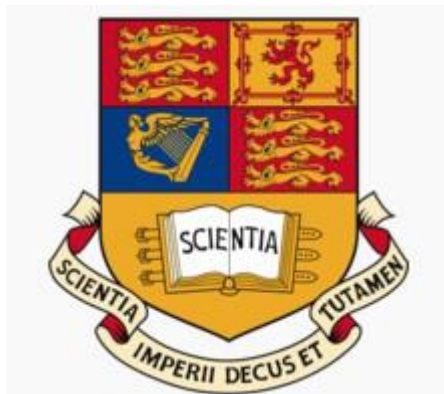


# ***Modelling Vaporizing Fluid Flow through Porous Media with Applications to Liquefied Natural Gas***



A thesis

submitted to the Department of Earth Science and Engineering

Imperial College London

in partial fulfilment of the requirement for the degree of

***Doctor of Philosophy***

by

***Emeka Joachim Okafor***

August 2012

# ***Modelling Vaporizing Fluid Flow through Porous Media with Applications to Liquefied Natural Gas***

*A Doctor of Philosophy Thesis*

This thesis and the research it reports are the product of my own work and any ideas or quotations from other people are fully acknowledged in accordance with the standard referencing practices of the discipline. This thesis is submitted to Imperial College London for the Degree of Doctor of Philosophy. No part of this thesis has already been or is being submitted for any other degree or qualification.

***Emeka Joachin Okafor***

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***This Thesis is dedicated to ...***

Almighty **GOD**

and to

E. Okafor – my unborn son

Dear Okafor Jr., If you had made it to mother earth, I am sure you would have loved to say a smiley “thank you” to all the wonderful people who made the successful completion of your father’s thesis possible.

# ***ABSTRACT***

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The problem of vaporizing flow of liquefied natural gas (LNG) through porous or penetrable media has received very little attention despite its importance in assessing the performance and risk-based safety of large membrane tank LNG ships under barrier leakages. In this work, a fluid flow model is proposed and used to analyse the vaporizing flow behaviour of LNG through soil and glass wool porous materials. Furthermore, a modified vaporizing liquid pool model is implemented and used to examine the problem of vaporizing LNG pool on non-penetrable solid substrates. We employed an explicit, finite difference and a fourth-order Runge-Kutta algorithms coded in FORTRAN to respectively solve the flow and pool models. Both models were successfully verified and validated by comparisons to experimental data, analytical solutions, and to predictions of a commercial software (TOUGH2).

Results from the vaporizing flow and pool analyses demonstrate that, for some of the applications considered, the liquid is expected to reach considered threshold depths, seep through the porous layer and contact, contaminate and/or embrittle surrounding natural or engineered systems. For the specific application to LNG cargo containment systems (or cargo tanks), this work has shown that there are safety risks associated with LNG leakage, which are ultra-low temperature of the inner hull, cryogenic damage and subsequent failure of the cargo containment system. Thus, for any LNG membrane cargo containment system to continue to be safe and secure, the various structural members of the insulation system should be designed and equipped with new and improved materials that possesses the necessary mechanical and thermophysical properties to maintain and/or improve the critical temperature standard and low-temperature performance of these systems.

Further work should consider additional experimental evidence in order to fully validate and establish that solution predictions by the proposed models are describing the actual physical effects.

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# Nomenclature

---

|                  |   |                                                                     |
|------------------|---|---------------------------------------------------------------------|
| $a$              | = | radius of particles in Moorhouse and Carpenters' model, m           |
| $A_{lsa}$        | = | heat exchange surface area, m <sup>2</sup>                          |
| $A$              | = | cross-sectional area of porous medium, m <sup>2</sup>               |
| $A$              | = | Coefficient dependent on soil properties in JGA Model, ...          |
| $A_{contact}$    | = | area of contact between the liquid and solid, m <sup>2</sup>        |
| $A_{pores}$      | = | total surface area of open pores, m <sup>2</sup>                    |
| $A_{s,specific}$ | = | specific surface area of porous material, m <sup>-1</sup>           |
| $Ar$             | = | Archimedes number                                                   |
| $B$              | = | Coefficient dependent on convection and radiation in JGA Model, ... |
| $B_{reid}$       | = | Boiling parameter in Reid's model, ...                              |
| $C$              | = | specific heat capacity, J/kg/K                                      |
| $C_n$            | = | specific heat capacity of non-penetrable substrate, J/kg/K          |
| $C_{ps}$         | = | specific heat capacity of substrate, J/kg/K                         |
| $C_{pl}$         | = | specific heat capacity of liquid, J/kg/K                            |
| $C_{vap}$        | = | specific heat capacity of vapour film, J/kg/K                       |
| $C_x$            | = | stability criterion in the $x$ -direction, ...                      |
| $C_y$            | = | stability criterion in the $y$ -direction,...                       |
| $f$              | = | interpolation function for transition boiling heat flux, ...        |
| $f_1$            | = | dimensionless function in Klimenko's correlation, ...               |
| $f_2$            | = | dimensionless function in Klimenko's correlation, ...               |
| $G$              | = | sink term in heat conduction equation, ...                          |
| $g$              | = | gravitational acceleration constant, m/sec <sup>2</sup>             |
| $\Delta H_{vap}$ | = | heat of vaporization of liquid, J/kg                                |
| $H$              | = | height of liquid pool, m                                            |
| $h$              | = | heat transfer coefficient, W/m <sup>2</sup> /K                      |
| $h_{trans}$      | = | transition boiling heat transfer coefficient, W/m <sup>2</sup> /K   |
| $h_{film}$       | = | film boiling heat transfer coefficient, W/m <sup>2</sup> /K         |
| $k_{dsf}$        | = | dimensionless shape factor, ...                                     |

|            |   |                                                                               |
|------------|---|-------------------------------------------------------------------------------|
| $k_l$      | = | thermal conductivity of liquid, W/m/K                                         |
| $k_k$      | = | kozeny constant, ...                                                          |
| $k_n$      | = | thermal conductivity of non-penetrable substrate, W/m/K                       |
| $k_o$      | = | hydraulic conductivity, m/sec                                                 |
| $k_s$      | = | thermal conductivity of substrate, W/m/K                                      |
| $K_s$      | = | permeability of porous medium, m <sup>2</sup>                                 |
| $k_{vap}$  | = | thermal conductivity of vapour film, W/m/K                                    |
| $L_{vap}$  | = | latent heat of vaporization, J/kg                                             |
| $L_{eff}$  | = | effective length, m                                                           |
| $L_t$      | = | number of grid points in the vertical direction, ...                          |
| $L_{st}$   | = | straight length, m                                                            |
| $l_c$      | = | critical wavelength of the instability, ...                                   |
| $\ell$     | = | penetration length for heat conduction, m                                     |
| $M$        | = | liquid pool mass, kg                                                          |
| $\Delta M$ | = | increment in liquid pool mass, kg                                             |
| $M_0$      | = | mass of liquid vaporized, kg                                                  |
| $m(z)$     | = | mass flow into the porous layer at point $z$ , kg                             |
| $m(z+dz)$  | = | mass flow out of the porous layer at point $z+dz$ , kg                        |
| $M_{vap}$  | = | mass of liquid vaporized, kg                                                  |
| $M_0$      | = | initial mass of liquid, kg                                                    |
| $M_{tot}$  | = | total mass of liquid vaporized, kg                                            |
| $M_{REM}$  | = | mass of liquid remaining, kg                                                  |
| $\dot{m}$  | = | mass flow rate of liquid, kg/sec                                              |
| $N_t$      | = | number of grid points in the horizontal direction, ...                        |
| $Nu$       | = | Nusselt number,...                                                            |
| $\Delta P$ | = | pressure gradient, Pa/m                                                       |
| $p_1$      | = | 1 <sup>st</sup> time-dependent fit parameter in Vinsome-Westerfeld model, ... |
| $p_2$      | = | 2 <sup>nd</sup> time-dependent fit parameter in Vinsome-Westerfeld model, ... |
| $P_1$      | = | inlet pressure, Pa                                                            |
| $P_2$      | = | outlet pressure, Pa                                                           |
| $Pr$       | = | Prandtl number,...                                                            |
| $q$        | = | heat flux, W/m <sup>2</sup>                                                   |
| $q_{max}$  | = | maximum heat flux, W/m <sup>2</sup>                                           |

|                  |   |                                                                       |
|------------------|---|-----------------------------------------------------------------------|
| $q_{min}$        | = | minimum heat flux, W/m <sup>2</sup>                                   |
| $q_{tak}$        | = | heat flux based on Takeno's model, W/m <sup>2</sup>                   |
| $q_{trans}$      | = | transition boiling heat flux, W/m <sup>2</sup>                        |
| $Q$              | = | quantity of heat released by solid matrix, W                          |
| $Q_v$            | = | heat content at a given vertical and/or horizontal node, W            |
| $r'$             | = | radius at a considered distance, m                                    |
| $r(t)$           | = | time-dependent pool radius, m                                         |
| $R$              | = | radius, m                                                             |
| $\Delta R$       | = | increment in liquid pool radius, m                                    |
| $R(t)$           | = | time-dependent pool radius, m                                         |
| $Re$             | = | Reynolds number,...                                                   |
| $s$              | = | number of solid particles in Moorhouse and Carpenter's model, ...     |
| $S$              | = | source term in heat conduction model, ...                             |
| $S$              | = | source parameter in advection-conduction model, sec <sup>-1</sup>     |
| $S_{reid}$       | = | parameter denoting slope of vaporized mass in Reid's model, kg/sec    |
| $\Delta t$       | = | time step size, sec                                                   |
| $t$              | = | time, sec                                                             |
| $t_{cr}$         | = | critical time above which vertical heat conduction is important, sec  |
| $t_{max}$        | = | maximum vaporization time of liquid pool, sec                         |
| $t_{ss}$         | = | time at steady state, sec                                             |
| $t_z$            | = | time of liquid penetration in $z$ -direction, sec                     |
| $t'$             | = | arrival time of spreading liquid at a considered radius ( $r'$ ), sec |
| $\Delta T$       | = | temperature difference, K                                             |
| $\Delta T_{min}$ | = | minimum temperature difference, K                                     |
| $\Delta T_{max}$ | = | maximum temperature difference, K                                     |
| $T$              | = | transient temperature of substrate, K                                 |
| $T_n$            | = | time-dependent temperature of non-penetrable substrate, K             |
| $T_s$            | = | initial temperature of non-penetrable substrate, K                    |
| $T_{s0}$         | = | uniform initial temperature of substrate, K                           |
| $T_{n0}$         | = | uniform initial temperature of non-penetrable substrate, K            |
| $T_b$            | = | boiling temperature of liquid, K                                      |
| $T_{crit}$       | = | critical temperature of liquid, K                                     |
| $u$              | = | Darcy velocity, m/s                                                   |
| $V_L$            | = | liquid penetration velocity, m/s                                      |

|              |   |                                                                           |
|--------------|---|---------------------------------------------------------------------------|
| $V$          | = | vaporization rate, kg/m <sup>2</sup> /sec                                 |
| $V_{bv}$     | = | bulk volume of porous medium, m <sup>3</sup>                              |
| $V_{Br\_sh}$ | = | vaporization rate based on Briscoe and Shaw model, kg/m <sup>2</sup> /sec |
| $V_{JGA}$    | = | vaporization rate based on <i>JGA</i> model, kg/m <sup>2</sup> /sec       |
| $V_{dreid}$  | = | vaporization rate based on Drake and Reid's model, kg/m <sup>2</sup> /sec |
| $V_{MrCa}$   | = | Moorhouse and Carpenter's vaporization rate model, kg/m <sup>2</sup> /sec |
| $V_{reid}$   | = | Reid's vaporization rate model, kg/m <sup>2</sup> /sec                    |
| $V_{ops}$    | = | vaporization rate based on Opschoor's model, kg/m <sup>2</sup> /sec       |
| $x$          | = | spatial distance in the $x$ -direction, m                                 |
| $x_w$        | = | weight fraction of moisture in substrate, ...                             |
| $\Delta x$   | = | horizontal grid size, m                                                   |
| $X$          | = | lateral or horizontal distance of porous layer, m                         |
| $y$          | = | spatial distance in the $y$ -direction, m                                 |
| $Y$          | = | variable used in Reid's heat flux calculations, ...                       |
| $\Delta y$   | = | vertical grid size for heat conduction model, m                           |
| $Z$          | = | depth or vertical distance of porous layer, m                             |
| $z$          | = | spatial distance in the $z$ -direction, m                                 |
| $z(t)$       | = | time-dependent liquid penetration depth, m                                |
| $z_{pen}$    | = | liquid penetration depth, m                                               |
| $\Delta z$   | = | vertical grid size for liquid flow model, m                               |

### Greek Symbols

|                  |   |                                           |
|------------------|---|-------------------------------------------|
| $\mu$            | = | liquid viscosity, Pas                     |
| $\mu_{vap}$      | = | vapour viscosity, Pas                     |
| $\sigma$         | = | surface tension, N/m                      |
| $\varphi$        | = | vapour mass fraction, ...                 |
| $\phi$           | = | porosity of substrate, ...                |
| $\rho$           | = | density, kg/m <sup>3</sup>                |
| $\rho_l$         | = | density of liquid, kg/m <sup>3</sup>      |
| $\rho_{vap}$     | = | density of vapour, kg/m <sup>3</sup>      |
| $\rho_{lp}$      | = | density of liquid pool, kg/m <sup>3</sup> |
| $\rho_{vap,sat}$ | = | density of vapour film, kg/m <sup>3</sup> |
| $\rho_s$         | = | density of substrate, kg/m <sup>3</sup>   |

|                |   |                                                                      |
|----------------|---|----------------------------------------------------------------------|
| $\rho_n$       | = | density of non-penetrable substrate, kg/m <sup>3</sup>               |
| $\nu$          | = | kinematic viscosity, m <sup>2</sup> /sec                             |
| $\nu_{vap}$    | = | kinematic viscosity of vapour film, m <sup>2</sup> /sec              |
| $\alpha_{vap}$ | = | thermal diffusivity of the vapour film, m <sup>2</sup> /sec          |
| $\alpha_s$     | = | thermal diffusivity of substrate, m <sup>2</sup> /sec                |
| $\alpha_n$     | = | thermal diffusivity of non-penetrable substrate, m <sup>2</sup> /sec |
| $\tau$         | = | tortuosity, ...                                                      |
| $\partial$     | = | thickness of vapour film, m                                          |
| $\partial$     | = | thickness of vapour film, m                                          |
| $\chi$         | = | substrate surface roughness factor, ...                              |
| $\lambda$      | = | correction term in Opschoor's model, ...                             |
| $\beta_1$      | = | first constant defined in Opschoor's model, ...                      |
| $\beta_2$      | = | second constant defined in Opschoor's model, ...                     |
| $\beta_3$      | = | third constant defined in Opschoor's model, ...                      |
| $\zeta$        | = | relative error notation, ...                                         |

### Subscripts

|       |   |                                          |
|-------|---|------------------------------------------|
| b     | = | boiling                                  |
| bv    | = | bulk volume                              |
| crit  | = | critical                                 |
| dreid | = | Drake and Reid                           |
| eff   | = | effective                                |
| film  | = | film boiling regime                      |
| i     | = | initial space counter                    |
| JGA   | = | Japan Gas Association                    |
| l     | = | liquid                                   |
| L     | = | liquid                                   |
| lp    | = | liquid pool                              |
| max   | = | maximum                                  |
| min   | = | minimum                                  |
| MrCa  | = | Moorhouse and Carpenter                  |
| n     | = | non-penetrable substrate                 |
| $n$   | = | initial space counter in the x-direction |

|         |   |                                          |
|---------|---|------------------------------------------|
| ops     | = | Opschoor                                 |
| pen     | = | penetration                              |
| pl      | = | liquid                                   |
| ps      | = | substrate                                |
| p       | = | time counter in liquid pool model        |
| REM     | = | remaining                                |
| s       | = | substrate                                |
| s0      | = | substrate                                |
| Br_sh   | = | Briscoe and Shaw                         |
| st      | = | straight                                 |
| tak     | = | Takeno                                   |
| tot     | = | total                                    |
| trans   | = | transition boiling regime                |
| $v$     | = | initial space counter in the y-direction |
| vap     | = | vapour, vaporization                     |
| vap.sat | = | vapour at saturation temperature         |

### Superscripts

|       |   |                                                             |
|-------|---|-------------------------------------------------------------|
| A-SS  | = | analytical solution of steady-state vaporizing flow model   |
| A-US  | = | analytical solution of unsteady-state vaporizing flow model |
| $GPT$ | = | number of grid points                                       |
| $k$   | = | initial time counter                                        |
| $n1$  | = | initial time counter in the 2-D model                       |
| $n0$  | = | initial time counter in the unsteady state 1-D model        |

### Acronyms

|       |   |                              |
|-------|---|------------------------------|
| B.O.R | : | Boil-Off Ratio               |
| CFD   | : | Computational Fluid Dynamics |
| CFL   | : | Courant, Friedrich, Levy     |
| CNG   | : | Compressed Natural Gas       |
| CCS   | : | Cargo Containment System     |
| cpu   | : | Central Processing Unit      |
| 2-D:  | : | Two-Dimensional              |
| 1-D   | : | One-Dimensional              |

|                 |                                                                    |
|-----------------|--------------------------------------------------------------------|
| EOS             | : Equation of State                                                |
| FDM             | : Finite Difference Method                                         |
| FTCS            | : Forward-time centred-space                                       |
| GHz             | : Gigahertz                                                        |
| GTT             | : Gaztransport and Technigaz                                       |
| IBS             | : Inter-barrier Space                                              |
| IEA             | : International Energy Agency                                      |
| IS              | : Insulation Space                                                 |
| K               | : Kelvin                                                           |
| LH <sub>2</sub> | : Liquid Hydrogen                                                  |
| LN <sub>2</sub> | : Liquid Nitrogen                                                  |
| LNG             | : Liquefied Natural Gas                                            |
| LPG             | : Liquefied Petroleum Gas                                          |
| LPG             | : Liquefied Petroleum Gas                                          |
| m               | : Meters                                                           |
| mm              | : Millimetres                                                      |
| NGH             | : Natural Gas Hydrate                                              |
| PDE             | : Partial Differential Equation                                    |
| PNG             | : Pipeline Natural Gas                                             |
| R-PUF           | : Reinforced Polyurethane Foam                                     |
| RPT             | : Rapid Phase Transition                                           |
| Sec             | : Seconds                                                          |
| SIGTTO          | : Society of International Gas Tanker and Terminal Operators       |
| SO              | : Soil                                                             |
| SS              | : Sandstone                                                        |
| SST             | : Stainless Steel                                                  |
| tcm             | : Trillion Cubic Meters                                            |
| VA              | : Void area (also called void space (VS) or insulation space (IS)) |

# 1

## Introduction

---

Natural gas is an important energy resource used as fuel and feedstock in the chemical and petrochemical industries (Dawe, 2000). For examples, it is used for power generation, domestic heating, and as vehicle fuels. The current increase in the demand for natural gas (IEA, 2011) is, in part, due to its cleanliness and ease of burning when compared to other heavier hydrocarbons such as coal and crude oil (Dawe, 2000). Figure 1.1 shows the projected rapid increase in natural gas consumption between 1990 and 2025.

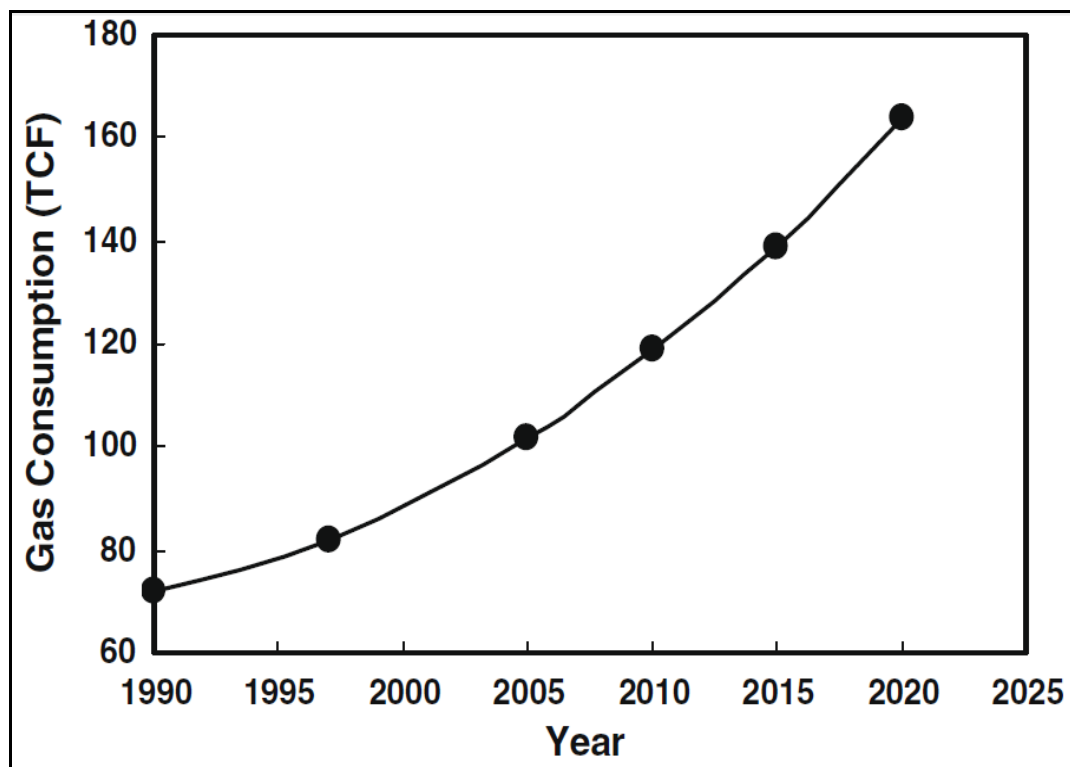


**Figure 1.1:** Projected share of global natural gas consumption between 1990 and 2025 (Seunyeoung, 2001; Najibi et al., 2009).

