions as shown in Figure 6. However, the polynomials are of a very high degree, a special attention was paid to avoid the Runge error. The polynomials were defined and used only within a range of interest from 70 K to 320 K and defined using the least squares fit principle. Finally, polynomials up to 59 degrees were used. The main reason for using the high degree function was to assure a sudden change in properties caused by a phase change and to take advantage of an OpenFoam functionality which is available by default. Thanks to this, the presented methodology and results could be repeated using a standard version of the OpenFoam CFD toolbox.
The left and right plates were defined as solid regions with thermal properties of stainless steel 304 taken from [42]. It was assumed that the thermal conductivity and specific heat was a function of temperature as shown on Figure 7.

To apply the equation describing the boiling HTC of LNG (14) to the model, LNG regions were treated as solid. Consequently, this made it possible to apply to the Equation (14) directly to the properties of the LNG regions as properly defined LNG thermal conductivity $\lambda_{\text{LNG}}$. Consequently, a constant temperature of LNG on the outer boundaries could be assumed (Figure 5).

The relation between the LNG boiling HTC ($h_{\text{LNG}}$) and thermal conductivity $\kappa$ is based on equal heat transfer between the LNG regions and the plates: $\dot{q}_{\text{boiling}} = \dot{q}_{\text{conduction}}$. In the case of boiling LNG, the equation for the heat transfer can be written as:

$$\dot{q}_{\text{boiling}} = h_{\text{LNG}}(\Delta T) \cdot \Delta T_{\text{wall-LNG}}$$

and on the other hand, the heat conduction can be expressed as:

$$\dot{q}_{\text{conduction}} = \int_{T_{\text{LNG}}}^{T_{\text{wall}}} \lambda_{\text{LNG}}(T) dT$$

Consequently, a characteristic of an equivalent thermal conduction $\lambda_{\text{LNG,A}}$ for the LNG region is shown on Figure 8 and can be expressed as:

$$\lambda_{\text{LNG,A}}(T) = -1.613 \cdot 10^{-7} \cdot T^3 + 1.532 \cdot 10^{-4} \cdot T^2 - 0.0472 \cdot T + 5.1187$$
The Equation (22) assumes the width of LNG region to be $x_{LNG} = 2$ mm.

![Figure 8](image1.png)

**Figure 8.** Thermal conduction curves applied to the LNG regions with the width of $x_{LNG} = 2$ mm.

The numerical mesh independence study showed the strong influence of grid size in the horizontal direction, which corresponds to the direction of the ice formation. Finally, the cell size in the horizontal direction was set to 0.025 mm (40 cells per millimeter) and 1 mm in the vertical direction. The numerical calculations were performed until a stable level of liquid fraction inside the whole domain of the water channel was obtained. Figure 9 shows the variations of the global liquid fraction changes over time. Case C is observed to require the shortest amount of time (30 s) to achieve steady state, with case A needing the longest amount of time (60 s).

![Figure 9](image2.png)

**Figure 9.** The function of the global liquid fraction in respect to time for three considered cases A, B, and C, for $Re_w = 1000$.

The height of the computational domain was set individually based on $Re_w$ number, inlet water temperature and the considered case (A, B, or C). The exemplary views of the liquid fraction field inside the water channel of 25 cm in height and for different $Re_w$ number are presented in the Figure 10. Additionally, the impact of the water’s inlet temperature and the LNG boiling case, on the liquid fraction field is presented on Figures 11 and 12 respectively. The liquid fraction fields presented on Figure 12 correspond to the global liquid fraction of the steady state solution shown in Figure 9.
Figure 10. Liquid fraction inside the 25 cm high water channel for different \(Re_w\) and for the case A and 300 K temperature of the inletting water. Red and blue color indicates water and ice, respectively.

(a) \(Re_w = 100\)  (b) \(Re_w = 250\)  (c) \(Re_w = 500\)  (d) \(Re_w = 1000\)  (e) \(Re_w = 1500\)  (f) \(Re_w = 2000\)

Figure 11. Liquid fraction inside the 25 cm high water channel for different inlet water temperatures calculated for case A and for \(Re_w = 1000\).

(a) \(T_w = 290\)K  (b) \(T_w = 300\)K  (c) \(T_w = 310\)K

Figure 12. Liquid fraction inside the 25 cm high water channel for three considered cases A, B, and C, for 300 K inlet water temperature and \(Re_w = 1000\).

(a) Case A  (b) Case B  (c) Case C

The comparison of the numerical results with the simplified 0D calculation was done along horizontal lines for every millimeter along the height of the water channel. The values of the
temperature and liquid fraction were read from each mesh cell where corresponding profiles were created and the average liquid fraction for corresponding heights was calculated.

The water inlet was placed at the bottom of the water channel region, Figure 5. The water inlet velocity was calculated based on $Re_w$ numbers considered in this study: $Re_w = (100, 250, 500, 1000, 1500, 2000)$. Additionally, the kinematic viscosity of water was assumed to be temperature dependent. The water temperature at the inlet was assumed to be fixed and the temperature on the outer boundaries of LNG regions was set as 122 K.