Natural gas reserves are usually remotely located and far away from gas consuming nations. Therefore, for natural gas to reach consumers of the product, it has to be transported to gas consuming nations and customers by the most cost-effective transportation technology. Examples of natural gas transportation technology include: Pipeline Natural Gas (PNG), Liquefied Natural Gas (LNG), Compressed Natural Gas (CNG), and Natural Gas Hydrate (NGH). Pipeline Natural Gas involves the use of large diameter pipes to transport the product. LNG technology involves cooling the gas until it turns into a liquid and then transporting the liquefied product via large LNG ships. CNG technology uses a pressurization process rather than a cooling process (as in LNG). NGH transportation alternative involves the conversion of stranded gas to hydrate for easier transportation to customers. Figure 1.2 shows that the most cost-effective

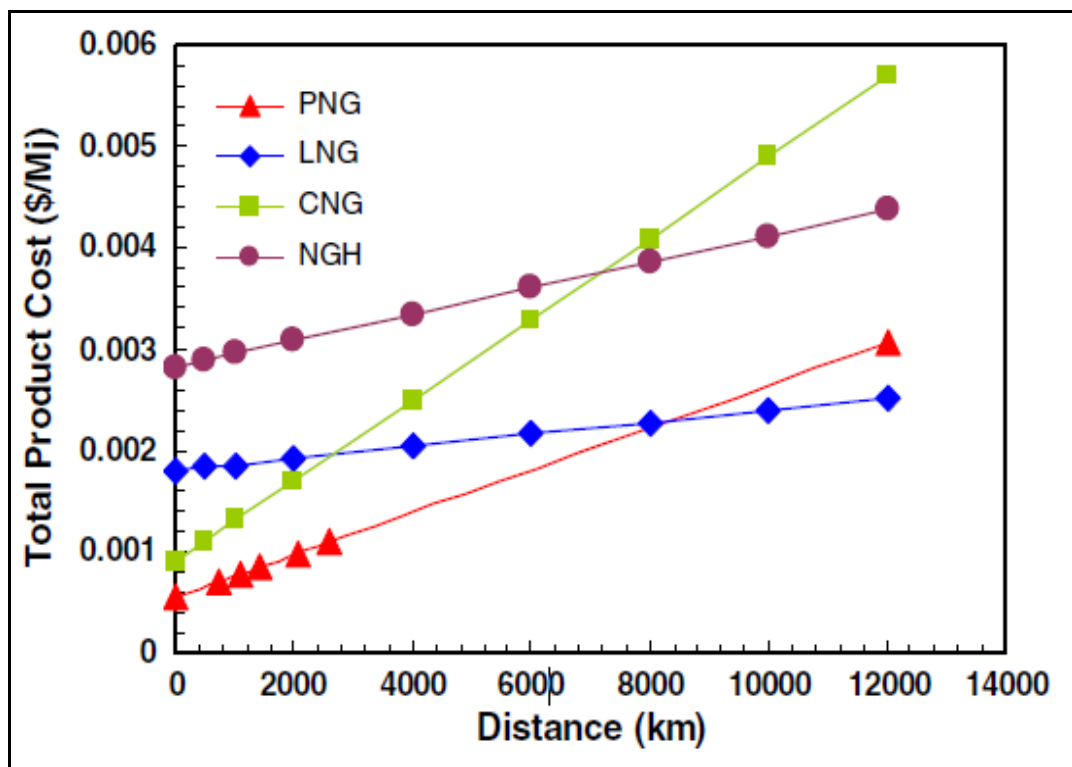


**Figure 1.2:** A graph showing the effect of distance on the profitability and/or total product cost based on the four different natural gas transportation alternatives (Najibi et al., 2009).

means of natural gas transportation is affected by the distance between the place where the natural gas is produced and the place where it is consumed. It is evident from Figure 1.2, which is based on a study evaluating the profitability of

natural gas transportation from Iran's South-Pars gas field to market, that there is a critical distance where LNG would become the most profitable means of natural gas transportation.

An LNG transportation chain (or value chain) is a diagram showing the different stages of natural gas transportation via LNG, from the point where the natural gas is explored and produced to the point where it is distributed and consumed. The LNG transportation chain shown in Figure 1.3 depicts a typical LNG transport scenario. First, natural gas is sent to a liquefaction plant from the gas field. The liquefaction process involves cooling the natural gas at atmospheric pressure to a temperature of approximately 112 K. This cooling process reduces the volume of the gas to about 1/600<sup>th</sup> times its original volume.



**Figure 1.3:** An LNG transportation chain showing a typical LNG transport scenario. Safety is also considered across the entire LNG transportation chain.

This resort to the use of LNG transportation technology for long-distance natural gas transportation rather than the other transport alternatives is mainly to reduce cost as no cost-reducing transport alternative, say pipeline technology,

would surpass the economic benefits of using the LNG alternative. In addition, safety across the entire LNG transportation chain are usually ensured.

There are safety implications associated with LNG transportation. For example, with new orders for extremely large LNG ships being placed, there are persistent safety and security concerns in respect of the consequences of both small and large LNG spillages, whether through accidental events such as grounding and collision or through intentional events such as a terrorist action. Whichever accidental spill situations may lead to damage to property (Figure 1.4 shows the type of damage to an LNG cargo tank caused by grounding) or to more catastrophic consequences which could endanger lives and property. Thus, it is necessary to continuously address these safety and security concerns regardless of the exemplary safety standard that the LNG industry has upheld over the years (Hightower et al., 2004).

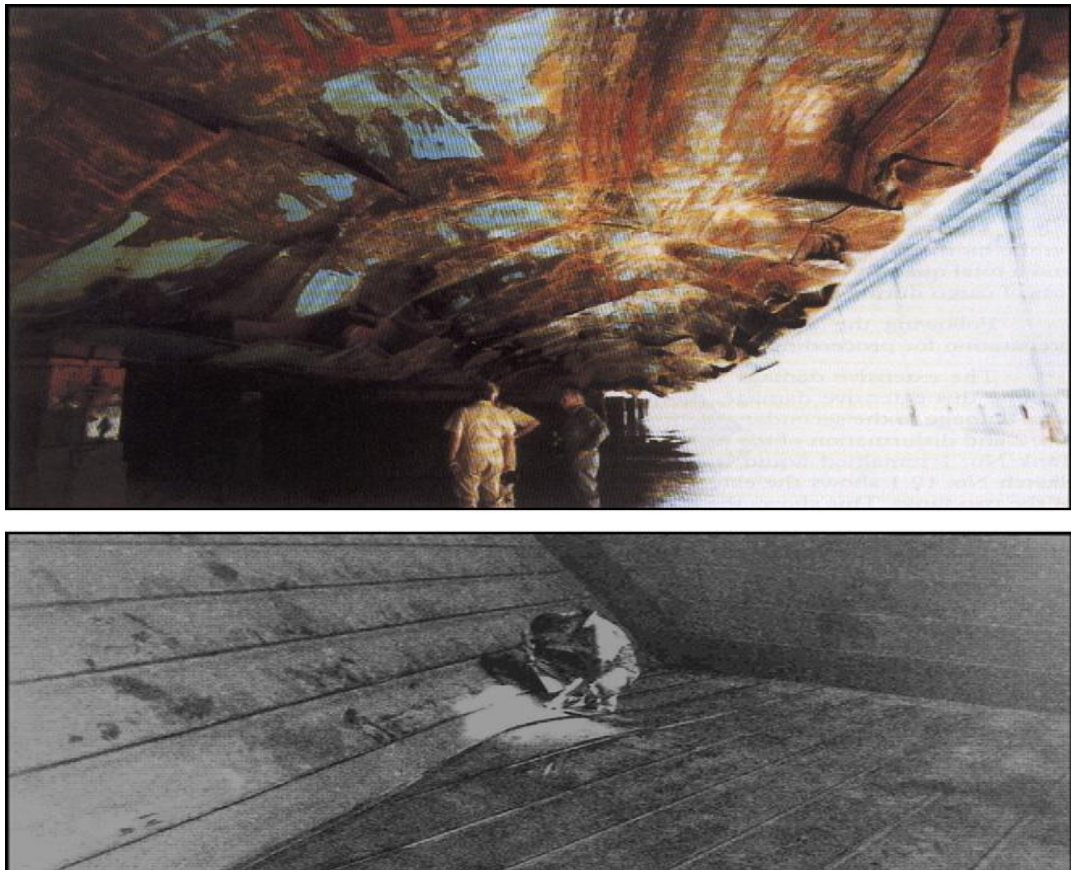


**Figure 1.4:** Damage from grounding of an LNG ship (Woodward and Pitblado, 2010).

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The safety and security concerns associated with LNG operations are mainly due to the chemical and flammability characteristics of the cryogen. Table 1.1 lists some of the chemical properties of LNG produced from three different natural gas producing countries of the world - Nigeria, Oman and Algeria (Woodward and Pitblado, 2010). We note from Table 1.1 that the upper and lower flammability limits of LNG are approximately 5% and 15% by volume, respectively. This suggests that a mixture of LNG with air within this range of concentrations is flammable. The flammable nature of LNG creates a number of hazards and we will try to briefly identify, in the next two paragraphs, some of the specific hazards associated with the spillage of LNG.

**Table 1.1:** Some chemical properties of LNG (Woodward and Pitblado, 2010).

| <i>Property</i>                         | <i>Nigeria</i> | <i>Oman</i> | <i>Algeria</i> |
|-----------------------------------------|----------------|-------------|----------------|
| Molecular weight                        | 17.8           | 18.6        | 18.8           |
| Boiling point, K                        | 113            | 113         | 113            |
| Liquid density, kg/m <sup>3</sup>       | 453            | 464         | 453            |
| Vapour density, kg/m <sup>3</sup>       | 1.78           | 1.76        | 1.78           |
| Temperature at lift-off, K (293 K air)  | 186            | 192         | 199            |
| Heat of combustion (higher MJ/kg)       | 49.4           | 49.2        | 49.2           |
| Carbon footprint (gCO <sub>2</sub> /MJ) | 56.3           | 57          | 56.9           |
| Flammable range for vapour (mole %)     | 4.6-14.6       | 4.4-14.4    | 4.4-14.4       |
| Vapour reactivity classification        | low            | low         | low            |

Figure 1.5 shows a typical LNG spillage scenario including an identification of the hazards associated with the spillage. First and foremost, the release of LNG from containment will result in an immediate vapour flash or aerosol formation. This is followed by the spillage of the liquid onto a substrate where a pool of the cryogen will form immediately on the surface of the substrate. The type of substrate onto which the LNG has spilled can, depending on the location of the breached containment system or process equipment, be liquid or solid. The most common type of liquid substrate is water while the most common type of solid substrate is soil. For either substrate, the developing pool of liquid simultaneously spreads and vaporizes. If the LNG pool is immediately ignited, a pool fire results. If the pool is unignited, a dense vapour-cloud forms above the pool and this vapour-cloud can be highly asphyxiating if inhaled. An unignited vapour-cloud would continue to hover above the LNG pool and then slowly

disperse with time. If the vapour-cloud is later ignited, a vapour-cloud fire that may burn back towards the pool may result, eventually forming a pool fire. LNG pool fires can cause severe thermal radiation hazards to humans and property. Moreover, if an ignited vapour-cloud is within confined areas, a combustible explosions can result. Contact of the cryogen with humans can cause tissue damage and contact with metals and materials can cause embrittlement or brittle fracture. For spillages of LNG onto water, rapid phase transition (RPT)

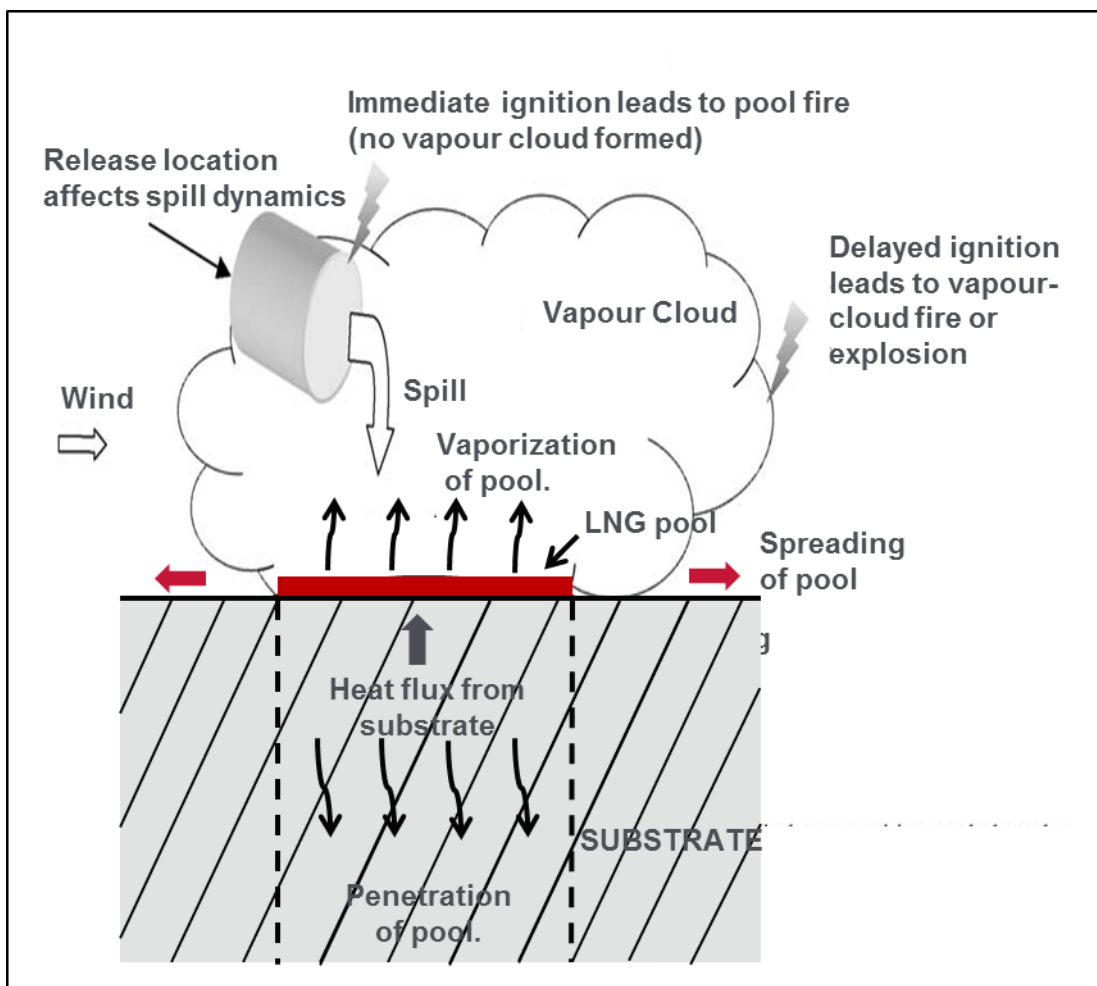


**Figure 1.5:** Pool formation, spreading, vaporization and penetration when LNG is spilled (Adapted from Luketa-Hanlin (2004)).

which is essentially a physical explosion may result. For spillages of LNG onto permeable substrate, penetration of the cryogen into the substrate may result, with the depth of penetration affected by the structural and thermophysical properties of the substrate such as moisture content and porosity.

There has been, at least, thirty six minor and major LNG accidents since 1944 with Table D.1 in Appendix D providing a chronological summary of these accidents. Some of the causes of LNG accidents include: sloshing damage, grounding, collision, overfilling, containment system leakage, valve leakage, rollover, contact, fire and explosion, cargo tank failure, and brittle fracture of structural members as a result of cryogenic damage. Table 1.2 below lists the distribution of a few of these accidents according to category while Figure 1.6 shows a typical fracture caused by a 16 mm thick deck stiffener as a consequence of LNG contact.

**Table 1.2:** Distribution of historic LNG accidents (Vanem, 2007).

| <i>Accident Category</i> | <i>Accidents</i> |
|--------------------------|------------------|
| Collisions               | 19               |
| Groundings               | 8                |
| Contact                  | 8                |
| Fire and explosion       | 10               |
| Failure of cargo tank    | 27               |

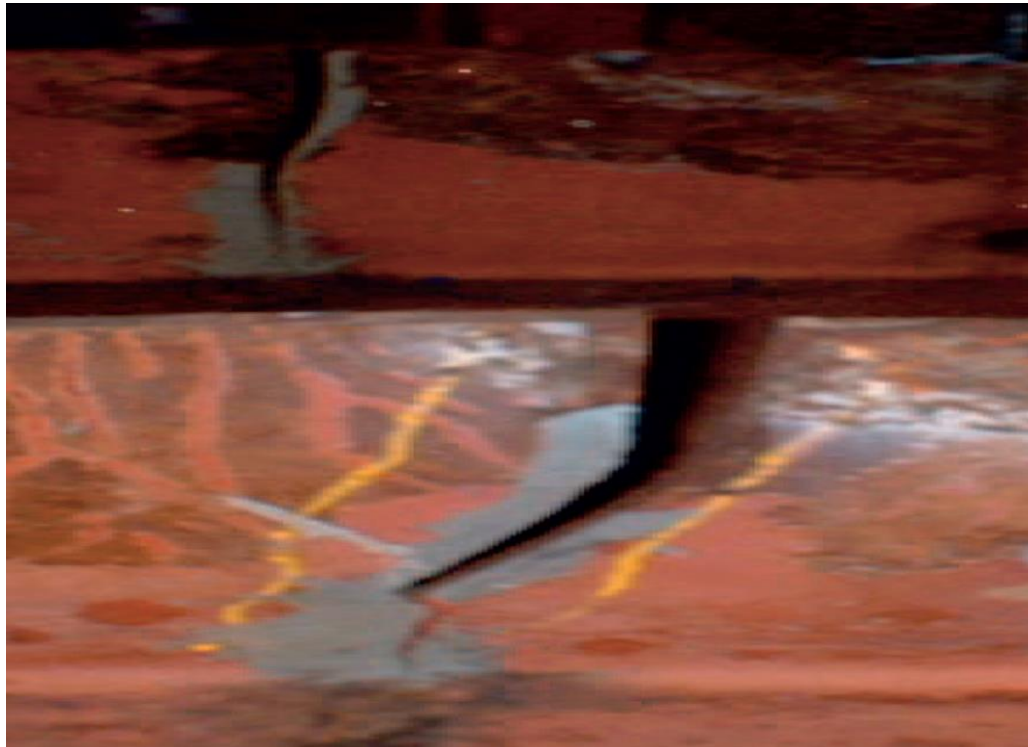


**Figure 1.6:** Photographs showing fractures caused in a 16 mm thick deck stiffeners as a consequence of an LNG spillage (SIGTTO, 2007).

Because of the consequences of LNG accidents that may lead to some of the LNG hazards identified above, questions are still being asked by industry and regulators alike on how to prevent and/or mitigate the risks and hazards associated with potential spillages of this liquefied gas. It has been recognized by industry and regulators that both consequence modelling and experimental research are vital means of answering some of these questions, and research efforts have centred primarily on modelling the different physical processes which occur in the event of an LNG spillage including:

- Release rate of LNG following a containment breach or rupture;
- Spreading rate of LNG pool on substrate surface;
- Vaporization rate of LNG pool on substrate surface;
- Cold and dense vapour-cloud dispersion upon pool vaporization;
- LNG pool/vapour-cloud combustion upon ignition;
- Thermal radiation from pool and/or vapour-cloud fires;
- Gas explosion if ignition occurred within complex geometries;
- Rapid phase transition (RPT) for spills on water;
- LNG penetration into substrate for spills on penetrable substrates.

All of these modelling issues are discussed in excellent review articles and reports detailing the state-of-the-art in the consequence modelling of LNG spillages (e.g. Hightower, et al., 2004; Hightower, et al., 2006, Luketa-Hanlin, 2004; Government Accountability Office, 2007; LNG Special Issue, 2007; Woodward and Pitblado, 2010). These reports and articles highlighted the limits of current understanding in consequence modelling of LNG spillages and provided suggestions for further research and development. In this work, we will focus on addressing one aspect of the problem – the problem of vaporizing flow of LNG through porous or penetrable media. As already indicated, this type of problem occurs for spill situations where the substrate is porous and penetrable. We are particularly interested in modelling the simultaneous vaporization and penetration rates of the cryogen as it flows downwards through the soil, glass wool or sandstone porous materials.

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A review of literature revealed that only a few record of modelling studies investigating this specific subject has been reported (Takeno, et al., 1994; Opschoor, 1981; Daish et al., 1998; Melhem, 2006). Other authors like Reid and Wang (1978), Reid (1980), Shaw and Briscoe (1980), Jensen (1983) and Cook and Woodward (1993) simply ignored LNG penetration through the substrate despite several experimental evidence (e.g. Drake and Reid, 1975; Japan Gas Association, 1976; Reid and Wang, 1978; Reid, 1980; Moorhouse and Carpenter, 1986; Takeno et al., 1994) which have demonstrated that a spill of LNG on penetrable, solid substrates results in the penetration and flow of the cryogen downwards through the substrate. There are safety implications associated with such vaporizing flow behaviour, one of which is cryogenic damage of equipments and materials, and subsequent failure of the system.

Providing some detail on existing model limitations, Melhem (2006) presented a one-dimensional analytical solution for gravity-driven penetration of a liquid into soil, giving penetration distance of the liquid front versus time. However, his analytical solution omitted heat conduction and vaporization effects, thus making the solution applicable only for non-vaporizing liquids, ones that are outside the configuration relevant in this work. The solutions by Opschoor (1981) cannot answer the question of how far a continuous flow of a given LNG volume can penetrate into the soil before the liquid front has reached a characteristic threshold depth inside the porous layer. Takeno et al. (1994) did not study LNG behaviour. In addition, their calculation of liquid penetration velocity is unrealistic. Choi et al. (2012) ignored modelling the vaporizing flow of LNG through glass wool. Thus, existing models are lacking because they either ignored liquid penetration (e.g. Drake and Reid, 1975), quantified liquid penetration using a multiplicative correction factor (e.g. Opschoor, 1980), modelled liquid penetration but assumed an unrealistic and/or difficult-to-measure constant penetration velocity (e.g. Takeno et al., 1994), measured liquid penetration but ignored vaporization effects (Melhem, 2006), or predicted leakages but ignored penetration rates (Choi et al., 2012).

Models are therefore needed to improve understanding in the prediction of vaporizing flow of fluids through porous media, most especially, LNG.

Specifically, the lack of appropriate physical model to describe the vaporizing flow of LNG through porous or penetrable substrates is the main motivation for this work. Sections 1.1 – 1.3 respectively state the objectives, key contributions and structure of the current thesis.

## 1.1 Project Objectives

The objectives of the present work have been to address the following issues:

- develop a generic fluid flow model which can be used to predict vaporizing LNG flow through porous or penetrable media;
- evaluate and understand the role of heat transfer on vaporizing cryogenic liquid flow through porous or penetrable media;
- estimate and assess the role of boiling regime on vaporizing fluid flow through porous or penetrable media;
- identify some of the key parameters influencing the fluid and heat transfer aspects of the flow because of their effects on liquid penetration rates;
- predict the behaviour of a vaporizing pool on non-penetrable media
- apply the developed model to barrier leakages of large LNG ships.

## 1.2 Thesis Contribution

The contributions of this thesis include among others:

- 1) development of a general model that accounts for the effects of heat transfer to LNG flowing through porous or penetrable media. Although the research is driven by LNG needs, the model is more generic and can be applied to other problems.
  - 2) provision of original contribution by advancing current understanding in the area of cryogenic liquid penetration into porous or penetrable solid substrates. Previous related work such as those by Reid and co-workers (Reid, 1980; Reid and Wang, 1978); Melhem (2006); and Takeno and co-workers (Takeno et al., 1994) either ignored or inadequately addressed this problem.
-

- 3) provision of original contribution by advancing current understanding in the area of vaporizing LNG flow through porous regions of large membrane type LNG ships under barrier leakages. Previous related work such as those by Bae and co-workers (Bae et al., 2007); Lee and co-workers (Lee et al., 2011); and finally Choi and co-workers (Choi, et al., 2012) either ignored or inadequately addressed this problem.

## 1.3 Thesis Structure

This thesis is split into 8 chapters:

- Chapter 2      reviews the relevant literature underlying the present work including a description of the physics of fluid and heat flow.
- Chapter 3      presents a derivation of the vaporizing flow model in which a one-dimensional steady-state heat conduction is applied. This chapter also presents both numerical solution and verification of a derived steady-state model.
- Chapter 4      presents an extension of the model developed in Chapter 3, now addressing the problem of vaporizing liquid flow with one-dimensional unsteady-state heat conduction. It also addresses the issue of heat transfer coefficient (or boiling regime effects) on liquid flow and vaporization behaviour.
- Chapter 5      presents a two-dimensional unsteady-state heat conduction model. The aim here is to compare the 2-D model results with those obtained from running the model developed in Chapter 4 to see how they differ in their predictions of relevant flow phenomena.
- Chapter 6      presents a vaporizing pool model formed from seepages from a porous medium. The aim here is to predict the cooling rate of non-penetrable, solid substrate surfaces as well as the spreading and vaporization rates of the seeped liquid.

Chapter 7      presents simulated results for three applications including:

- (i)      LNG flow through flat joints of large LNG ships,
- (ii)     LNG pool on inner hull of large LNG ships,
- (iii)    Water penetration through superheated rock.

Chapter 8      presents main conclusions and makes suggestions for further work.

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# 2

## ***Literature Review***

---

The main objective of this chapter is to review the literature on experimental and modelling studies of vaporizing flow of cryogenic liquids following accidental spillages on porous or penetrable substrates. The review also discussed liquid vaporization studies in which the substrates are non-penetrable such as can be found during LNG vaporization on inner hull steel structures of membrane tank LNG ships. Please note that, in this thesis, the words “vaporizing”, “vaporization” and “boiling” are considered synonyms and may therefore be used interchangeably, and unless otherwise stated, the word liquid refers to LNG modelled in this work using the properties of pure liquid methane. The organization of this chapter is as follows: In Section 2.0, the mechanisms of vaporizing liquid flow processes are discussed. The concept of porous material is introduced. In addition, relevant fluid flow, heat transfer and vaporization equations and/or correlations are provided. Section 2.1 presents a review of previous cryogenic liquid vaporization studies and this section is split into two themes. The first theme briefly discusses (in Sub-section 2.1.1) some previous studies on cryogenic liquid vaporization following spills on non-penetrable, liquid substrates such as water. This discussion of spills on water substrates is mainly to give a “big picture” of the research problem and to explain what happens when cryogenic liquids are spilled on water. A more extensive state-of-the-art review of previous studies of cryogenic liquid vaporization following spills on porous or penetrable substrates are presented in Sub-section 2.1.2. In Section 2.2, a table of all factors needing to be modelled and who modelled each one are presented to help set the motivation and scope of the work presented in this thesis. Finally in Section 2.3, the conclusions drawn from this chapter are presented.

## 2.0 Vaporizing Flow Processes

It is useful to show the nature of the real problem by first presenting in some detail a description of the vaporizing flow process. When liquid at its saturation temperature is injected into a porous medium (that is initially filled with air) of relatively higher initial temperature, an immediate vaporization of the flowing liquid will ensue, resulting in a multiphase flow within the porous layer bounded by solid matrices. Figure 2.1 shows a simple schematic diagram of the vaporizing

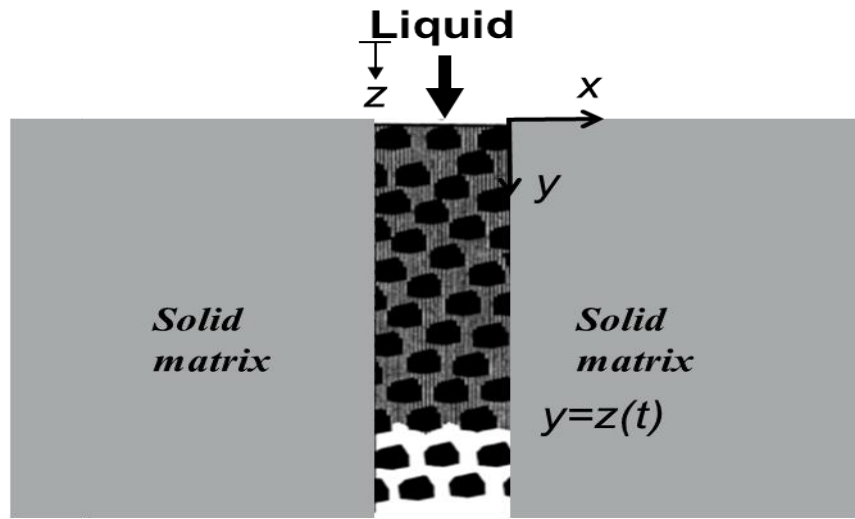


**Figure 2.1:** A schematic diagram showing the vaporizing flow of liquid through a porous medium bounded by two impermeable solid matrices.

flow process. The bounding solid matrices are assumed impermeable to fluid flow but may allow multi-dimensional heat exchange at the interface between the flowing liquid and the solid matrix. Figure 2.1 indicates that while the liquid flows downwards in the vertical  $z$ -direction, heat conduction within the solid matrix may occur in both the lateral (horizontal) and vertical direction. The characteristics of the vaporization process including penetration rate will have considerable bearing on the amount of heat being released by the solid matrices. As the liquid vaporizes, a vaporizing two-phase zone (or region) will evolve within the porous layer and the leading edge of this two-phase zone will, depending on the heat and fluid flow characteristics, propagate deeper into the porous layer. As time progress, more zones will subsequently evolve within the porous layer including a single-phase liquid zone, a two-phase liquid-vapour zone, a single-phase vapour zone and a single-phase air-saturated zone. More details of the

respective zones will be elucidated in Sections 3.11 and 4.3.3 of Chapters 3 and 4, respectively.

### 2.0.1 Description of Porous Materials

The term porous materials is used to describe solid materials consisting of void spaces or pores. Several common examples of porous materials abound including soil, sand, sandstone and glass wool. Porous materials have many industrial applications, for example, glass wool is used as a heat insulator in membrane tank LNG ships because of its low thermal conductivity properties while soil and sand can be used as dyke floor materials in LNG containment system design and construction.

To characterize porous materials, use can be made of their macroscopic parameters (Bear, 1988; Bear and Bachmat, 1991; Dullien, 1979). Porosity is a very good and important macroscopic parameter and can be defined as the fraction of the total volume occupied by the void space of a porous layer (Bear, 1988). The void space consist of pores that are either inter-connected, dead end or isolated (Kaviany, 1999). When the pores are inter-connected, their porosity is termed effective porosity. However, for loosely-packed nonconsolidated media, the porosity and the effective porosity are assumed to be equivalent whereas the opposite is the case for some consolidated media (Kaviany, 1999). The magnitude of porosity depends on the type of porous material or packing arrangements. Fibrous porous materials such as glass wool have higher porosity compared to those of soil or sandstone (Kaviany, 1999). Table 2.1 gives a summary of average bulk porosities of some fibrous and non-fibrous porous materials.

Another macroscopic parameter is permeability which describes the ability of fluids to flow through porous materials. The Darcy equation (Bear, 1988; Darcy, 1856) given as Equation 2.1 of Section 2.0.2 contains the permeability parameter,  $K_s$ . Different methods can be used to determine the permeability of a porous material. The so-called capillary models apply the Navier-Stokes equations in the determination of permeability (Bear, 1988). These models express the relationship between permeability and the matrix property parameters (Kaviany, 1999). There is also the hydraulic radius models which use

the concept of hydraulic radius in the expression of permeability as a function of porosity and specific surface area of the porous material (Carman, 1937). The hydraulic radius model was used to derive the so-called Kozeny-Carman equation and more details of this equation will be provided shortly.

**Table 2.1** Examples of average bulk porosities of some fibrous and non-fibrous porous materials (Kaviany, 1999).

| <i>Property</i>           | <i>Porosity</i> |
|---------------------------|-----------------|
| Foam metal                | 0.98            |
| Fibre Glass               | 0.88-0.93       |
| Soil                      | 0.43-0.54       |
| Sand                      | 0.37-0.50       |
| Brick                     | 0.12-0.34       |
| Sandstone                 | 0.08-0.38       |
| Limestone, dolomite       | 0.04-0.10       |
| Coal                      | 0.02-0.12       |
| Concrete (ordinary mixes) | 0.02-0.07       |

There is also the drag models where the porous matrix is envisioned as a collection of objects and then applying the Navier-Stokes equations to solve for flow over these objects (Kaviany, 1999). Thereafter the total resistance to the flow is compared with the Darcy resistance, ultimately yielding the permeability for these drag models. Table 2.2 gives bulk permeability of some porous materials.

**Table 2.2:** Bulk permeabilities of some porous materials (Kaviany, 1999).

| <i>Property</i>     | <i>Permeability (<math>m^2</math>)</i>        |
|---------------------|-----------------------------------------------|
| Fibre Glass         | $2.4 \times 10^{-11}$ - $5.1 \times 10^{-11}$ |
| Soils               | $2.9 \times 10^{-13}$ - $1.4 \times 10^{-11}$ |
| Sand                | $2.0 \times 10^{-11}$ - $1.8 \times 10^{-10}$ |
| Brick               | $4.8 \times 10^{-15}$ - $2.2 \times 10^{-13}$ |
| Sandstone           | $5.0 \times 10^{-16}$ - $3.0 \times 10^{-12}$ |
| Limestone, dolomite | $2.0 \times 10^{-15}$ - $4.5 \times 10^{-14}$ |

Like other solids, porous materials can conduct heat. The rate of heat conduction in porous materials is primarily influenced by the thermal conductivity of the solid matrix. Because of the presence of void space or pores in a porous material, this thermal conductivity effect can also be visualized in terms of effective thermal conductivity defined as the total measured thermal conductivity of a

block of porous material (Kaviany, 1999). Thus, the effective thermal conductivity is not only controlled by the thermal conductivity of the solid matrix, but also by the topology of the internal structure (Kaviany, 1999).

### 2.0.2 Liquid Flow

Fluid flow through porous media can be described by Darcy's law (Darcy, 1858; Dullien, 1979; Kaviany, 1999; Bear, 1988). The Darcy law for a uni-directional linear, liquid flow through a porous medium is given by,

$$u = -\frac{K_s}{\mu}(\nabla P - \rho_l g) \quad (2.1)$$

where  $\nabla P$  is the pressure gradient (Pa/m),  $K_s$  is permeability ( $\text{m}^2$ ),  $\mu$  is dynamic viscosity (Pas),  $\rho_l$  is the liquid density ( $\text{kg/m}^3$ ) and  $g$  is the acceleration due to gravity ( $\text{m/sec}^2$ ). The symbol,  $s$ , denotes substrate. Equation 2.1 indicates that the liquid flowrate for a porous medium would be proportional to the pressure gradient and this proportionality is essentially the permeability of the porous medium,  $K_s$ , divided by the dynamic viscosity,  $\mu$ , of the fluid. As already stated, a relation used to estimate permeability is the Carman-Kozeny equation (Dullien, 1979, Bear, 1988). This equation, initially developed for creeping flow in a packed bed of spherical particles, relates the permeability to the geometry of a porous medium and has also been used for modelling Darcy flow through other pore structures including fibrous structures. Further discussion on the Carman-Kozeny equation is presented in Section 3.6 of Chapter 3.

### 2.0.3 Heat Transfer

There are several heat transfer applications of porous materials. For example, a high permeability glass wool porous material with an interconnected void phase would allow fluids to penetrate and access the internal surfaces of the substrate, thus making the glass wool porous layers potentially suitable for convective heat exchange. In heat transfer analysis, the convective heat exchange between a flowing cryogenic liquid and the solid matrix can be estimated from the Newton's Law of Cooling (Bird et al., 2002) given by,

$$q = h(T - T_b) \quad (2.2)$$

where  $q$  (W/m<sup>2</sup>) is the heat flux,  $h$  (W/m<sup>2</sup>/K) is the convective (boiling) heat transfer coefficient,  $T$  (K) is the temperature at the interface between the liquid and solid and  $T_b$  (K) is the boiling temperature of the flowing liquid. Equation 2.2 is applicable only during film and transition boiling regimes (Bird, et al., 2002; Conrado and Vesovic, 2000). For nucleate boiling, the heat flux can be modelled using Equation 2.3 (Carslaw and Jaeger, 1959b),

$$q = -k_s \frac{\partial T}{\partial x} \quad (2.3)$$

where  $k_s$  (W/m/K) is the thermal conductivity of the solid matrix, and  $\partial T/\partial x$  is the lateral temperature gradient in the solid matrix. In the next section (Section 2.0.3.1), we discuss the possible boiling regimes that could occur during vaporizing liquid flow through porous media as well as identify and present relevant correlations for estimating boiling heat transfer coefficients.

### 2.0.3.1 Boiling Regimes

The boiling curves shown in Figures 2.2 and 2.3 are simply heat flux,  $q$ , versus temperature difference,  $\Delta T$ , curves first observed by Nukiyama (1934) who noted the two stable regions (or regimes) during his experiment involving the pool boiling of water around a heated platinum wire (Nukiyama, 1934). These two regions were identified by Nukiyama (1934) as nucleate boiling region (Region II of Figure 2.2) and the film boiling region (Region IV of Figure 2.1), respectively. The two extremes of the curves – the so-called maximum and minimum heat fluxes ( $q_{\max}$  and  $q_{\min}$ ) were also located by Nukiyama (1934). The transition boiling region of the curve (Region III of Figure 2.2) were not studied until Drew and Muller (1937) observed this region of the boiling curve. Figure 2.2 indicates that the boiling curve begins with a region of pure convection (Region I) where heat is transferred by superheated liquid rising to the liquid-vapour interface where evaporation takes place (McNelly, 1953). This is followed by a

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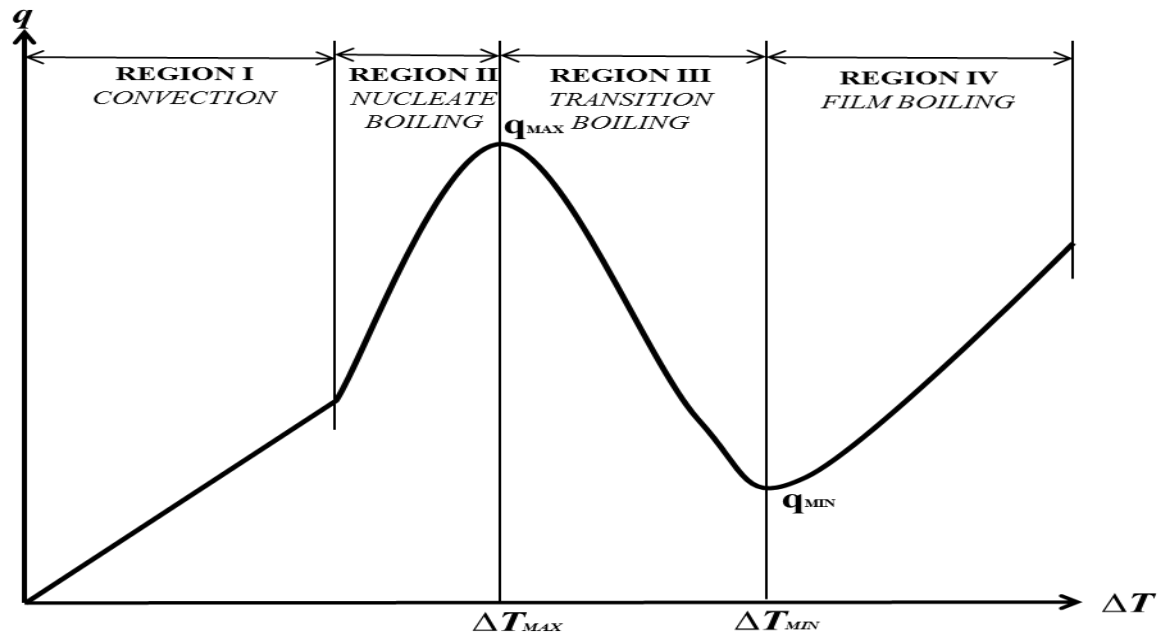


**Figure 2.2:** A schematic diagram showing the boiling curve with the three main boiling regimes (regions) identified.

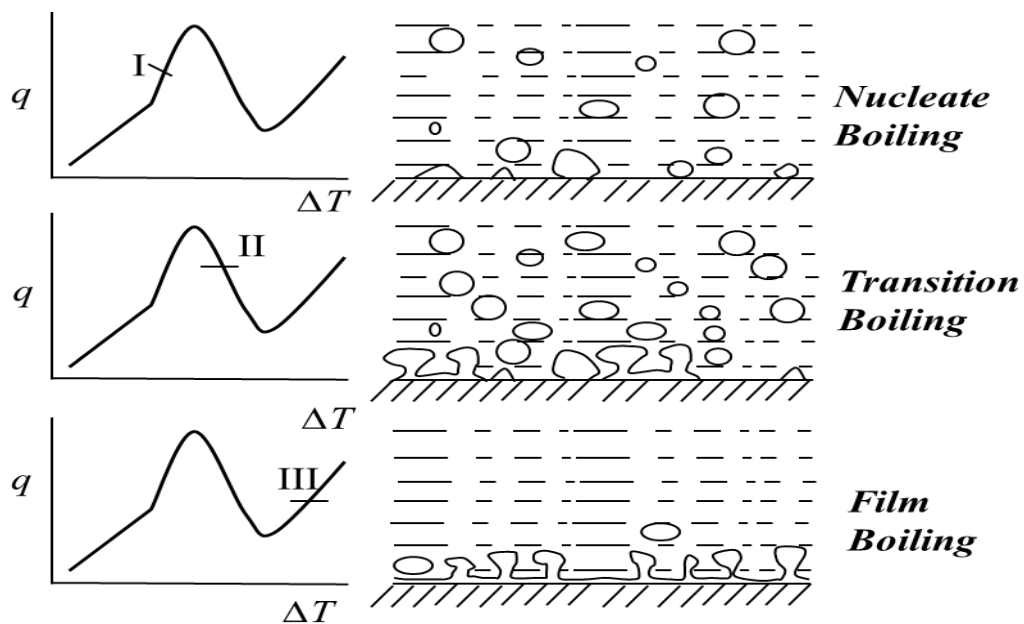


**Figure 2.3:** A schematic representation of the boiling curve showing the difference between the three main regimes of boiling – nucleate, transition and film boiling regimes.

region of nucleate boiling (Region II) characterized by bubble condensation in superheated liquid which then rises to the interface between the liquid and the vapour where vaporization takes place (McNelly, 1953). The third region (Region

III) – the so-called transition boiling region is characterized by partial nucleate boiling combined with unstable vapour-film preventing the liquid from reaching the heating surface. The stable film boiling region (Region IV) is the region where the heat transfer occurs by convection through the vapour film. Figure 2.3 shows, diagrammatically, the exact difference between the three main regimes of boiling (i.e. film, transition and nucleate boiling). In the next paragraphs, these three main boiling regimes including other regions of the boiling curve are explained in more detail. Furthermore, some of the correlations and equations used in predicting each portion or point of the boiling curve (shown in Figure 2.2) are discussed.

### **Nucleate Boiling Heat Flux**

Nucleate boiling is the boiling process that results in a liquid phase transforming into a vapour phase at the active nucleate sites on a heating surface. The nucleate boiling region in Figure 2.2 is shown as Region II. In a nucleate boiling process, the surface characteristics of the heating surface, the dynamics of bubble growth and the population of active nucleation sites all affect nucleate boiling behaviour. There are several correlations reported in literature that can be used to predict the nucleate boiling region. Westwater (1961a, 1961b) has reported on 14 of these correlations. Kosky and Lyon (1968) compared some suggested nucleate boiling correlations with their experimental results measured on a horizontal, flat, platinum-plated copper disk. Other useful nucleate boiling correlations were those of McNelly (1953) and Kutateladze (1963). For boiling of cryogenic fluids, Brentari and Smith (1976) presented relevant nucleate boiling data from the literature and selected Kutateladze's (1963) correlation for comparison, after which they concluded that the correlation fell within the spread of experimental data, no matter what geometries, orientations and surfaces of heater were used. The correlations by Grigorev et al. (1973) and Stephan and Abdelsalam (1980) further included the thermophysical properties of the solid material or its coating. For the purpose of this work, nucleate boiling calculations were based on heat conduction theory, which will be discussed in Section 2.0.4 of this chapter.