4. Results and Discussion

A comparison of the 0D results and numerical solutions is presented in Figures 13–15 for LNG boiling cases A, B, and C, respectively. The numerical and 0D results show good agreement with discrepancies not exceeding 30%, 21% and 15% for LNG boiling cases A, B, and C. Moreover, accuracy increases with each increase of the $Re_w$ number. It is worth noting that the $Re_w$ number within the range of 100–2000 appears to not have a significant effect on the thickness of the ice layer.

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{fig13.png}
\caption{Comparison of the simplified 0D and numerical calculations of the ice layer thickness with respect to the heating water temperature for the case A of LNG boiling HTC.}
\end{figure}
On the other hand, the water inlet temperature significantly influences the thickness of the ice layer. It varies from 0.8 to 1.3 mm for a water inlet temperature of about 310 K and increases to about 2 mm for a temperature of 275 K and clogs the channel. It is worth noting that the thickness of the ice layer is greater for LNG boiling HTC cases B and C, as compared to case A. The increase of the LNG boiling HTC results in the increased possibility of freezing and consequently, also increases the thickness of the ice layer.

The proposed simplified 0D model shows satisfactory qualitative and quantitative agreement with the CFD-based numerical calculations. This assures that all the necessary physics is kept in the 0D model. The developed methodology is, however, significantly less time-consuming and permits the study of a much wider parameter space.
Figure 15. Comparison of the simplified 0D and numerical calculations of the ice layer thickness with respect to the heating water temperature for the case C of LNG boiling HTC.

It should be mentioned that the proposed CFD-based numerical model can have significantly wider applications, e.g., allowing the study of the dynamics of freezing water, which can then be used to study the influence of more complex geometries on the water freezing process.

Figure 16 presents a simplified 0D solution for the thickness of the ice layer with respect to the heating water temperature and $Re_w$ number for LNG boiling HTC cases A, B, and C, respectively. The analysis points out the dominant influence of the LNG boiling HTC and the heating water temperature. Furthermore, it suggests that the $Re_w$ number has a considerably smaller impact on the thickness of the ice layer. The lower the water temperature, the smaller the effect of the $Re_w$ change on the thickness of the ice layer. The ice layer formation causes the channel to be constricted and consequently, according to Equation (8), the HTC of water is also increased. Therefore, for very narrow channels, the OHTC, which governs the ice layer thickness, is nearly exclusively dependent on LNG boiling HTC. In the case of wider channels, the water HTC is small enough to play an important part in the determination of the OHTC and consequently, enhances the impact of the $Re_w$ number.
Figure 16. Simplified 0D solution of the ice layer thickness with respect to the heating water $Re_w$ number and the temperature for the cases of LNG boiling HTC A, B, and C.

Based on the results presented in Figure 16, a sensitivity analysis of the HTC of LNG on the thickness of the ice layer was carried out and can be seen in Figure 17. The increase of the HTC (represented by cases A, B, and C) can be observed to result in increased ice layer thickness, which is also a function of the water temperature and $Re_w$. The thickness of the ice layer is very sensitive to changes in the HTC, especially in the case of higher water temperatures and $Re_w$. Nevertheless, the relative increase in ice layer thickness caused by an increase in the HTC seems to always be out of proportion, especially as compared to the relative increase of the HTC of LNG. Even a small increase in the HTC of LNG can result in a significantly increased risk of freezing, which emphasizes the need to study deeply the influence of factors such as surface roughness and flow characteristics on LNG boiling.

The investigated range of water temperatures and $Re_w$ numbers can suggest a potential research direction to help minimize the risk of ice layer formation. To estimate the appropriate (minimal) requirements that can prevent freezing, an analysis of the minimum water HTC coefficient was performed. The analysis was based on heat transfer conditions that must be satisfied at the water side to prevent ice formation.

Figure 18 shows a minimum HTC at the water side $h_{w,\text{min}}$ that is necessary to prevent ice formation with respect to the water temperature and LNG boiling cases A, B, and C. The simulation was performed within the range of the $Re_w$ number from 100 to 20,000. The lower the water temperature, the higher the water HTC needs to be to prevent freezing. The results indicate that if the water HTC remains sufficiently high in the whole heat exchanger, freezing could be prevented, even for water temperatures as low as 280 K, and for any of the considered LNG boiling HTC cases.

The results suggest that safe LNG regasification that prevents freezing and clogging can be satisfied by maintaining the stable HTC of a heating medium in the heat exchanger. In the opinion
of the authors, this can be used as heat exchanger selection criteria. Consequently, safe regasification of LNG with minimal risk of freezing can only be performed in heat exchangers characterized by geometries promoting a constant HTC of the heating medium.