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### Maximum Heat Flux

The maximum heat flux  $q_{\max}$  is the critical heat flux above which the nucleate boiling regime will cease to exist (see Figure 2.2). Some researchers have either described the boiling situation at maximum heat flux as the interfacial instability of two counter current-flowing streams of inviscid fluids or the Kelvin-Helmholtz instability (Kutateladze, 1948; Zuber, 1959; Zuber et al., 1962). A number of correlations have been developed for the prediction of maximum heat flux. In this work, the maximum heat flux will be calculated using the following correlation (Opschoor, 1980; Conrado and Vesovic, 2000),

$$q_{\max} = 0.16L_{\text{vap}}\rho_{\text{vap,sat}}^{1/2}[\sigma g(\rho_l - \rho_{\text{vap,sat}})]^{1/4} \quad (2.4)$$

where  $L_{\text{vap}}$  is the latent heat of vaporization per unit mass,  $\rho_{\text{vap,sat}}$  is the density of the vapour with the subscript “sat” indicating that the thermophysical properties of the vapour will be determined from the boiling temperature of the cryogenic liquid. The other variables are as defined previously or on the list of nomenclature. The maximum temperature difference corresponding to the maximum heat flux will be calculated using the following expression (Kalinin et al., 1975, Opschoor, 1980; Conrado and Vesovic, 2000),

$$\Delta T_{\max} = 0.625(q_{\max}\sigma T_b)^{1/3} \frac{\left[ \frac{10}{(\rho_s C_{ps} k_s)^{1/2}} + \frac{\sqrt{\nu}}{k_l} \right]^{2/3} \left[ 1 + 10 \left( \frac{\rho_l C_{pl} k_l}{\rho_s C_{ps} k_s} \right)^{1/2} \right]^{1/3}}{1 + 10 \left( \frac{\rho_{\text{vap,sat}}}{\rho_l - \rho_{\text{vap,sat}}} \right)^{2/3}} \quad (2.5)$$

All the variables in Equation 2.5 are defined on the list of nomenclature. A knowledge of Equation 2.5 (which is mainly controlled by fluid, vapour film and substrate thermophysical and transport properties) together with the expression for the minimum temperature difference (to be discussed shortly) will be used in the determination of the transition boiling heat flux discussed next.

### Transition Boiling Heat Flux

The next region of the boiling curve is the transition boiling region shown in Figure 2.2 as the region between the maximum heat flux,  $q_{\max}$  and the minimum heat flux,  $q_{\min}$  (Westwater and Santangelo, 1955; Berenson 1960; Zuber, 1959; Kalinin et al., 1975). Transition boiling is the most complex and the least studied boiling regime and the mechanism of this boiling region has always been based on whether there exists liquid-solid contact. Kalinin et al. (1975) also studied transition boiling and successfully developed a model to predict the heat flux of transition boiling for cryogenic fluids. The correlations of Kalinin et al. (1975) are based on the assumption that the overall heat flux was controlled by the combined effect of nucleate and film boiling, and also by the effect of transient conduction at the points of liquid-solid contact. The following expression for the transition boiling heat flux was suggested by Kalinin et al. (1975),

$$q_{trans} = q_{\max} f + q_{\min} (1 - f) \quad (2.6)$$

where  $f$  is the statistically averaged fraction of the surface in contact with liquid at a given moment of time (also called the interpolation function for the determination of the transition boiling heat flux). The interpolating function,  $f$ , being used in Equation 2.6 can be correlated by the following equation (Kalinin et al., 1975),

$$f = \left( 1 - \frac{\Delta T - \Delta T_{\max}}{\Delta T_{\min} - \Delta T_{\max}} \right)^7 \quad (2.7)$$

where  $\Delta T$  is the temperature difference between the flowing liquid and the solid matrix and  $\Delta T_{\min}$  is the minimum heat flux which is discussed next.

### Minimum Heat Flux

The onset of film boiling is marked by the minimum heat flux,  $q_{\min}$  (see Figure 2.2) which, as already noted above, is required for the determination of the transition boiling region. The vapour film that is present at the minimum heat

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flux is just sufficient to maintain the mechanism of stable film boiling phenomena. The expression for the minimum heat flux used in this thesis is given by Equation 2.8 (Kalinin et al., 1975; Opschoor, 1980),

$$q_{\min} = 0.18k_{\text{vap}}\Delta T_{\min} \left[ \frac{g}{\nu_{\text{vap}}\alpha_{\text{vap}}} \left( \frac{\rho_l}{\rho_{\text{vap}}} - 1 \right) \right]^{1/3} \quad (2.8)$$

where  $k_{\text{vap}}$  is the thermal conductivity of the vapour film,  $g$  is the gravitational acceleration,  $\nu_{\text{vap}}$  is the kinematic viscosity of the vapour film,  $\alpha_{\text{vap}}$  is the thermal diffusivity of the vapour film,  $\rho_l$  and  $\rho_{\text{vap}}$  are liquid and vapour film densities, respectively. The minimum temperature difference,  $\Delta T_{\min}$ , corresponding to the minimum heat flux was estimated in this work using the following relation (Kalinin et al., 1975; Opschoor, 1980),

$$\Delta T_{\min} = (T_{\text{crit}} - T_b) \left[ 0.16 + 2.4 \left[ \frac{\rho_l C_{pl} k_l}{\rho_s C_{ps} k_s} \right]^{1/4} \right] \quad (2.9)$$

It may be worthwhile to point out that Equation 2.9 was formulated using the expression of convective heat transfer coefficient,  $h$ , (Berensen, 1960; Berensen, 1962; Kalinin et al., 1975, Opschoor, 1980, Conrado and Vesovic, 2000). For transition boiling, convective heat transfer coefficient can be calculated from Equation 2.10,

$$h_{\text{trans}} = q_{\max} f + q_{\min} (1 - f) / \Delta T \quad (2.10)$$

which is simply Equation 2.6 divided by the temperature difference between the flowing liquid and the solid matrix.

### Film Boiling Heat Flux

Film boiling can be defined as the type of boiling regime where a stable vapour film covers the heating surface. Since the generated vapours are of low thermal

conductivity, heat transfer rates during film boiling are low and there is little or no influence of the variables of the heating surface on the heat transfer characteristics except for situations where radiative heat transfer (not shown in the boiling curve of Figure 2.2) is large, and emissivity must be considered or where the surface roughness is great enough to disturb the vapour film. A wide variety of correlations have been developed for film-boiling (e.g. Bromley, 1950; Chang, 1957; Chang, 1959; Zuber, 1959; Zuber, 1958a; Zuber, 1958b; Berenson, 1960; Berensen, 1961; Klimenko, 1981). Klimenko (1981) proposed an approach for predicting film boiling by considering four limiting solutions for film boiling regime. He also considered both laminar and turbulent vapour flow in the film with and without allowance for friction at the liquid-vapour interface. According to Klimenko (1981), his suggested film boiling expressions correlated all of the available experimental data with a accuracy of  $\pm 25\%$ . For the purposes of this work, we will be taking a cue from other authors (Conrado and Vesovic, 2000; Vesovic, 2007; Zubairu, 2011) by using the correlations of Klimenko (Klimenko, 1981) for the estimation of the film boiling heat transfer coefficient given by,

$$h_{film} = \frac{k_{vap} Nu}{l_c} \quad (2.11)$$

where  $k_{vap}$  is the vapour thermal conductivity in film conditions,  $Nu$  is the Nusselt number for film boiling and  $l_c$  is the critical wavelength of instability. The Nusselt number,  $Nu$ , which represents the ratio of convective to conductive heat transfer either across a boundary or normal to it (boundary) essentially correlates the heat transfer coefficient between the fluid and the boundary. For the laminar region, the Nusselt number is given by,

$$Nu = 0.19[Ar Pr]^{1/3} f_1 \quad (2.12)$$

while, for the turbulent region, the Nusselt number,  $Nu$ , is expressed as,

$$Nu = 0.0086(Ar)^{1/2}(Pr)^{1/3} f_2 \quad (2.13)$$


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where  $Ar$  is the Archimedes number (which determines the motion of fluids due to density differences). The symbol,  $Pr$ , is the vapour phase Prandtl number (a dimensionless number which expresses the ratio of momentum diffusivity to thermal diffusivity) as,

$$Pr = \frac{C_{vap} \mu_{vap}}{k_{vap}} \quad (2.14)$$

where  $C_{vap}$ ,  $\mu_{vap}$  and  $k_{vap}$  are the vapour specific heat capacity, vapour viscosity and vapour thermal conductivity, respectively. The Archimedes number,  $Ar$ , is given by

$$Ar = (2\pi)^3 \frac{\sigma^{3/2}}{\rho_{vap} \nu_{vap}^2 (g(\rho_l - \rho_{vap}))^{1/2}} \quad (2.15)$$

where  $\nu$  is the kinematic viscosity which defines the ratio between dynamic viscosity and fluid density ( $\nu = \mu / \rho_l$ ). Equation 2.15 defines the relative strengths of free and forced convection and can be used to determine when the flow is in the laminar or turbulent region. The expression for the near unity correction factor,  $f_1$  as proposed by Klimenko (1981) for the laminar region is,

$$f_1 = 1 \quad \text{for} \left( \frac{\Delta H_{vap}}{C_{pl} \Delta T} \right) \leq 1.4 \quad (2.16a)$$

$$f_1 = 0.89 \left( \frac{\Delta H_{vap}}{C_{pl} \Delta T} \right)^{1/3} \quad \text{for} \left( \frac{\Delta H_{vap}}{C_{pl} \Delta T} \right) > 1.4 \quad (2.16b)$$

and the correction factor,  $f_2$  for the turbulent region is,

$$f_2 = 1 \quad \text{for} \left( \frac{\Delta H_{vap}}{C_{pl} \Delta T} \right) \leq 2 \quad (2.17a)$$

$$f_2 = 0.71 \left( \frac{\Delta H_{vap}}{C_{pl} \Delta T} \right)^{1/2} \quad \text{for} \quad \left( \frac{\Delta H_{vap}}{C_{pl} \Delta T} \right) > 2 \quad (2.17b)$$

The symbol,  $\Delta T$ , denotes the temperature difference (also called the “superheat”) between the liquid and the substrate and,  $\Delta H_{vap}$ , the heat of vaporisation.

#### 2.0.4 Heat Conduction

To be able to estimate the heat exchange rate between a flowing liquid and the solid matrix, unique heat conduction problems have to be formulated and solved. Furthermore, the temperature of the solid matrix is expected to change with time. The calculation of this temperature change may be achieved through a solution of the Fourier’s law of heat conduction (Carslaw and Jaeger, 1959b). Fourier’s law in three dimensions can be expressed as,

$$\rho_s C_{ps} \frac{\partial T}{\partial t} = \frac{\partial}{\partial x} \left( k_x \frac{\partial T}{\partial x} \right) + \frac{\partial}{\partial y} \left( k_y \frac{\partial T}{\partial y} \right) + \frac{\partial}{\partial z} \left( k_z \frac{\partial T}{\partial z} \right) + (S - G) \quad (2.18)$$

where  $T$  refers to the time-and-space-dependent temperature and  $k_x$ ,  $k_y$ , and  $k_z$  are known functions of  $x$ ,  $y$ ,  $z$ ,  $T$  and  $t$ , and stand for the thermal conductivities along the principal axes of the solid material, respectively. Equation 2.18 is derived by applying the Fourier’s equation in each Cartesian direction and considering the flow of heat through an infinitesimal volume element oriented in the three-dimensional coordinate system shown in Figure 2.4 where the sides  $dx$ ,  $dy$  and  $dz$  have been taken as parallel to the  $x$ ,  $y$  and  $z$  axis, respectively. The symbol  $G$  is a sink term and it is a known function of  $x$ ,  $y$ ,  $z$ ,  $T$  and  $t$  which refers to the volumetric rate of heat generation. Also,  $S$  is a source term and is a known function of  $x$ ,  $y$ ,  $z$ ,  $T$  and  $t$  which represents the volumetric rate of the thermal energy sink. The other two variables  $\rho_s$  and  $C_{ps}$  are the density and the specific heat capacity of the substrate (at constant pressure), respectively. Equation 2.18 can be solved for the unknown quantity,  $T$ , which represents the temperature distribution all over the solution domain. A solution

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of this problem can, in principle, be obtained for any given set of sufficient boundary conditions. It is worthwhile to mention that heat conduction problems can be classified as linear or non-linear. In Equation 2.18, the thermal conductivities depend on temperature and this dependence makes the governing heat conduction equation (i.e. Equation 2.18) non-linear. In linear heat conduction problems, thermal conductivities are independent of temperature.

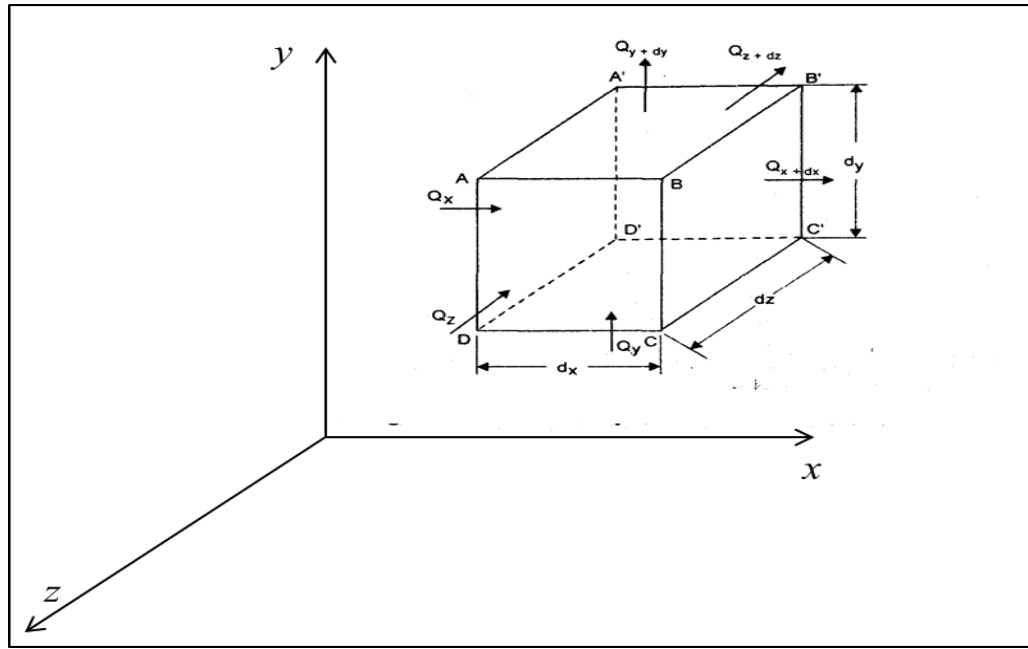


**Figure 2.4:** A schematic representation three-dimensional heat conduction analysis in Cartesian co-ordinates.

Equation 2.19 is a reduced form of Equation 2.18 considering a linear two-dimensional heat conduction case of an isotropic material where  $k_x = k_y$ ,

$$\frac{1}{k}(\rho_s C_{ps} \frac{\partial T}{\partial t} - S + G) = \left( \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right) \quad (2.19)$$

A linear three-dimensional formulation of Equation 2.18 will then reduce to,

$$\frac{\partial T}{\partial t} = \alpha_s \left( \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right) \quad (2.20)$$

where  $T$  is now a function of  $t$ ,  $x$ ,  $y$ , and  $z$  that represents the temperature at time  $t$  and at position  $(x, y, z)$  in the porous medium. The constant,  $\alpha_s$ , depends on the type of materials involved, and is called the thermal diffusivity of the porous matrix. In this thesis, we will be formulating and solving both one-dimensional and two-dimensional linear and transient heat conduction problems for both plain and porous media vaporization problems.

### **2.0.5 Liquid Vaporization Rate**

From our knowledge of the heat exchange rate between the flowing liquid and the solid matrix, the determination of the liquid vaporisation rate is straightforward and can be calculated by using the following first order ordinary differential equation (ODE) (Conrado and Vesovic, 2000),

$$V = \frac{dM_{vap}}{dt} = \frac{A_{lsa}q}{L_{vap}} \quad (2.21)$$

where  $V$  is the vaporisation rate,  $A_{lsa}$  ( $m^2$ ) is the liquid-solid contact area and  $M_{vap}$  is the mass of vaporised liquid.

## **2.1 Review of Previous Studies on Spills on Penetrable and Non-penetrable Substrates**

In this section, the state-of-the-art on cryogenic liquid vaporization following spills on both penetrable, solid substrates and non-penetrable, liquid substrates are presented but with greater emphasis on the former since these are of prime importance in this work. In Section 2.1.1, a somewhat descriptive review of what happens when cryogenic liquids such as LNG spills on water surfaces are provided while in Section 2.1.2, a critical review of fifteen studies dealing with cryogenic liquid vaporization following spills on penetrable, solid substrates are presented.

### **2.1.1 Spills on Non-penetrable Liquid Substrates**

Several experimental and modelling studies have been reported for liquid (LNG) spills on water. The earliest experiments on water spills were laboratory tests by

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Drake (1975a) and Jeje and Reid (1974) in which they measured vaporisation rates of liquid methane, ethane and nitrogen. Further tests were later conducted for light hydrocarbon mixtures (Drake et al., 1975b; Jeje and Reid, 1974). Details of these experimental series can be found in Drake et al. (1975a), Drake et al. (1975b) and Jeje and Reid (1974). Observations from the cited experimental investigations (Drake et al., 1975a; Drake et al., 1975b; Jeje and Reid, 1974) confirm that LNG vaporization rate is affected by chemical composition of the cryogen. Studies on the influence of chemical composition on cryogenic liquid pool behaviour following spills on water were further reported by a number of researchers (Drake et al., 1975; Boe, 1998; Valencia-Chavez, 1978, Valencia-Chavez, 1979; Conrado and Vesovic 2000). Boe (1998), for example, performed laboratory scale experiments with liquefied methane-ethane and methane-propane mixtures boiling on water and found that addition of ethane or propane increases the vaporization rate of the cryogen. Boe (1998) also concluded that film boiling ceased due to closer contact between the mixture and water causing a higher heat flux and lower temperature difference below that to maintain a continuous vapour film. Conrado and Vesovic (2000) also examined the importance of the chemical composition on the vaporization rate of LNG spilled on water surfaces and concluded that both the vaporization rate and the time for complete vaporisation are sufficiently different for pure liquid methane and LNG spills to warrant the use of the mixture model when predicting LNG behaviour following spills on water.

Later work by Vesovic (2007) examining the influence of ice formation on the vaporization rate of LNG has been reported. Figure 2.5 is a schematic diagram of the model by Vesovic (2007). The bulk water is separated from the LNG by a thermal layer of water and a vapour film. Vesovic's (2007) analytical solution for the vaporisation rate was derived and used to calculate the maximum vaporization time for spills for which the surface temperature of water is maintained constant. From one example simulation result which was used to show the performance of his model, Vesovic (2007) found that 50 tonnes of LNG took approximately two minutes to completely vaporize, indicating a rather rapid vaporization process. His steady-state model results were compared to that of Fay (2003), Opschoor (1977), and Raj and Kalelkar (1974) and a discrepancy was

noted by Vesovic (2007) which he attributed to the use of a different LNG spreading rate model. The author validated his model with LPG data (Vesovic, 2007) and noted better agreement over the results of the other contributors (Fay, 2003; Opschoor, 1977; Raj and Kalelkar, 1974).

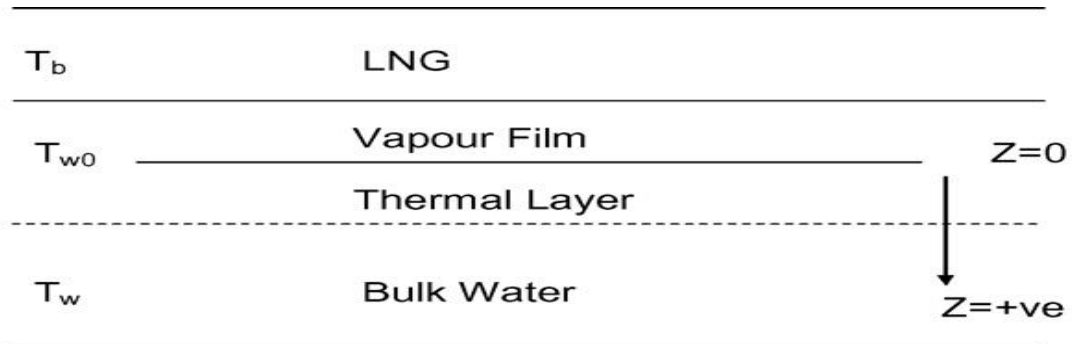


**Figure 2.5:** A schematic diagram of vaporisation of LNG on a liquid water surface. The author argues that there is a thin, thermal layer of water including a vapour film that initially separates the spilled LNG from the bulk water. Heat transfer is assumed to take place in the  $z$ -direction as shown (Vesovic, 2007).

Vesovic [2007] further developed a heat transfer model for confined spills in which Fourier's law of heat conduction was used to calculate the surface temperature of water as a function of time. His equation was used to calculate the time it takes for ice formation to occur assuming film boiling of the cryogen. His results reveal that it takes approximately 1 to 2 s for ice to form. The author went on to investigate LNG vaporization process after ice formation (the 'Ice' model is shown in Figure 2.6) and noted that ice significantly influences the heat

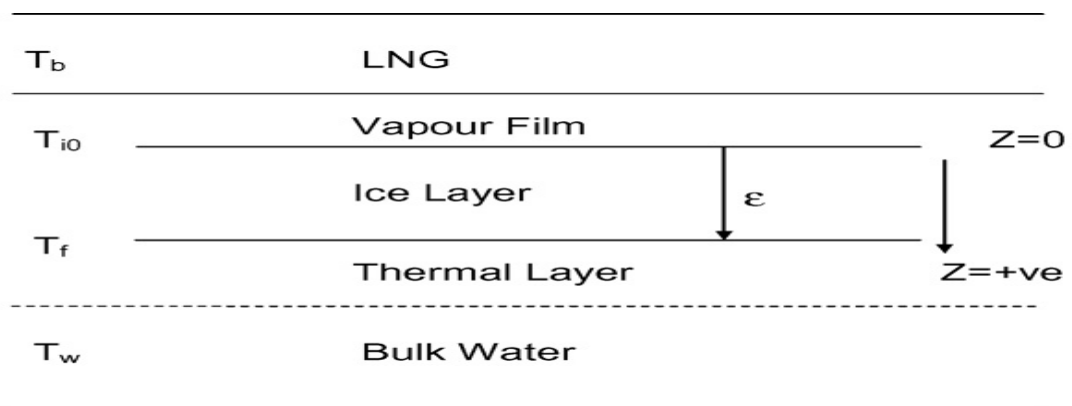


**Figure 2.6:** A schematic diagram of vaporization of LNG on water in the presence of ice. The author assumes that a third layer of ice including the thermal and vapour film layers now separates the spilled LNG from the bulk water. Heat transfer is assumed to take place in the  $z$ -direction as shown (Vesovic, 2007).

transfer between LNG and water. The author demonstrates in one result (see Figs. 2.7 and 2.8) that the ice surface temperature decreased rapidly within a few seconds at the beginning (see Figure 2.7) with corresponding increase in ice thickness (see Figure 2.8). A decrease in the surface temperature reduces the heat flux into the LNG pool. His preliminary simulation results for transition boiling revealed that the overall effect of transition boiling would be to increase the vaporization rate, an observation that was in tune with experimental observation (Vesovic, 2007; Valencia-Chavez, 1979).

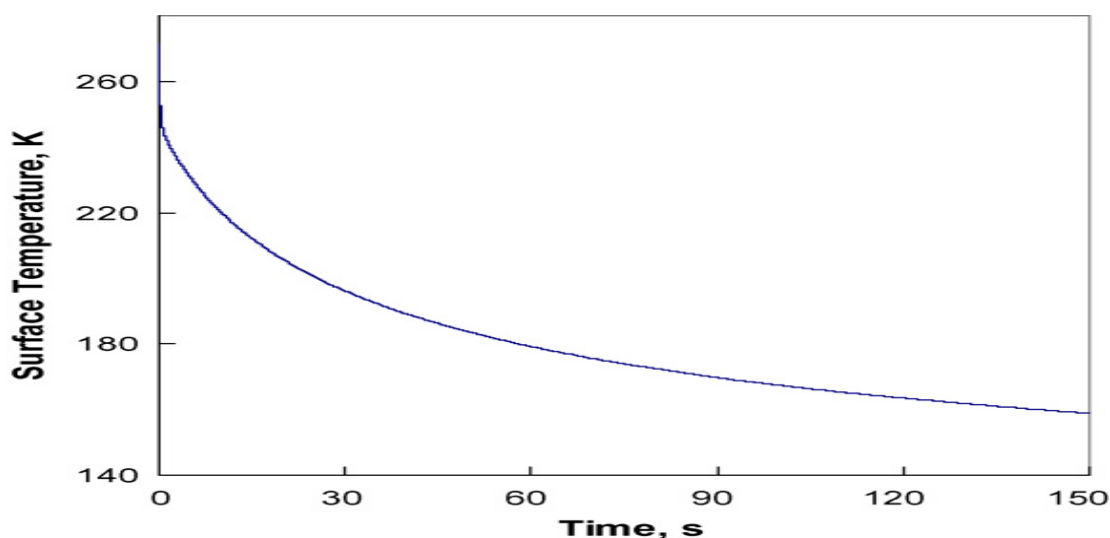


**Figure 2.7:** Profiles of the surface temperature of ice following an LNG spill (Vesovic, 2007).

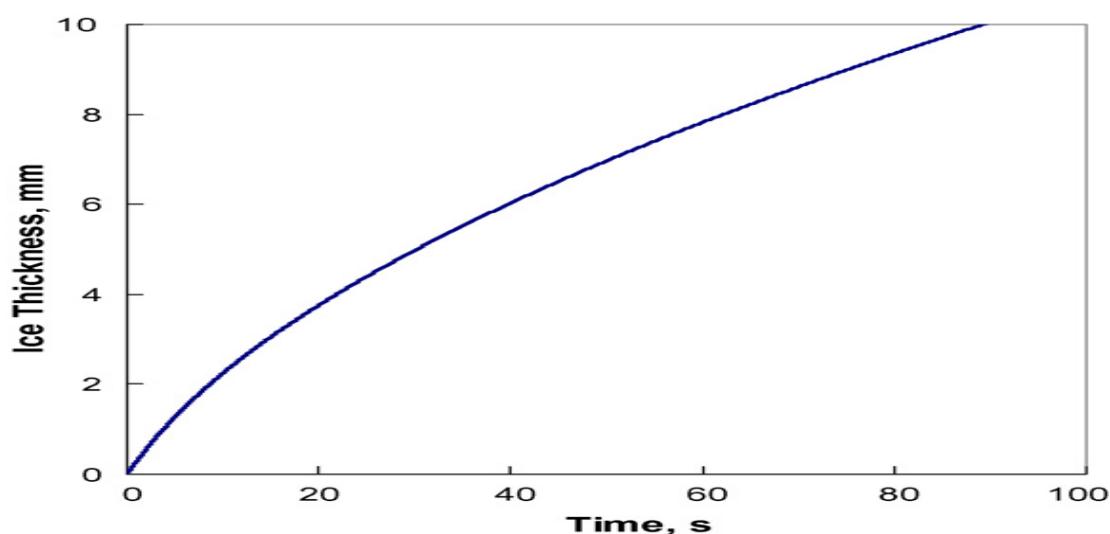


**Figure 2.8:** Profiles of the thickness of ice layer following the onset of nucleate boiling (Vesovic, 2007).

In addition to compositional effects, some of the identified literature dealing with LNG spills on water indicate that there could be an effect of LNG-water turbulence on vaporization rate (Burgess et al., 1970a; Burgess et al., 1970b; Burgess et al 1972; Feldbauer et al., 1972; Boyle and Kneebone 1973). As a result, modelling investigations on the effect of LNG-water turbulence on liquid vaporization rates have been reported recently (Hissong, 2007; Zubairu, 2011; Morse and Kytomaa, 2011). LNG-water turbulence studies was pioneered by Hissong (2007) when he presented a model that describes the effects of LNG–water turbulence on vaporization rate. Hissong’s model (Hissong, 2007) includes a fundamentally based framework with model formulations based on some of the largest LNG spills tests (Burgess et al., 1970a; Burgess et al., 1970b; Burgess et al 1972) on water. The value calculated for the heat transfer coefficient was obtained by applying a “turbulence factor” to the value obtained from correlations for quiescent film and transition boiling (Klimenko, 1981; Conrado and Vesovic, 2000). Figure 2.9 shows the turbulence factors calculated by Hissong (2007) for

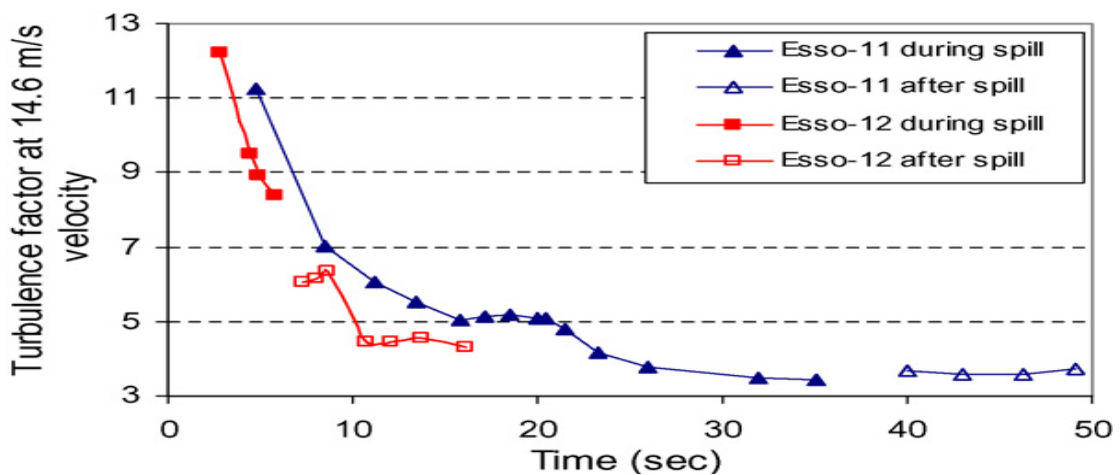


**Figure 2.9:** Turbulence factors calculated by Hissong (2007) for the Esso tests (Hissong, 2007)

the Esso tests (May et al., 1973). The knowledge of turbulence factors enables the calculation of the effect of LNG-water turbulence on vaporization process via an improved estimation of the film and transition boiling heat transfer coefficients between the fluid and substrate.

Zubairu (2011) also investigated LNG-water turbulence including the heat transfer mechanism involved in LNG spill on confined water surfaces. Zubairu

(2011) derived turbulence factors using experimental results from Valencia-Chavez (1978) and Boe (1998). One main conclusion of Zubairu's study (Zubairu, 2011) was the evidence that vaporization of LNG is strongly affected by LNG-water turbulence. The range of turbulence factors derived from simulating the experimental data of Valencia-Chavez (1978) and Boe (1998) was between 2-4. The effect of LNG-water turbulence was most recently presented by Morse and Kytomaa (2011) when they measured the rate of vaporization of LNG and liquid nitrogen ( $\text{LN}_2$ ) floating on a water surface with different levels of turbulence intensity. Figure 2.10 is the experimental channel used by Morse and Kytomaa (2011). The tests demonstrated that the rate of vaporization depends significantly on the turbulence intensity of the water surface, and to a lesser extent on the thickness of the cryogenic liquid layer on the water surface. Figure 2.11 shows the experimental channel with some degree of turbulence being established at the interface between the cryogen and water.



**Figure 2.10:** Experimental channel test section used by Morse and Kytomaa (2011).

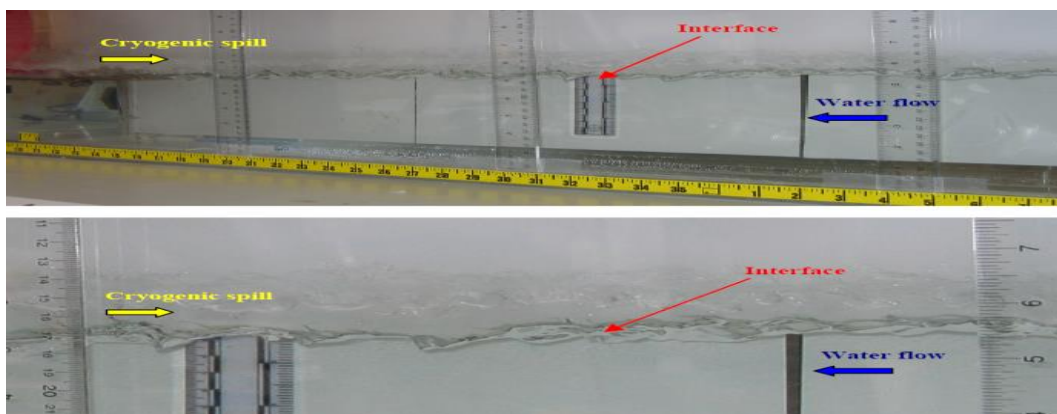


**Figure 2.11:** Water channel test section showing some detail of liquid-water turbulence at the interface between the cryogenic liquid and the substrate (Morse and Kytomaa, 2011).

The above description has revealed the characteristics of LNG vaporization for spills on liquid substrates such as water. It is also clear that the rate of vaporization of LNG on water will be influenced by several factors including ice formation, chemical composition, type of cryogen, amount of cryogen spilled, boiling regime and LNG-water turbulence. Despite the efforts of the identified researchers (e.g. Conrado and Vesovic, 2000; Vesovic, 2007; Hissong, 2007; Zubairu, 2011; Morse and Kytomaa, 2011), some of the proposed methods still have unresolved issues that would require further analysis and/or possible model extensions. However, addressing these issues is outside the core objective of this chapter which is to critically assess vaporization studies involving liquid spills on penetrable substrates (see Section 2.1.2).

### **2.1.2 Spills on Penetrable Solid Substrates**

Previous workers in the LNG spill research field have not adequately addressed spill problems in which the solid substrates are porous or penetrable. This thesis is an effort at addressing some of these research needs by modelling the vaporizing flow behaviour of cryogenic liquids following spillages or leakages onto/into penetrable, solid substrates. The following fifteen studies are considered of most relevance to the present thesis problem and will be reviewed in this section: Drake and Reid (1975), Japan Gas Association (1976), Reid and Wang (1978), Reid (1980), Briscoe and Shaw (1980), Opschoor (1981), Jensen (1983), Moorhouse and Carpenter (1986), Cook and Woodward (1993), Takeno et al. (1994), Daish et al. (1998), Melhem (2006), Bae et al. (2007), Lee et al. (2011), and Choi et al. (2012). The main goal of the review is to identify relevant knowledge ‘gaps’ in the identified publications which the present project hopes to fill.

#### **Drake and Reid (1975)**

When a pool of cryogenic liquid is spreading and vaporizing on a penetrable, solid substrate, liquid penetration into the substrate is inevitable. Despite this incontrovertible fact, much of the classic work in this research field has apparently ignored liquid penetration rates when modelling or experimentally investigating the vaporization rate of cryogens on penetrable, solid substrates. A review of the work of Drake and Reid (1975) will attest to this neglect. Drake and

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Reid (1975) experimentally studied how LNG vaporizes on penetrable, solid substrates such as soil and sand. Their experiments involved the placement of a 30 cm Styrofoam cube with a cylindrical well (18 cm in diameter and 22 cm deep) on a load cell to directly monitor mass loss. There was evidence of liquid penetration into the soil. It was also evident from the small holes on the soil surface at the end of the vaporization process that penetration of the LNG occurred as the cryogen vaporized. A dry soil spill test indicated even stronger penetration effects than moist soils. The relatively lower penetration effects for moist soils is expected since ice formation occurred inside the substrate which blocked further liquid penetration. When a thin plastic sheet was placed 6 mm beneath the dry substrate surface, penetration was not possible whereas when the plastic sheet was removed penetration was again confirmed. The implication of these observations is that LNG vaporization on penetrable, solid substrates may involve significant penetration of the liquid into the substrate and the extent of liquid penetration is influenced by substrate properties including permeability and porosity. It is therefore useful to understand and predict the behaviour of cryogenics penetrating a solid substrate because there are many real circumstances where the results of such predictions would be useful (Melhem, 2006, Choi, 2012). For example, predictions of cryogenic liquid vaporization and penetration rates would be needed for safety evaluation of large membrane LNG ships under barrier leakages. Figure 2.12 shows the insulation layer of a typical membrane type LNG ships under primary and secondary barrier leakages. We can notice from the figure that the vertical panel – the so-called flat joint was initially penetrated by LNG (Bae et al., 2007).

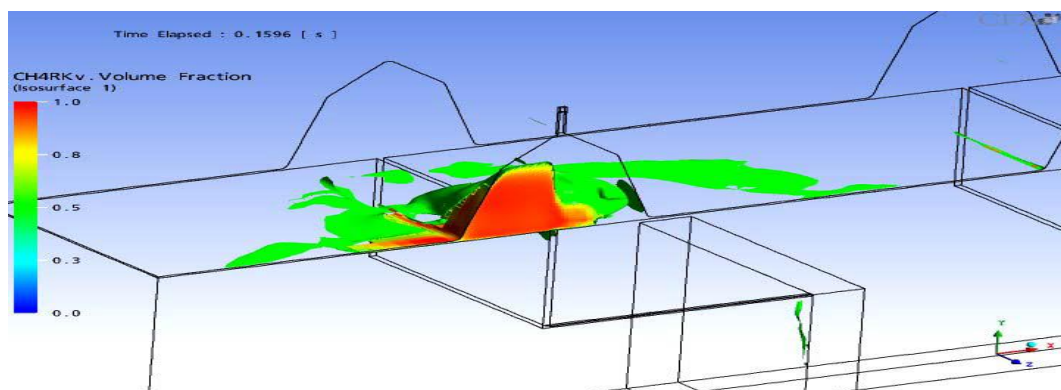


**Figure 2.12:** Volume of LNG vaporized following primary and secondary barrier leakages (Bae et al., 2007)

**Japan Gas Association (1976)**

About a year after the work of Drake and Reid (1975) was reported, the Japan Gas Association (1976) reported results of their experimental investigations aimed at studying the spreading and vaporization rate of LNG spilled on ground surfaces. Four continuous spillages of the cryogen onto soil were carried out with thermocouples in the dike detecting LNG spreading speeds. Measurements of the LNG pool were taken and weather conditions and moisture content of the soil were also recorded. From their experiments, the following expression for the vaporization rate of LNG was derived,

$$V_{JGA} = At^{-1/2} + B \quad (2.22)$$

where  $V_{JGA}$  is the vaporization rate in mm/min (regression rate),  $t$  is the time (s),  $A$  is a coefficient dependent on the soil properties and  $B$  is a coefficient dependent on convection and radiation. They gave values of the constants  $A$  and  $B$  as 90 and 0.5, respectively. As observed in the experiments of Drake and Reid (1975), LNG penetrated the ground as its pool spread and vaporized. This observed penetration behaviour was not investigated by the authors (Japan Gas Association, 1976).

**Reid and Wang (1978); Reid (1980)**

Reid and co-workers (Reid and Wang, 1978; Reid, 1980) measured the vaporization rates of LNG spilled on typical dike floor materials. This tests involved cutting slabs of solid materials into rectangular or square shapes and then placing these slabs into a Styrofoam form such that the exposed surface area of the slab will hold the pool of LNG after a spill. Different types of substrates were investigated including four types of concrete, soil, sand, pebbles, polyurethane and corrugated aluminium over soil. In the sand and soil tests, there was evidence of liquid penetration into the soil and sand. In the results for the sand runs, for example, the authors found significant tailing of the mass loss profiles towards the end of a run, suggesting liquid penetration. The moisture content also affected the vaporization and penetration rates significantly. Again, despite the evidence of liquid penetration observed in the studies (Reid and

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Wang, 1978; Reid, 1980), the authors did not report on the penetration behaviour of the liquid as it flowed and vaporised inside the porous medium. This modelling issue will be addressed in this thesis.

### **Briscoe and Shaw (1980)**

The vaporization behaviour of cryogenic liquid spills on land was investigated by Briscoe and Shaw (1980) in which they assumed that the heat transfer to the liquid is solely from the solid substrate. They used one-dimensional heat conduction theory in determining the heat flux to the cryogen. Their vaporization rate model had difficulties in accounting for the effect of ground roughness, porosity or penetration. A multiplicative correction factor was introduced to their vaporization rate equation in order to allow for these uncertainties. This vaporization rate equation was coupled to their pool spreading relationship given (Equations 2-4 of Briscoe and Shaw (1980)),

$$V_{Br\_sh} = \frac{q}{L_{vap}} = \left( \frac{\chi k_s (T_s - T_b)}{L_{vap} (\pi \alpha_s)^{1/2}} \right) \int_0^{r(t)} \frac{2\pi r' dr'}{(t - t')^{1/2}} \quad (2.23)$$

where  $\chi$  is the surface roughness factor and  $t'$  is the arrival time of the spreading liquid at radius,  $r'$ . Since the effects of ground penetrability or porosity was not explicitly discussed in their paper, we expect that the multiplicative correction factor introduced by the authors was an attempt to include these effects. However, predicting simultaneous vaporization and penetration will not be possible using the model of the form proposed by Briscoe and Shaw (1980).

### **Opschoor (1981)**

Opschoor (Opschoor, 1981) distinguished between cryogenic liquid spills on non-penetrable soil and those on penetrable soils and made efforts to model these two different spill configurations. Opschoor (1981) explained that the heat exchange process that occurs when a liquefied gas is spilled on a slab-like non-penetrable medium can be modelled by the Fourier's law of heat conduction (Carslaw and Jaeger, 1959). His heat conduction equation is the same as those used by

previous researchers (Drake and Reid, 1975; Japan Gas Association, 1976; Reid and Wang, 1978; Reid, 1980; Briscoe and Shaw, 1980) in addressing problems of liquid spills on penetrable and non-penetrable solid substrates. For spills on non-penetrable substrates, the surface heat flux calculation as used by Opschoor (1981) gives the following vaporization rate equation,

$$V_{ops} = \left( \frac{k_s (T_s - T_b)}{L_{vap} (\pi \alpha_s)^{1/2}} \right) t^{-1/2} \quad (2.24)$$

where  $T_s$  and  $T_b$  are the initial temperature and boiling temperature of the ground, respectively. In respect of spill on penetrable soil, a correction was made in Equation 2.24 to account for liquid penetration rates as well as ice formation within the capillaries of the porous medium. Opschoor's first limiting case assumed that the soil was dry with a weight fraction of zero while a second limiting case assumed that the soil was wet with a weight fraction of water in the soil greater than 8% (thus causing freezing of water in the pores). Opschoor compared his vaporization rate equation (Equation 2.24) for spills on non-penetrable substrates with that described by Drake and Reid (1975) in their experiments with dry sand. The vaporization rate based on Drake and Reid (1975) is given by,

$$V_{dreid} = 0.95t^{-1/2} \quad (2.25)$$

where,  $V_{dreid}$ , is the vaporization rate and  $t$  is time. When Opschoor (1981) reduced his own vaporization rate equation, he obtained the following vaporization rate expression,

$$V_{ops} = 0.12t^{-1/2} \quad (2.26)$$

Opschoor's equation (Equation 2.26) was found to be a factor of 8 lower than the experimentally determined one (Equation 2.25) by Drake and Reid (1975). Opschoor attributes these differences to the penetration of the liquefied gas into

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the soil. Ultimately, Opschoor (1981) proposed the following modified form of Equation 2.24 for the calculation of the rate of vaporization (with penetration) of liquefied gases on a dry penetrable soil,

$$V_{ops} = \left( \frac{8k_s(T_s - T_b)}{L_{vap}(\pi\alpha_s)^{1/2}} \right) t^{-1/2} \quad (2.27)$$

The influence of pore water (moisture content) on the vaporization of liquefied gases on moist penetrable soil was also investigated by Opschoor (1981) with experimental evidence indicating that the water in the pores of the substrate freezes and prevents further penetration of the liquid through the pores. The following relation was then introduced for the calculation of the rate of vaporization, with freezing effect incorporated in the model,

$$V_{ops} = \left( \frac{\lambda k_s(T_s - T_b)}{L_{vap}(\pi\alpha_s)^{1/2}} \right) t^{-1/2} \quad (2.28)$$

with  $\lambda$  and  $k_s$  given by Equations 2.29 and 2.30, respectively.

$$\lambda = 1 + 1.4x_w - 3x_w^2 \quad (2.29)$$

$$k_s = (\beta_1 \log_{10} x_w + \beta_2) 10^{\beta_3} \quad (2.30)$$

The estimation of the variables  $x_w$ ,  $\beta_1$ ,  $\beta_2$ , and  $\beta_3$  in Equations 2.29 and 2.30 are explained in Opschoor (1981). The correction term,  $\lambda$ , gives the effect of the freezing of the water in the substrate. From this review of Opschoor's work (Opschoor, 1981), it is clear that the vaporization rate of liquefied gases on non-penetrable, solid substrates can be calculated using the theory of heat transfer from a semi-infinite medium (Equation 2.24). However, calculating the vaporization rates of liquids on penetrable, solid substrates should account for penetration since porosity and permeability of the ground would cause the fluid

to flow through the substrate. Hence, realistic account must be taken of liquid penetration rates and this has not been adequately treated by Opschoor (1980) who only introduced, on the basis of experimental data, a factor of 8 into Equation 2.24 to account for liquid penetration. However, it was not clear how such a study could provide understanding and/or information on the actual penetration behaviour of the liquid inside the porous medium. In this work, we will extend the work of Opschoor (1981) by modelling the simultaneous vaporization and penetration of the cryogen through soil.