Figure 17. Sensitivity analysis of the HTC of LNG on the ice layer thickness with respect to ReW and temperature.

Figure 18. Minimum water HTC assuring no ice formation with respect to the water temperature and for the LNG boiling cases A, B, and C.
5. Conclusions

This paper has introduced a fast and reliable zero-dimensional computational methodology to study the freezing process in a straight rectangular channel. The proposed geometry could be seen as a simplified generalization of the heat exchanger. The proposed simplified 0D model showed qualitative and quantitative agreement with state-of-the-art CFD numerical calculations. This provides assurance that all the necessary physics have been preserved in the developed 0D model. Its main virtue is the possibility for calculations that are significantly less time- and resource-consuming. Consequently, it also makes it possible to study a wide range of parameter space and could be of potential interest for designers of heat exchangers and engineers.

The developed 0D model was used to analyze the LNG regasification process for a wide parameter space and to investigate the risk of freezing for a heating medium. Consequently, the mutual interactions and influence of the temperature, the Reynolds number, the HTC of the heating medium and the boiling HTC of LNG were examined. Based on the achieved results, the following conclusions could be drawn:

1. Ice layer formation during LNG regasification strictly depends on the heat transfer conditions. The dominant factors are the heating medium temperature and the LNG boiling HTC.
2. The regasification process can proceed without ice layer formation even for low-heating medium temperatures, at about 280 K, so long as the LNG boiling heat transfer is kept low and the heating medium HTC is sufficiently high.
3. Sensitivity analysis shows the overproportional influence that an increase in the HTC of LNG has on the potential risk of freezing. This means that even a small increase in HTC of LNG can result in a significant increase in ice layer thickness and rise in risk of freezing.
4. The LNG boiling HTC is a key factor in ice layer formation and should be subjected to an extensive experimental analysis. Moreover, the effects of low boiling HTC, surface roughness, and other possibly important factors should also be deeply analyzed.
5. Due to high temperature differences, the LNG boils in a film boiling regime which is characterized by a diminished HTC as compared to the nucleate boiling regime. However, in the considered case, limited LNG boiling HTC is necessary to prevent ice formation.
6. The proper construction of a heat exchanger for LNG regasification can help protect itself from risk of freezing and clogging. The heating medium HTC much be maintained and remain high enough in the whole heat exchanger. This should, in the authors’ opinion, be one of the main heat exchanger selection criteria in the regasification of LNG.


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Conflicts of Interest: The authors declare no conflict of interest.
Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>$A_{mush}$</td>
<td>mushy zone constant</td>
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<tr>
<td>$c_p$</td>
<td>specific heat</td>
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<tr>
<td>$D$</td>
<td>the characteristic size of the channel</td>
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<td>$D_h$</td>
<td>hydraulic diameter</td>
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<td>$g$</td>
<td>thickness</td>
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<td>$h$</td>
<td>convection heat transfer coefficient</td>
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<td>$H$</td>
<td>enthalpy</td>
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<td>$k$</td>
<td>heat conductivity</td>
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<tr>
<td>$L$</td>
<td>length of the channel</td>
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<tr>
<td>$Nu$</td>
<td>Nusselt number</td>
</tr>
<tr>
<td>$OHTC$</td>
<td>overall heat transfer coefficient</td>
</tr>
<tr>
<td>$Pr$</td>
<td>Prandtl number</td>
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<tr>
<td>$\dot{q}$</td>
<td>heat flux density</td>
</tr>
<tr>
<td>$r$</td>
<td>latent heat of solidification/melting</td>
</tr>
<tr>
<td>$Re$</td>
<td>Reynolds number</td>
</tr>
<tr>
<td>$T$</td>
<td>temperature</td>
</tr>
<tr>
<td>$x$</td>
<td>region width</td>
</tr>
<tr>
<td>$\Delta H$</td>
<td>latent heat content</td>
</tr>
<tr>
<td>$\Delta T$</td>
<td>temperature difference</td>
</tr>
<tr>
<td>$a_1$</td>
<td>liquid fraction</td>
</tr>
<tr>
<td>$\epsilon$</td>
<td>constant</td>
</tr>
<tr>
<td>$\lambda$</td>
<td>thermal conduction</td>
</tr>
</tbody>
</table>

Subscripts

- $A$ refers to case A of LNG boiling HTC
- $B$ refers to case B of LNG boiling HTC
- $C$ refers to case C of LNG boiling HTC
- $boiling$ refers to process of LNG boiling
- $cond$ refers to process of heat conduction
- $ice$ refers to ice
- $if$ refers to ice–water interface
- $l$ refers to laminar flow
- $LNG$ refers to LNG
- $melt$ refers to melting point
- $min$ refers to minimum
- $ref$ refers to reference point
- $ss$ refers to stainless steel
- $t$ refers to turbulent flow
- $w$ refers to water
- $wall$ refers to wall
- $wall1$ refers to the wall at the water side
- $wall2$ refers to the wall at the LNG side

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