### **Jensen (1983)**

The vaporization of cryogenic liquids on penetrable, solid substrates was studied by Jensen (1983) who used a heat conduction equation for the estimation of vaporization rate. As expected, the vaporization rate was found to decrease with the square root of time. Although Jensen (1983) noted the importance of understanding liquid penetration rates and ice formation effects, he did not investigate these but rather studied another problem relating to an erroneous boundary condition invoked at heat exchange surfaces. Although this study by Jensen (1983) helped describe basic spreading and vaporization phenomena of a liquid pool on land, his omission of penetration rates are unwarranted since penetrable, solid substrates were used in his work. This present work extends Jensen's (Jensen, 1983) work by investigating liquid penetration rates.

### **Moorhouse and Carpenter (1986)**

Moorhouse and Carpenter (1986) performed measurements of the rate of vaporization from a pool of liquefied gas following spillages unto different types of penetrable and non-penetrable solid substrates. The vaporization behaviour follows the forms already proposed by previous researchers (Drake and Reid, 1975; Japan Gas Association, 1976; Reid and Wang, 1978; Reid, 1980; Shaw and Briscoe, 1980; Opschoor, 1981) and includes a modification that accounts for the presence of small particles. The following expression for the estimation of vaporization rate has been provided by Moorhouse and Carpenter (1986),

$$V_{MrCa} = \frac{4sa^2k_s\sqrt{\pi}(T_s - T_b)t^{-1/2}}{\sqrt{\alpha_s}} \frac{\exp(-\pi^2\alpha_s t)}{a^2} \quad (2.31)$$


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where  $s$  is the number of spherical solid particles immersed in the cryogen,  $a$  is the radius of the particles and  $T_s$  is the initial temperature of the particles. Equation 2.31 accounts for two main factors influencing vaporization rate, the first being the thermal properties of the substrate and the second being the contact area between the cryogen and the substrate surface. Higher values of vaporization rates have been observed when spillages have been made onto penetrable, solid substrates such as a layer of limestone chippings laid onto a base of polystyrene. In this case, LNG penetrated into the matrix pores. To partially account for penetrability effect, Moorhouse and Carpenter (1986) adopted the total surface area of the bed of particles as Equation 2.31 would indicate. For shallow depths of particulate materials, the authors found that LNG penetrated through the whole depth. From their observations for situations involving spills of LNG on finer particles or greater depth of particles, the authors cautioned that the use of Equation 2.31 may not be suitable. We agree with the authors and also suggest that a good step to evaluate such vaporization problems would be to model and predict the simultaneous vaporization and penetration of the liquid as it flows inside the porous media.

### **Cook and Woodward (1993)**

Several sub-models already reported in literature were integrated by Cook and Woodward (1993) to form a complete model of spreading and vaporization of a pool of liquefied gas on both penetrable and non-penetrable solid substrates. The pool spreading rate equation was based on a spreading relationship provided by Opschoor (1979). The net rate of heat flow into the pool came from various sources including conduction, convection, electromagnetic radiation, solar radiation and evaporation. The equations for calculating these heat sources were based on the work of Shaw and Briscoe (1978) and Fleischer (1989). Their vaporization model were validated by a number of experiments including those from Reid and Wang (1978) and Reid and Smith (1978a; 1978b). However, this model would be incapable of predicting liquid penetration into the solid substrate since such vaporizing flow effects are not incorporated in the program. This study will extend the work of Cook and Woodward (1993) by modelling liquid penetration behaviour upon spills of liquefied gases on penetrable, solid substrates.

### **Takeno et al. (1994)**

Laboratory experiments on the vaporization of liquid hydrogen ( $T_b=20$  K) and oxygen ( $T_b=90$  K) were performed and useful data on the vaporization rates were accumulated by Takeno (1994). To measure vaporization rates, the selected solid substrates were sand as well as limestone concrete. For the sand substrate, two different kinds of sand layers were used: a dry sand layer and a wet sand layer with 7 wt% water content. Figure 2.13 shows the results for the vaporization of liquid oxygen on the layer of dry sand which indicated that the liquid vaporized while constantly rolling up the upper section of the sand layer with the leading edge of the rolled-up sand layer further penetrating the sand substrate. By rolling-up, we imply that the sand layer increased from its initial depth as the sand layer is blown-up while the soaking liquid vaporizes. Takeno (1994) determined the vaporization rate in terms of liquid penetration velocity given as,

$$q_{tak} = \rho_s C_{ps} \Delta T (1 - \phi) V_L \quad (2.32)$$

where  $q_{tak}$  is the heat flux for liquid vaporization,  $\rho_s$  is the density of sand,  $C_{ps}$  is the specific heat capacity of sand,  $\phi$  is the void space ratio in the sand layer, and  $V_L$  is the penetration velocity. For spills on wet sand layer containing 7 wt% water, the rolling of the wet sand layer were observed to be different from those for the dry sand layer. It was also noted that the depth of the rolled-up sand layer was limited to 5 to 10 mm from the top surface of the sand layer, and the

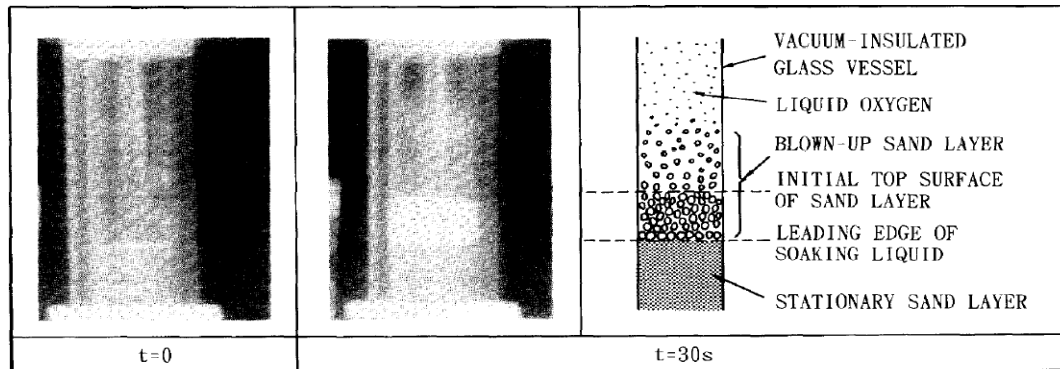


**Figure 2.13:** Observed results for the vaporization of liquid oxygen and a schematic diagram of dry sand layer during vaporization (Takeno et al., 1994).

layer under it remained stationary during vaporization. These results led to the conclusion that the frozen layer of water became a barrier preventing the liquid from penetrating or soaking deeper into the sand layer. Takeno et al. (1994) applied a model that solved this phase change (solidification) process. Overall, the study by Takeno (1994) neither studied how LNG would behave when spilled on the sand layer nor modelled the simultaneous vaporization and penetration of the cryogenics. This shortcoming as well as the others identified in Section 2.2 will be resolved in this work.

#### **Daisch et al. (1998)**

Daisch et al. (1998) presented a model that predicts the spreading and vaporization of cryogenic liquids on land. They validated their model using experimental data from Moorhouse and Carpenter (1986), Reid (1980) and Dienhart (1995). The effects of permeable substrate was investigated and a direct loss of liquid into the ground was based on Darcy's law (Bear, 1988). The heat flux into the pool has the following broad contributions: i) due to convection in the porous matrix; ii) due to conduction from the ground; iii) due to heat passing directly from the matrix into seeped liquid and; iv) due to bubbles which were formed in substrate recondensing in the pool. The various contributions become important under different circumstances. However, a model to realistically describe the vaporization of liquids penetrating a porous medium is lacking. Daisch et al. (1998) concluded that there remains much to be investigated concerning the behaviour of spills on penetrable, solid substrates, in particular the simultaneous penetration and vaporization aspects of the problem (Daisch et al., 1998). A modelling investigation that will describe both liquid vaporization and penetration rates are studied in this work.

#### **Melhem (2006)**

Melhem's (Melhem, 2006) approach to estimating the penetration of cryogenic liquids through porous media was achieved by neglecting physiochemical effects, vaporization effects, lateral spreading effects and heat conduction effects. His model assumes liquid penetration to behave as saturated piston flow influenced by gravity only. The penetration of liquid is dependent on fluid capacity (i.e.

saturation with liquid) of the substrate layer. The penetration depth of the liquid is given by the following linear relation proposed by Melhem (2006),

$$z_{pen} = \frac{k_0 t_z}{\phi} \quad (2.33)$$

where  $z_{pen}$  is the depth of liquid penetration in (m),  $t_z$  is the time of penetration (sec) and  $k_0$  is the hydraulic conductivity. Overall, Melhem's model (Melhem, 2006) was used to calculate the mass of the liquid that has penetrated into the soil, the number of moles of liquid that has penetrated into the soil, the soil volume occupied by the liquid, and the actual amount of chemical retained in the soil. But the study by Melhem (2006) did not incorporate heat conduction and/or vaporization effects and a more realistic model should include these effects, most especially for liquids of the cryogenic type. Furthermore, understanding the effect of vaporization on cryogenic liquid flow and its penetration behaviour is important. The present thesis will address these problems.

#### **Bae et al. (2007)**

This CFD study (Bae et al., 2007) investigated vaporization phenomenon following potential leakages of LNG within insulation layers of large LNG ships. They considered several leakage scenarios one of which involved leaking LNG into both primary and secondary barriers of a cargo tank and allowing the cryogen to be vaporized by nitrogen gas which has been injected through the insulation spaces of the cargo containment system. The effect of heat conduction and corresponding vaporization were not investigated. Temperature distributions of the inner hull of the ship following seepages from the secondary insulation layer were not investigated as well. In this thesis, all these problems will be addressed.

#### **Lee et al. (2011)**

The modelling, experiments and validation study by Lee et al. (2011) helped to understand the temperature distribution of the inner hull surface of a typical membrane type LNG cargo tank. Their study was aided by the use of a

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commercial CFD software (Lee, et al., 2011). Their model problem is based on a worst case secondary barrier leak scenario in which the LNG cargo tank is assumed to rupture at the bottom. In their study, they assumed that a secondary barrier leakage led to LNG flowing through a flat joint made up of glass wool porous material. They noted that since the temperature in the secondary insulation layer of the cargo tank was maintained at the normal temperature, the leaking LNG vaporizes as it flows downward through the porous regions of the insulation. However, no details were provided by the authors describing the simultaneous liquid vaporization and penetration that would occur inside porous regions of the cargo tank. The influence of boiling regime during the cooling of the inner hull of the ship was not investigated. In addition, the physical controls for simultaneous vaporization and penetration inside the insulation layers were not delineated. In this work, a model of vaporizing LNG flow was developed to address these inadequacies.

#### **Choi et al. (2012)**

In the work by Choi et al. (2012), the temperature distribution in the insulation layers of NO 96 and Mark III membrane LNG cargo tanks were studied. The study also involved an experimental determination of the thermal conductivity of some of the insulation materials used in LNG cargo containment systems. Furthermore, thermal analyses were conducted for scenarios involving LNG leakages into insulation boxes or layers of cargo tanks and the thermal safety was evaluated with the criterion that the inner hull of the ship was susceptible to cryogenic damage once the ductile-to-brittle transition temperature of  $-60\text{ }^{\circ}\text{C}$  has been reached. For the main materials that make up the insulation layers, which include perlite, plywood, and polyurethane foam, the temperature-dependent thermal conductivity was measured by the guarded hot plate method (Choi et al. 2012). Porous layers exist within the insulation layers and these layers provide preferential pathways for the LNG to flow when a cargo tank is breached or ruptured. The vaporizing flow behaviour of LNG through such porous regions were not modelled by Choi et al. (2012). Choi and co-workers (Choi et al., 2012) pointed this out in their paper by suggesting a need to address this problem. The work of Choi, et al. (2012) did not also consider the vaporization of an LNG pool on inner hull of the ship. All these problems will be addressed in this thesis.

## 2.2 Discussion

In Section 2.1.2 of this chapter, we reviewed fifteen studies developed over the past four decades which experimentally and/or computationally investigated the vaporization of LNG and other liquids following spills on penetrable, solid substrates and it was revealed that adequate consideration has not been given to the modelling of vaporizing flow problems that may result when LNG spill, penetrate and propagate down a porous medium. In other words, the existing studies do not provide adequate treatment of vaporizing LNG flow through porous media. In Table 2.3, a summary of all factors needing to be modelled including their author(s) is presented.

**Table 2.3** A summary of model developments involving vaporizing LNG flow on/through porous/penetrable media.

| <i>Year</i> | <i>Substrate Studied</i> | <i>Chemical Studied</i> | <i>Factor (s) Needing to be Modelled</i>                                       | <i>Literature/ Author</i>      |
|-------------|--------------------------|-------------------------|--------------------------------------------------------------------------------|--------------------------------|
| 1978        | Land                     | LNG                     | Vaporization rate                                                              | Reid and Wang (1978)           |
| 1980        | Land                     | LNG                     | Vaporization rate                                                              | Reid (1980)                    |
| 1980        | Land                     | Cryogenes               | Spreading rate and vaporization rate                                           | Briscoe and Shaw (1980)        |
| 1981        | Soil                     | LNG                     | Spreading and vaporization rates, ice formation effect                         | Opschoor (1981)                |
| 1983        | Land                     | Cryogenes               | Spreading and vaporization rates                                               | Jensen (1983)                  |
| 1986        | Land                     | Cryogenes               | Spreading and vaporization rate, small particle effects, compositional effects | Moorhouse and Carpenter (1986) |
| 1993        | Land                     | Cryogenes               | Spreading and vaporization rates                                               | Cook and Woodward (1993)       |
| 1994        | Ground                   | LH2, LOX                | Vaporization rate, penetration                                                 | Takeno et al. (1994)           |
| 1998        | Land                     | Cryogenes               | Vaporization rate                                                              | Daisch et al. (1998)           |
| 2006        | Soil                     | Cryogenes, Chemicals    | Penetration without heat transfer and vaporization                             | Melhem (2006)                  |
| 2007        | Glass wool               | LNG                     | Vaporization                                                                   | Bae et al. (2007)              |
| 2011        | Glass wool               | LNG                     | Vaporization                                                                   | Lee et al. (2011)              |
| 2012        | Glass wool               | LNG                     | Vaporization                                                                   | Choi et al. (2012)             |

It is clear from Table 2.3 that a description of the simultaneous vaporization and penetration of LNG through porous media is still lacking. In our specific context, unresolved is a description of LNG penetration rates in the event of secondary barrier leakages of large membrane tank LNG ships. In addition, an accurate prediction of LNG penetration rates should include both the effects of heat conduction (heat transfer) and boiling regime. All of the ‘gaps’ identified in Sections 2.1.2 and 2.2 will be addressed in this thesis.

## **2.3 Conclusion**

In this chapter, we have discussed the phenomena of vaporization following spills of cryogenic liquids on a variety of substrates. Appropriate equations, theory and correlations for characterizing such flows were identified and discussed. The first review (see Section 2.1.1) considered the spreading and vaporization of a pool of liquid following accidental spillages of the cryogen on liquid substrates such as water. The second review (see Section 2.1.2) is a state-of-the-art survey of studies dealing with cryogenic liquid spills on solid substrates such as penetrable soil. We specifically reviewed fifteen studies relating to the later after which it became evident that the reviewed studies contain gaps and limitations in their analyses of vaporizing flow and/or penetration aspects of the problem. Most of the reviewed studies consistently agree that a spill or a leak of a liquid on penetrable, solid substrate will result in liquid penetration through the pores of the substrate. It is also clear that the liquid may be subject to vaporization and the rate of vaporization will be controlled by the rate of heat transfer from the solid matrix into the flowing liquid. In addition, the heat transfer aspect of the vaporization problems needs to be improved to account for the influence of boiling regime. Thus, there are clearly opportunities to further develop and/or extend the work of previous authors on the subject of cryogenic liquid spills on porous or penetrable, solid substrates. In the present thesis, the following broad as well as specific research needs and modelling issues will be addressed:

- Rate of vaporization of a cryogenic liquid flowing through porous media
- Rate of penetration of a cryogenic liquid in porous media

- Effect of heat transfer on cryogenic liquid flow, vaporization and penetration rates
  - Effect of boiling regime on cryogenic liquid flow, vaporization and penetration rates
  - Rates of cooling, spreading and vaporization following barrier leakages of large LNG cargo tanks.
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# 3

## ***Modelling Liquid Flow Through Porous or Penetrable Media***

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In Chapter 2, we presented a review of existing studies which describe several aspects of cryogenic liquid vaporization following spills on a variety of substrates. Particularly, we discussed the problem of vaporizing fluid flow which occurs when liquids such as LNG are spilled on porous or penetrable media. We established that existing models have not adequately addressed these type of problem. In this chapter, we present a derivation of a model of vaporizing liquid flow through porous or penetrable media. The presented model is useful for the prediction of a variety of liquid flow events, such as may occur following LNG leakages into porous regions of large membrane LNG ships (Bae et al., 2007; Choi, et al., 2012) or water penetration through superheated geologic formations (Chestnut, et al., 1992; Nitao, et al., 1992). In both situations, liquid actively flows downwards through a porous layer and a good understanding of the fate of this liquid is required for accurate risk-based safety analysis of these systems.

The presentation in this chapter will involve five steps. First, we present the partial differential equations (PDE) governing the vaporizing liquid flow phenomena. Second, we present the numerical solution of the vaporizing flow model. Third, we present results of sample problems including a presentation of the relevant parametric trends. Fourth, we verify the model by comparing with its analytical solution in order to ensure that the idealized model is fully operational. In the fifth and final step, we draw conclusions with suggestions for further model development. The model is simple but its simplicity at this stage is

deliberate since the objective here is to obtain realistic estimates and to provide a good platform from which further model development can be launched.

### 3.1 Liquid Flow Modelling

Let us assume that the liquid is flowing, under the influence of pressure, through a porous medium. The porous media is assumed to be liquid-wetable, we ignore capillary effects and assume that the vapour generated will disperse sufficiently fast that no build-up of pressure will occur. The velocity of the flow,  $u$ , is given by the Darcy law (Bear, 1988). We further assume that the porous medium is at some temperature  $T$ , which is, for the sake of argument maintained at some constant value. In this scenario, the heat released from the porous matrix and leading to the vaporization of the liquid is constant. The problem is to find at what depth,  $z$ , all of the liquid will vaporize. A schematic diagram describing the problem is shown in Figure 3.1. Liquid flows downwards through the porous media of uniform initial temperature,  $T_{s0}$ , depth,  $Z$ , and lateral distance,  $X$ . The heat conduction in the solid matrix is considered strictly lateral in the “ $x$ ” direction. The boiling temperature of the vaporizing liquid remained constant throughout the liquid flow duration.

### 3.2 Model Assumptions

The model formulation is based on the following simplifying assumptions:

1. The porous layer is homogenous and isotropic.
  2. The liquid boils solely in the film boiling regime.
  3. All physical properties of the liquid and substrate are constant.
  4. The liquid flows at its boiling temperature,  $T_b$ .
  5. The heat source from the solid matrix is infinite.
  6. The liquid flow is strictly one-dimensional, in the “ $z$ ” direction.
  7. The pores of the porous media are initially air-saturated.
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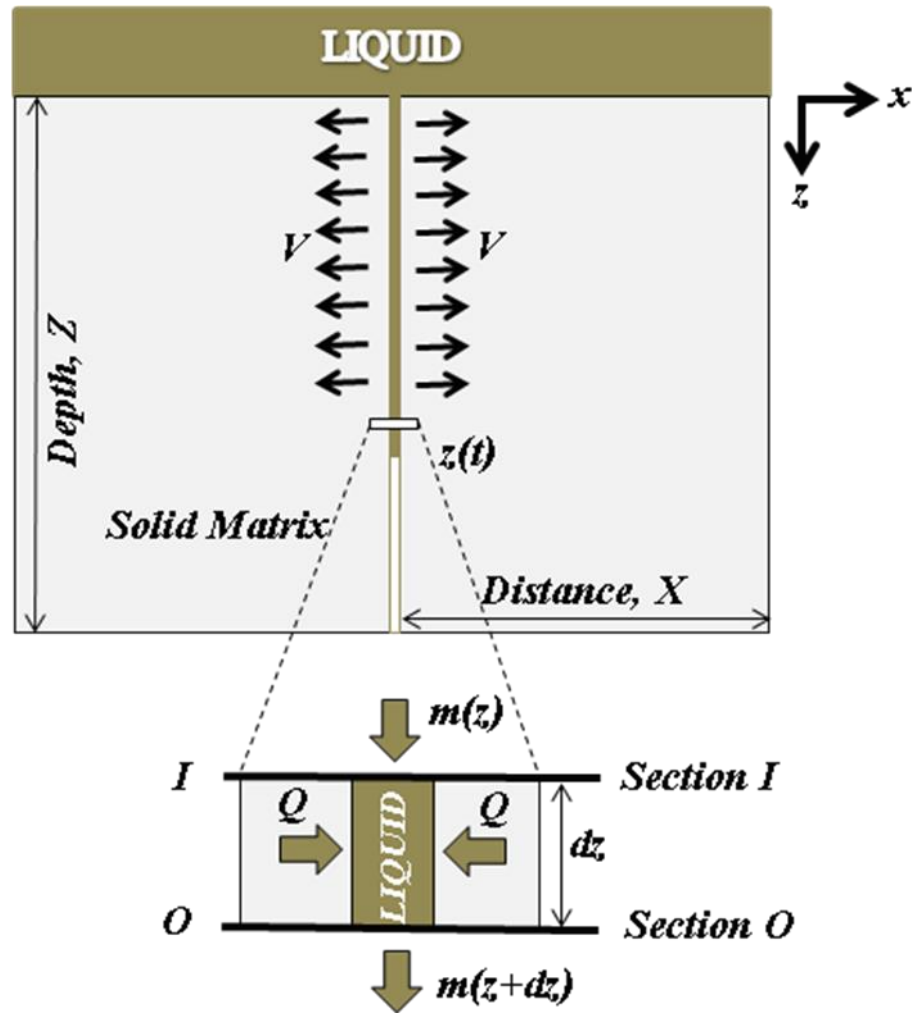


**Figure 3.1:** Schematic of the problem of vaporizing flow of fluids through porous or penetrable media. The dimensions of the porous layer, include a magnified portion of the diagram showing the control volume used in the modelling study.

### 3.3 Model Formulation

In deriving the liquid flow model (or Advection-Conduction Model), we first introduce a parameter,  $\phi$  to denote the amount of liquid that has vaporised (hereafter referred to as the “vapour mass fraction”) up to a point,  $z$ . Let us now consider an infinitesimal volume of a porous medium,  $dV$ , of size  $dz$  and cross-sectional area  $A$  (see Figure 3.1). The heat released by the solid matrix within this volume will be,

$$Q = qA_{\text{contact}} \quad (3.1)$$

where  $q$  is the heat flux ( $\text{W/m}^2$ ) and  $A_{\text{contact}}$  ( $\text{m}^2$ ) is the area of contact between the liquid and solid. Hence,  $A_{\text{contact}}$  would be the total surface area of the open pores. The amount of liquid vaporized per unit time, per unit volume of porous material by this heat would be,

$$\dot{m}_{\text{vap.}} = \frac{q A_{\text{contact}}}{L_{\text{vap}} V_{\text{bv}}} = \frac{q}{L_{\text{vap}}} A_{\text{s,specific}} \quad (3.2)$$

where  $A_{\text{s,specific}}$  ( $1/\text{m}$ ) is the specific surface area of the porous material (e.g.  $A_{\text{pores}}/V_{\text{bv}}$ ), a measurable quantity,  $V_{\text{bv}}$  is the bulk volume of the porous medium ( $\text{m}^3$ ) and  $L_{\text{vap}}$  is the latent heat of vaporization ( $\text{J/kg}$ ). The amount of liquid that flows per unit time through a volume  $dV$  is given by,

$$\dot{m} = \rho_l u A, \quad (3.3)$$

where  $\rho_l$  is the density of the liquid. Recalling the parameter  $\varphi$ , which we have used to denote the amount of liquid that has vaporized up to point  $z$ . Then the mass balance over the element  $dV$  will lead to

$$\dot{m}_{\text{vap.}} dV = \dot{m} d\varphi \quad (3.4)$$

$$\rho_l u L_{\text{vap}} \frac{d\varphi}{dz} = q A_{\text{s,specific}} \quad (3.5)$$

The depth at which all of the liquid has vaporized can now be obtained by a simple integration of the Equation (3.5). Hence for a given heat flux,  $q$ , the maximum liquid penetration depth can be estimated by knowing the properties of the liquid and the properties of the porous material through which it is flowing. Please note that, in the particular problem of interest to us in this work, the solid matrix will not remain at a constant temperature indefinitely, but will instead cool down with time. In order to model such a scenario we need to consider the conservation of energy in the porous media through which a vaporizing liquid is flowing. In analogue with Equation (3.5), the problem can

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now be cast in terms of the time-evolution of the vapour mass fraction,  $\varphi$ . The resulting equation that governs the amount of liquid that will vaporize up to a depth,  $z$  is now given as,

$$\rho_l L_{vap} \frac{\partial \varphi}{\partial t} + \rho_l u L_{vap} \frac{\partial \varphi}{\partial z} = q A_{s,specific} \quad (3.6)$$

The vapour mass fraction,  $\varphi$ , is now a function of both time and depth. Dividing all terms in Equation 3.6 by  $\rho_l L_{vap}$  gives Equation (3.7),

$$\frac{\partial \varphi}{\partial t} + u \frac{\partial \varphi}{\partial z} = S \quad (3.7)$$

where  $u$  is the Darcy velocity (Bear, 1988) and the term  $S$  (hereafter referred to as the source factor which represents the reciprocal of the maximum time ( $t_{max}$ ) required for the vaporizing flow to reach steady-state) is given by,

$$S = \frac{q A_{s,specific}}{\rho_l L_{vap}} \quad (3.8)$$

A simple analytical solution involving a combination of Equation (2.1) of Chapter 2 (without the gravity term) and Equations (3.7) (at steady-state) and (3.8), enables (after rearranging the terms) calculation of the maximum liquid penetration depth,  $z_{max}$ , at which all of the liquid has completely vaporized (i.e. when vapour mass fraction=1). This maximum penetration depth can be expressed as,

$$z_{max} = \frac{K_s \rho_l L_{vap} (P_2 - P_1)}{\mu q Z A_{s,specific}} \quad (3.9)$$

where  $K_s$  ( $m^2$ ) is the permeability of the porous media,  $P_1$  (Pa) and  $P_2$  (Pa) are inlet and outlet pressures, respectively,  $\mu$  (Pas) is the liquid dynamic viscosity and  $Z$  (m) is the thickness of porous medium. The heat flux,  $q$ , can be estimated from the Newton's Law of Cooling (Bird, et al., 2002) given as Equation (2.2) of Chapter 2 but repeated here for convenience,

$$q = h(T - T_b) \quad (3.10)$$

where the symbol  $h$  (W/m<sup>2</sup>/K) stands for the heat transfer coefficient,  $T$  (K) is the temperature at the surface of the porous matrix and  $T_b$  is the boiling temperature of the liquid. Please note that the heat transfer coefficient,  $h$ , is determined by using Equation (2.11) of Chapter 2, if the mode of boiling is film boiling, and by Equation (2.10) of Chapter 2, if a shift to transition boiling has occurred. For nucleate boiling, the heat flux is estimated using Equation (2.3) of Chapter 2, again repeated here for convenience,

$$q = -k_s \frac{\partial T}{\partial x} \quad (3.11)$$

where  $k_s$  (W/m/K) is the thermal conductivity of the porous matrix, and  $\partial T/\partial x$  is the lateral temperature gradient in the porous matrix.

### 3.4 Darcy (Flow) Velocity ( $u$ )

In this work, liquid flow is modelled using Darcy's law (Bear 1988; Kaviani, 1999; Bear and Bachmat, 1991) given by,

$$u = -\frac{K_s}{\mu} \nabla P \quad (3.12)$$

where  $\nabla P$  is the pressure gradient. Because we are assuming a pressure driven-flow, the gravity term,  $\rho g$ , indicated in Equation (2.1) of Chapter 2 is hereafter ignored.

### 3.5 Specific Surface Area ( $A_{s,specific}$ )

In Section 3.3, we defined the specific surface area,  $A_{s,specific}$ , as the ratio of the total surface area of the open pores,  $A_{pores}$ , and the bulk volume of the porous layer,  $V_{bv}$ . Although the specific surface area is a measurable quantity, the

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measurements are difficult and sparse. In the present work, we have made use of the permeability of the porous material, which is available in literature (Bear, 1988). The permeability of the porous media depends on porosity and structure of the porous medium. This is an area of active research and a number of theoretical, semi-empirical and empirical relationships are available (Carman, 1937; Bear, 1988). One of the best known of these relationships, the Kozeny-Carman equation (Carman, 1937; Dullien, 1979; Bear, 1988), which relates permeability to the porosity and the specific surface area can be expressed as,

$$K_s = k_k \frac{\phi^3}{A_{s,specific}^2} \quad (3.13)$$

where  $\phi$  is the porosity and  $A_{s,specific}$ , the specific surface area. The constant,  $k_k$ , takes into account the shape and orientation of solid material and configuration of the fluid path. Traditionally the value of  $k_k=1/5$  gives best agreement with experiment. Most of the experimental data used in a comparison refers to granular material. For flows through fibrous materials, a number of workers (Bear, 1988) have shown that the ‘constant’  $k_k$  is a function of porosity and the Reynolds number. For the purposes of this work, we can estimate the value of  $k_k$  of (Vesovic, 2009),

$$\begin{aligned} k_k &= \frac{1}{6} & 0.8 \leq \phi \leq 0.93; \\ k_k &= \frac{1}{8} & \phi = 0.95; \\ k_k &= \frac{1}{10} & \phi = 0.97; \\ k_k &= \frac{1}{12} & \phi = 0.98 \end{aligned} \quad (3.14)$$

at low Reynolds numbers ( $Re < 1$ , Darcy regime). Hence, the knowledge of permeability of the porous media and its porosity will allow, through the use of Equations (3.13 and 3.14), the estimation of the specific surface area relevant to the heat transfer considerations.

### 3.6 Tortuosity ( $\tau$ )

Tortuosity is a concept used to characterize the actual paths, of varying cross-section, which the fluid must follow as it flows through porous media (Bear, 1988). A more straightforward definition of tortuosity would be a ratio of the actual length transversed by a liquid to the straight length of a porous medium in the direction of flow (Bear, 1988). Thus, tortuosity can be used to represent the departure of a porous layer from being composed of a bundle of straight capillary tubes (Bear, 1988) thereby enhancing liquid spreading and reducing the depth of liquid penetration. Several techniques have been proposed for the determination of tortuosity (Carman, 1937; Bear, 1988). Mortensen et al. (1998) proposed the introduction of the tortuosity parameter into the Kozeny-Carman equation (Equation 3.13) through the replacement of the Kozeny “constant”,  $k_k$  with the following relation,

$$k_k = \frac{1}{k_{dsf} \left( \frac{L_{eff}}{L_{st}} \right)^2} \quad (3.15)$$

where  $k_{dsf}$  is a dimensionless shape factor,  $L_{eff}$  is the effective length through the porous medium and  $L_{st}$  is the straight length of the porous medium. In this case, Equation 3.13 becomes,

$$K_s = \frac{1}{k_k (\tau)^2} \frac{\phi^3}{A_{s,specific}^2} \quad (3.16)$$

where the tortuosity,  $\tau$ , is given by,

$$\tau = \frac{L_{eff}}{L_{st}} \quad (3.17)$$


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It is clear from the above equations that tortuosity is essentially introduced into the Kozeny-Carman equation (Equation 3.13) through the Kozeny constant relation (Equation 3.15).

### 3.7 Heat Transfer Coefficient ( $h$ )

In Section 2.0.3.1 of Chapter 2, we discussed the importance of heat transfer coefficient and boiling regime in the solution of vaporizing flow problems. However, it is assumed in this chapter that the flowing liquid will only boil in the film-boiling regime because we have modelled LNG using the properties of liquid methane, which is known to boil under film boiling (Conrado and Vesovic, 2001; Vesovic, 2007). We also discussed in Chapter 2 some of the most important correlations reported in literature for the calculation of the film-boiling heat transfer coefficient. As already indicated in Chapter 2, we will be using the correlations of Klimenko (1981) for the estimation of film boiling heat transfer coefficients as suggested by other authors (Conrado and Vesovic, 2000; Vesovic, 2007; Zubairu, 2011). The film-boiling heat transfer coefficient  $h_{film}$  as proposed by Klimenko (1981) is controlled by the properties of the various phases involved.

### 3.8 Numerical Solution

The numerical solution to the differential equation presented in Section 3.3 can be achieved by using one of the three major numerical discretisation methods, namely: (a) the finite volume method, (b) the finite element method and (c) the finite difference method (Chung, 2002). The end goal in discretizing the partial differential equation (PDE) (Equation 3.6 or 3.7) using any of the mentioned methods is to transform the PDE into a simpler discrete form that is amenable to computational solution. For the purpose of this work, the explicit finite difference method of solution is applied (Smith, 1985). Section 3.8.1 gives a brief introduction of the finite difference method.

#### 3.8.1 Finite Difference Method

The finite difference method is considered the simplest of the three major numerical methods used for the solution of differential equations and it involves the replacement of the partial derivatives of a partial differential equation with

its corresponding finite difference approximation. The resulting single algebraic equation or in some cases a system of algebraic equations can be solved to provide an approximate solution to the original partial differential equation at selected points of a solution domain. The assumptions made for this type of solution is based on the concept that the axes representing the independent variables in the solution domain can be divided into a number of intervals. This intervals or grids are usually regular with each numbered consecutively with identifiers (usually appearing as a set of indices). Since the grid points define the set of locations where the discrete solution is computed, two key parameters of the grid in respect of our derived partial differential equation (Equation 3.7) are  $\Delta z$ , the local depth between adjacent points in space, and  $\Delta t$ , the local distance between adjacent time steps.

The core idea of a finite difference method is to replace continuous derivatives with so-called difference formulas. In respect of our model (specifically, Equation 3.7), if the actual solution to Equation 3.7 is given by  $\varphi(z,t)$ , then the approximations in the nodes of the computational grid will be given by  $\varphi_i^k = \varphi(z_i, t_j)$ . Before further explaining the finite difference discretization technique applied to Equation 3.7, let us briefly describe three possible finite-difference approximation formulae that can apply to the derivative relating to our spatial variable,  $z$ . The forward difference formula is given by Equation 3.18a, the backward difference formula by Equation 3.18b and the central difference formula by Equation 3.18c. The expressions (i.e. Equations 3.18a-3.18b) can be extended to the derivative in time.

$$\frac{\partial}{\partial z} \varphi(z,t) = \frac{\varphi_{i+1}^k - \varphi_i^k}{\Delta z} + O(\Delta z) \quad (3.18a)$$

$$\frac{\partial}{\partial z} \varphi(z,t) = \frac{\varphi_i^k - \varphi_{i-1}^k}{\Delta z} + O(\Delta z) \quad (3.18b)$$

$$\frac{\partial}{\partial z} \varphi(z,t) = \frac{\varphi_{i+1}^k - \varphi_{i-1}^k}{2\Delta z} + O(\Delta z^2) \quad (3.18c)$$


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where the superscript  $k$  and the subscript  $i$  are the initial time and space counters, respectively. With regard to our time derivative, a central difference formula for a partial time derivative is given by Equation 3.18d. The order terms in the respective difference formulae suggest that the error incurred by the forward and backward difference formulae are of first order ( $O(\Delta z)$ ), while those for the central difference formulae (Equations 3.18c and 3.18d) are of second order in time and space ( $O(\Delta t^2)$ ,  $O(\Delta z^2)$ ),

$$\frac{\partial}{\partial t} \varphi(z, t) = \frac{\varphi_i^{k+1} - \varphi_i^{k-1}}{2\Delta t} + O(\Delta t^2) \quad (3.18d)$$

### 3.8.2 Discretization

The discretization of the partial differential equation derived for our vaporizing liquid flow formulation (Equation 3.7) is presented in this section. The spatial derivative is approximated using the backward difference formula. The choice of the backward difference formula is to avoid oscillation errors that would have arisen if, for example, a central difference formula was used to approximate the advection term (Smith, 1985). The discrete  $z$  are spaced in the interval between 0 and  $Z$ . As for our time discretisation, the derivative is approximated by applying the forward difference formula. The discrete time is also placed in the interval between 0 and  $t_{max}$  (equivalent to the total run time of the simulation). A Courant, Friedrich, Levy (CFL) condition (Smith, 1985; Chung, 2002) is the only restriction to the time step. An approximate solution to Equation 3.7 can be obtained at a finite set of  $z$  and  $t$ . Thus, the vapour mass fraction,  $\varphi_i^{k+1}$ , at the  $i$ th grid point and  $(k+1)$ th time-step, would take the form of Equation (3.19),

$$\varphi_i^{k+1} = \varphi_i^k - \frac{u\Delta t}{\Delta z} (\varphi_i^k - \varphi_{i-1}^k) + S\Delta t \quad (3.19)$$

In order to ensure stability of the solution and to avoid the drawback of explicitly treating the convection term of Equation (3.19), the CFL condition must be imposed.

### 3.8.3 Consistency, Stability and Convergence Testing

The numerical solutions are usually investigated for consistency, stability and convergence and these three properties are usually tested whenever the finite difference method of analysis is applied. In this section, we will be testing for consistency, stability and convergence of Equation 3.7.

#### 3.8.3.1 Consistency Testing

A finite difference approximation to a partial differential equation is said to be consistent if in the limit of the step size(s) going to zero, the original partial differential equation is recovered, i.e. the truncation error approaches zero (Smith, 1985). The finite difference equation (Equation 3.19) was analysed for consistency with the partial differential equation (Equation 3.7). One way to check for consistency is to express each term in the finite difference equation by a Taylor series with a particular base point, then simplifying the resulting equation in order to yield the exact form of the truncation error of the complete finite difference equation. Thereafter, consistency can then be investigated by letting the grid spacing go to zero. For the present case, we will be making a simplistic test for consistency of the finite difference method. Rearranging Equation (3.19), we have Equation 3.20,

$$\frac{\varphi_i^{k+1} - \varphi_i^k}{\Delta t} + u \frac{\varphi_i^k - \varphi_{i-1}^k}{\Delta z} = S \quad (3.20)$$

Equation 3.20 will reproduce the original equation (Equation 3.7) as  $\Delta z \rightarrow 0$  and  $\Delta t \rightarrow 0$ . This can be shown by noticing that the respective forward and backward difference formula and their corresponding partial derivatives including their respective first order terms (Equations 3.21a and 3.21b) are equivalent.

$$\frac{\varphi_i^{k+1} - \varphi_i^k}{\Delta t} = \left( \frac{\partial}{\partial t} \varphi(z, t) \right) + O(\Delta t) \quad (3.21a)$$

$$\frac{\varphi_i^{k+1} - \varphi_i^k}{\Delta z} = \left( \frac{\partial}{\partial z} \varphi(z, t) \right) + O(\Delta z) \quad (3.21b)$$


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Hence Equation (3.20) can be written as,

$$\left(\frac{\partial}{\partial t}\varphi(z,t)\right) + O(\Delta t) + u\left(\frac{\partial}{\partial z}\varphi(z,t)\right) + O(\Delta z) = S \quad (3.21c)$$

and as  $\Delta z \rightarrow 0$  and  $\Delta t \rightarrow 0$ , equation (3.21c) reduces to the original partial differential equation given Equation (3.21d),

$$\left(\frac{\partial}{\partial t}\varphi(z,t)\right) + 0 + u\left(\frac{\partial}{\partial z}\varphi(z,t)\right) + 0 = S \quad (3.21d)$$

Therefore, the proposed finite difference scheme for the numerical solution of Equation 3.7 is consistent.

### 3.8.3.2 Stability Testing

Stability is another property that must be checked when implementing a finite difference method. The concept of stability is explained thus: If the error is defined to be the difference between the computed solutions and the exact solution of the discrete approximation, then the finite difference method is stable if the error remains uniformly bounded for successive iterations (Smith, 1985). A number of methods exists for the stability analysis of a finite difference approximation of which the discrete perturbation method, the von Neumann method and the matrix method (Smith, 1985) are the most notable. For example, in the von Neumann method of stability analysis, the exact solution of the finite difference equation is obtained for the general Fourier component of a complex Fourier series representation of the initial property distribution. If the solution for the general Fourier component is bounded (either conditionally or unconditionally), then the finite difference equation is stable. However, if the solution for the general Fourier component is unbounded, then the finite difference equation is unstable. For this work, we will be taken an alternative approach to stability testing by directly applying the criteria based on the CFL condition (Smith, 1985; Chung, 2002). In the CFL concept, the time step must be less than a certain time in computer simulations in order to ensure that the

solution is stable. Thus our finite difference approximation given by Equation 3.19 has the following stability criteria,

$$\frac{u\Delta t}{\Delta z} < 1 \quad (3.22)$$

The CFL condition can be regarded as a necessary criterion for convergence of the explicit finite difference approximation (Equation 3.19).

### 3.8.3.3 Convergence Testing

The proof of convergence of a finite difference method can be visualised by considering the Lax Equivalence Theorem (Smith, 1985) which states as follows:

*“Given a properly posed linear initial-value problem and a finite difference approximation to it that is consistent, stability is the necessary and sufficient condition for convergence.”*

Having established the consistency and stability of the finite difference method used in the present study, the finite difference equation is convergent.

With a convergent finite difference equation, we can now check and select the optimum grid size for simulation purposes since this will ensure accuracy of the numerical results. In selecting the optimum grid size, we tried to check how small we need to make the grid sizes or rather how much we need to increase the number of grid points before we can trust the solution. In order to select the optimum grid size, several simulations were run for increasing number of grids from 100 grid points to 2000 grid points. We eventually selected a grid size of 12.5 millimetres (400 grid points) as the optimum grid size for the current simulations. Figure 3.2 shows a graph of our grid selection study. We also made a relative error,  $\zeta_i^{GPT}$  (%) plot based on a comparison of the numerical solution with the analytical solution. The relative error expression is given by,

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$$\zeta_i^{GPT} (\%) = \frac{|\varphi_i^{GPT} - \varphi_i^{ANALY-SS}|}{\varphi_i^{ANALY-SS}} * 100 \quad (3.23)$$

where  $\varphi_i^{GPT}$  is the vapour mass fraction based on the numerical solution using *GPT* number of grid points,  $\varphi_i^{ANALY-SS}$  is the analytical solution of Equation 3.7 based on the present steady-state case.

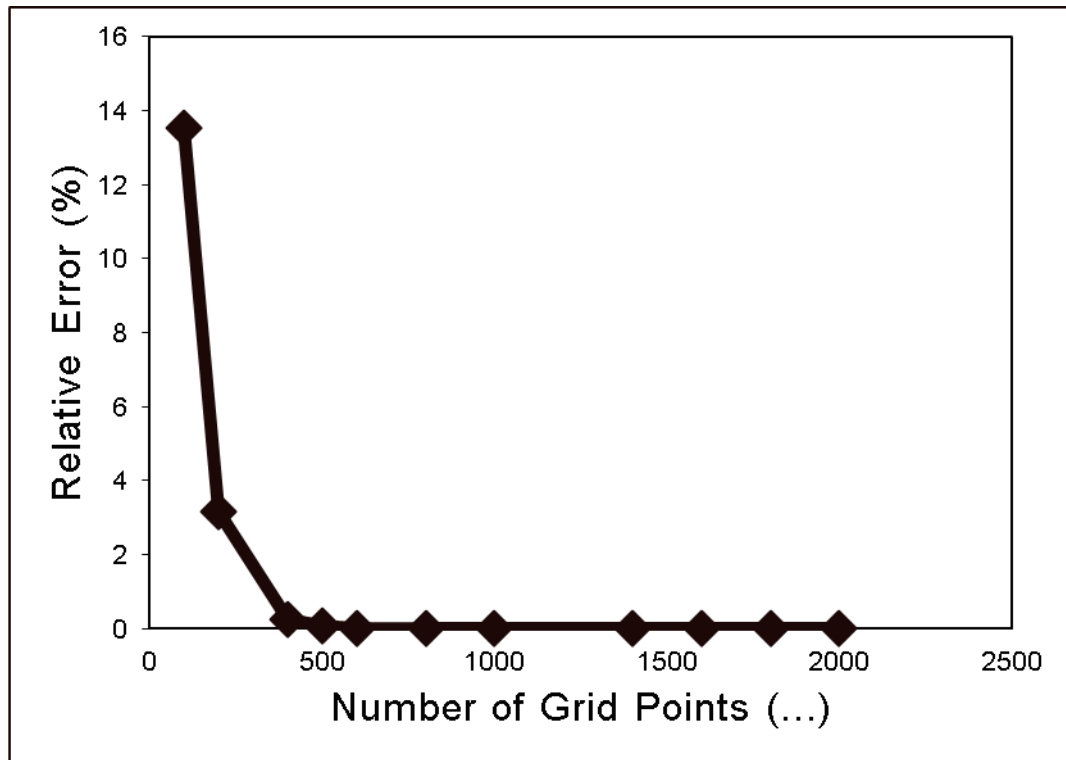


**Figure 3.2:** Relative error plot following a grid dependency study which aided the selection of the optimum grid size used for the simulation of the sample problems presented in this Chapter.

### 3.9 Model Verification

The liquid flow modelling work presented in this chapter assumes that the initial surface temperature of the porous matrix does not change throughout the vaporizing flow process. In this section, we will conduct a model verification study using Equation 3.23 in order to test and determine if the numerical solutions presented in Section 3.3 is accurately representing the mathematical model used for the description of the vaporizing flow phenomena. The verification methodology is simple as we are comparing the numerical model with its

analytical solutions which, because it is a steady-state case, can be readily derived. In what follows, an analytical solution is derived by first equating the partial time derivative of Equation 3.6 to zero,

$$\frac{\partial \varphi}{\partial t} = 0 \quad (3.24)$$

thus reducing Equation 3.6 to,

$$\rho u L \frac{d\varphi}{dz} = q A_{s,specific} \quad (3.25)$$

By integrating and rearranging Equation (3.25), we obtain a relationship that expresses the vapour mass fraction as a function of depth,

$$\varphi = \frac{q A_{s,specific}}{\rho L} \frac{z}{u} \quad (3.26)$$

where all the parameters on the right side of Equation (3.26) are as defined in Section 3.3. Equation (3.26) can then be used to calculate the maximum depth at which the liquid has completely vaporised (or the depth at which the vapour mass fraction is equal to one). By using the input specifications listed in Table 3.1 to 3.3, the results of the analytical solution is obtained and compared with those from the numerical solution. A comparison of the numerical and analytical solution in terms of vapour mass fraction as a function of depth was made. From Figure 3.3 the illustrated percent error of 0.04% suggests that the agreement between the numerical solution and the analytical solution is excellent. The error observed close to the vaporizing liquid front indicates that at these locations, more vaporization is occurring which would then affect the profiles of vapour mass fraction obtained using the numerical solution.

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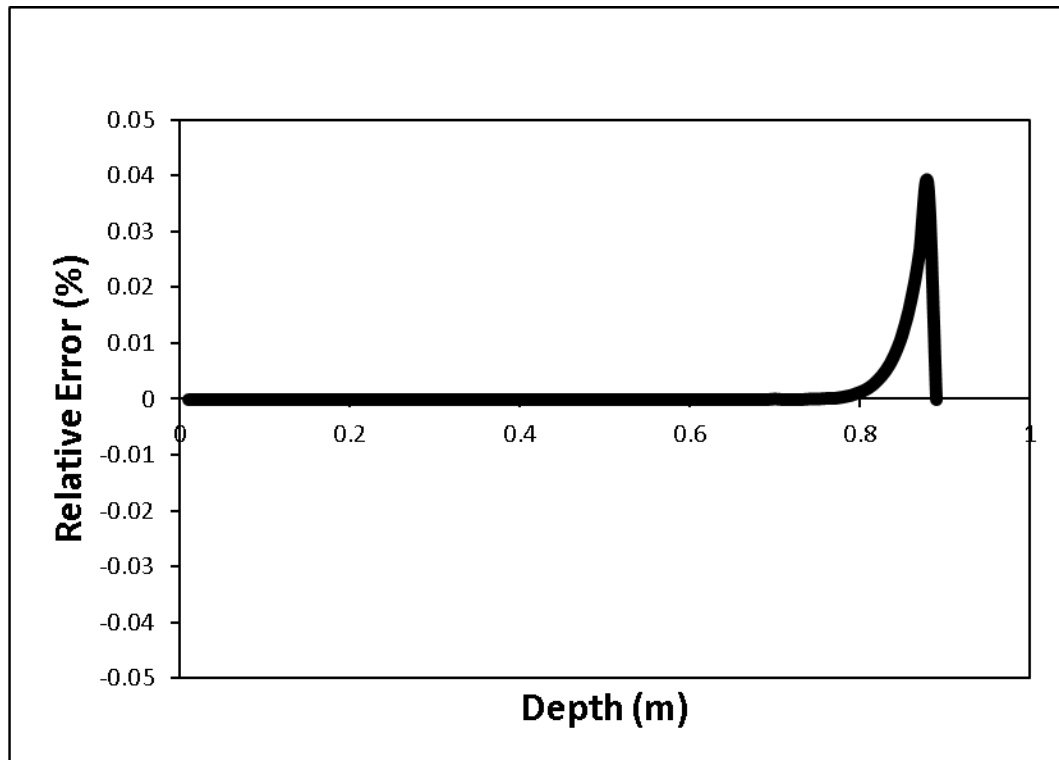


**Figure 3.3:** Liquid flow model verification – comparison of the numerical liquid flow model with its corresponding analytical solution: (a) vapour mass fraction versus depth, (b) percent error analysis.

### 3.10 FORTRAN Program

The solution of the system of differential and algebraic equations presented in this work were coded in FORTRAN. Figure A.1 of Appendix A is a flowchart showing the main program structure for the vaporizing liquid flow model. At the start of the liquid flow simulation, all initial liquid flow conditions and parameters are specified. The input data that could be provided at this step include properties such as: initial temperature of porous matrix, thermal properties of the porous matrix (thermal conductivity, thermal diffusivity) and geometry details of the porous media (e.g. depth, lateral distance), parameters describing the properties, type and geometry of liquid, time and space discretization parameters, and control parameters for simulation run (e.g. total run time). After the specification of input conditions and parameters, surface temperature of bounding substrate, heat flux to the liquid and the film boiling heat transfer coefficient can be calculated. The program also calculates the vaporisation rate of the flowing liquid. Finally, a knowledge of the specific surface area of porous material enables the calculation of the change in vapour

mass fraction. The program exits when the specified total simulation time has been reached. An appendix CD containing a full description of the source codes and algorithms is provided. Section 3.11 focuses on presenting the results obtained from simulating some sample problems using the derived model.

### 3.11 Simulation of Sample Problems

The vaporizing flow model presented in Section 3.3 of this chapter is now assessed by applying it to sample problems. The sample problems mainly simulate the vaporizing flow of LNG through porous or penetrable media. A description of how the liquid will evolve while flowing through the porous media under the influence of pressure is presented in Figure 3.4. For all simulations, LNG was injected at a constant rate into the porous medium from above as indicated in Figure 3.4. The LNG, being at its boiling temperature of 112 K, immediately vaporizes and a two-phase liquid-vapour zone develops within the porous medium. We call the leading edge of this two-phase zone or region as the “front” and the identified two-phase liquid-vapour zone as “Zone 1”. The distance

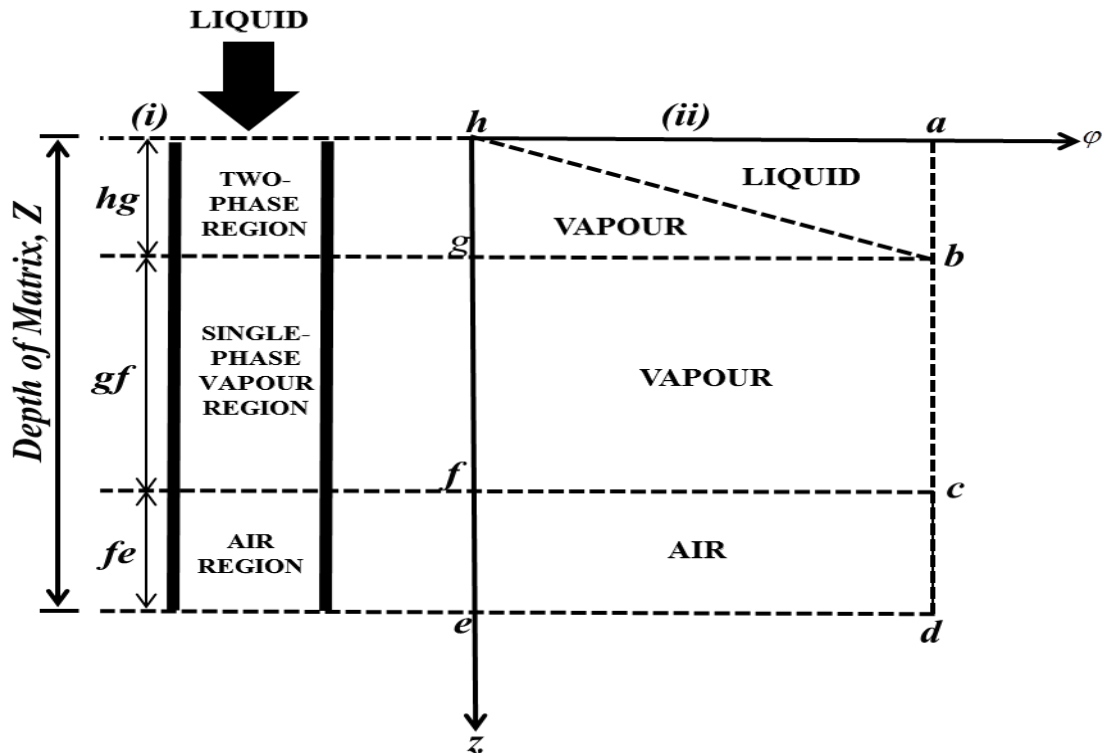


**Figure 3.4:** A schematic showing the evolution of the liquid and vapour in a porous medium in which a one-dimensional steady-state heat conduction model is applied.

between the liquid inlet position and the front (shown in Fig. 3.4 to correspond to the lines a-b and h-g or depth hg) is hereafter referred to as the “extent”. The extent of Zone 1 grows with time as the liquid penetrates and the front moves deeper into the porous medium. Figure 3.4 also shows a schematic plot of vapour mass fraction versus depth from which Zone 1 can be identified as the rectangular region a-b-g-h. The front position can be located at any depth within Zone 1 but must be below the position b-g or below the depth hg. As the magnitude of the front position increases from zero towards the position b-g and as the flowing liquid vaporizes, the magnitude of vapour mass fraction also increases. When the vapour mass fraction,  $\phi$ , (see Figure 3.4) becomes equal to unity, a second single-phase vapour zone will be generated immediately after the vapour mass fraction has reached unity. We refer to this single-phase vapour zone as “Zone 2”. Zone 2 is identified as the rectangular region b-c-f-g of Figure 3.4. Zone 2 also has an extent which corresponds to the lines b-c and g-f or depth gf. The extent of Zone 2, upon its formation, grows with time with its leading edge (line c-f) increasing. Please note that at the onset of Zone 2 formation, a new front hereafter referred to as the “interface” is generated. The “interface” can be differentiated from the “front” by simply establishing the magnitude of vapour mass fraction. The “interface” exists only when the vapour mass fraction is unity. The position of the “front” or “interface” inside the porous layer establishes the maximum penetration depth reached by the flowing liquid and so, the words “front” and “interface” will be used quite often in this thesis to describe the position of the leading edge of the flowing liquid. A third zone, called Zone 3, is assumed to be mainly air-saturated and we further assume that this zone is confined within the rectangular region c-d-e-f of Figure 3.4. In this work, we are viewing Zone 3 as a passive bystander. Some of the graphs presented in this work will follow the shape of Figure 3.4.

In Table 3.1, we list some values of parameters used in all simulations in this chapter. In respect of porous media properties, Table 3.2 lists the properties of the soil, sandstone and glass wool substrates used in simulating the sample problems presented in this chapter. The vapour film property values used for the calculation of film boiling heat transfer coefficients is listed in Table 3.3. Before

**Table 3.1** Values of parameters used for liquid flow simulations.

| <i>Parameters used for simulation work</i>                        | <i>Value</i>          |
|-------------------------------------------------------------------|-----------------------|
| Range of Initial Temperature of Porous Layer (K)                  | 115 - 293             |
| Liquid Boiling Temperature (K)                                    | 112                   |
| Gravitational Acceleration ( $\text{m/s}^2$ )                     | 9.81                  |
| Flow Velocity (m/s)                                               | 0.15                  |
| Liquid Width (m)                                                  | 0.02                  |
| Depth of Substrate (m)                                            | 5                     |
| Kozeny Constant (...)                                             | 0.17                  |
| Film Boiling Heat Transfer Coefficient ( $\text{W/m}^2\text{K}$ ) | 155                   |
| Porosity (...)                                                    | 0.80                  |
| Tortuosity (...)                                                  | 2                     |
| Permeability ( $\text{m}^2$ )                                     | $1.4 \times 10^{-11}$ |
| Thermal Conductivity ( $\text{W/m/K}$ )                           | 0.193                 |
| Liquid Density ( $\text{kg/m}^3$ )                                | 422                   |
| Viscosity (Pas)                                                   | $113 \times 10^{-6}$  |
| Latent Heat of Vaporisation (KJ/kg)                               | 512                   |
| Specific Heat Capacity (J)                                        | 3350                  |
| Surface Tension (N/m)                                             | 0.014                 |

**Table 3.2** Thermal properties of Soil, Glass Wool, and Sandstone used in the sample problems presented in this chapter (Shaw and Briscoe, 1978, NIST WEBBOOK).

| <i>Substrate Property</i>                              | <i>DRY SOIL</i> | <i>GLASS WOOL</i> | <i>SANDSTONE</i> |
|--------------------------------------------------------|-----------------|-------------------|------------------|
| Thermal Conductivity ( $\text{W/m/K}$ )                | 5.18            | 0.038             | 1.20             |
| Density of Substrate ( $\text{kg/m}^3$ )               | 1600            | 24                | 2540             |
| Thermal Diffusivity ( $10^{-6} \text{ m}^2/\text{s}$ ) | 3.88            | 2.26              | 0.525            |
| Specific Heat Capacity ( $\text{J/kg/K}$ )             | 835             | 700               | 900              |

**Table 3.3** Properties of the LNG-vapour film used for simulations (NIST WEBBOOK, 2011).

| <i>Vapour Property</i>                  | <i>LNG</i> |
|-----------------------------------------|------------|
| Thermal Conductivity ( $\text{W/m/K}$ ) | 0.021      |
| Density ( $\text{kg/m}^3$ )             | 1.02       |
| Specific Heat Capacity (J)              | 2100       |
| Critical Temperature (KJ/kg)            | 191        |

proceeding with the analysis of the results, we would like to make it clear once again that our primary interest in this chapter is to understand the general evolution of a vaporizing liquid flowing through porous or penetrable media. Particularly, we are interested in: (1) understanding the behaviour and configuration of Zone 1 and Zone 2 with their associated extents, interfaces and fronts. This is to give good insights on liquid penetration rates; and (2)

conducting parametric analyses in order to understand how some parameters influence the evolution of the cryogen as it flows through the porous layer.

Let us first examine the vapour mass fraction profiles to see how the liquid front moves with increasing vapour mass fraction. Figure 3.5 presents the graph of vapour mass fraction, ( $\varphi$ ), versus depth,  $z$  at three different times of 2, 4 and 6 seconds, respectively. It is seen that at 2 seconds, about 24 percent of the liquid has vaporized and the front can be located at a depth of 230 mm. At 4 seconds,

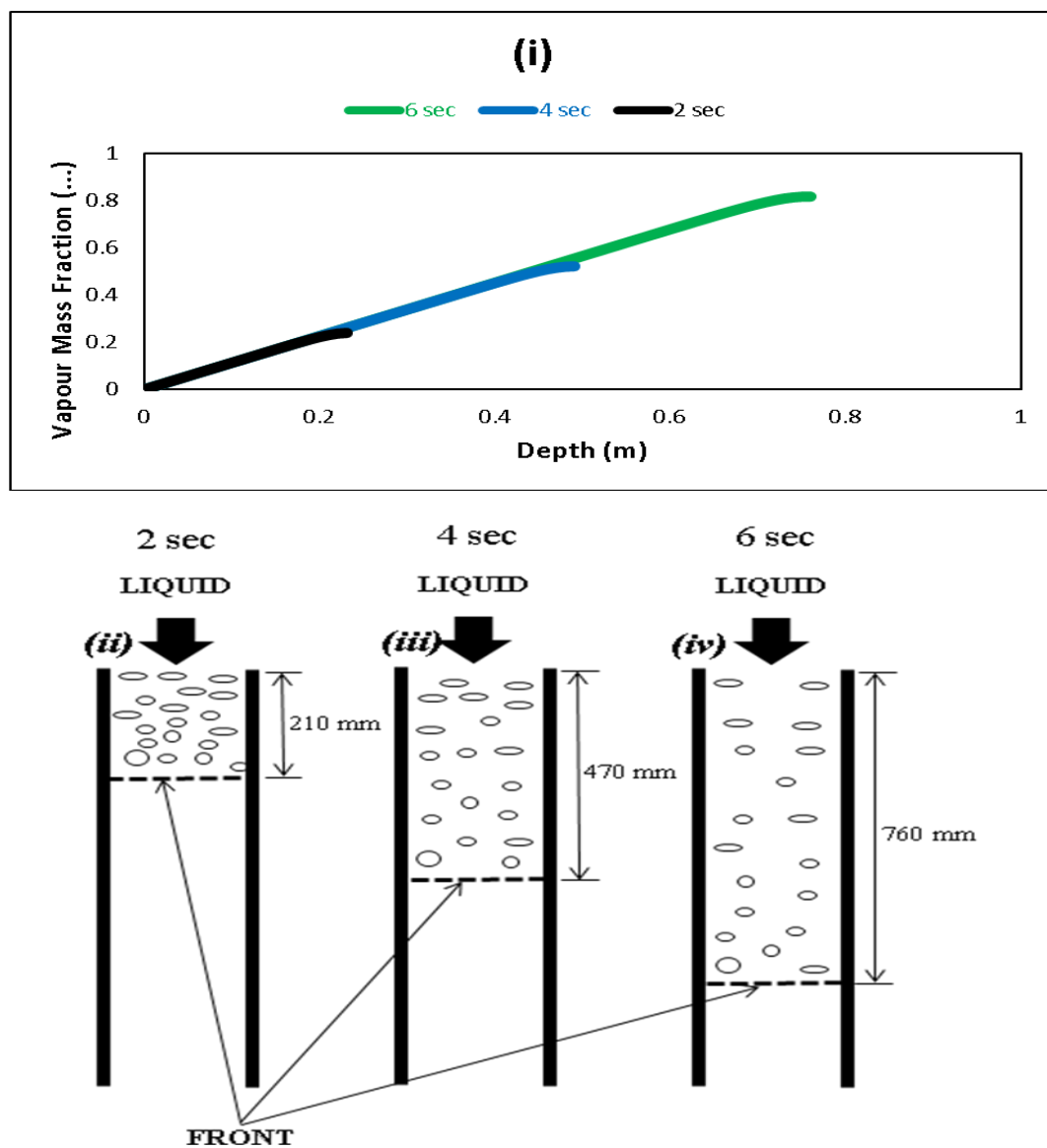


**Figure 3.5:** Graph of vapour mass fraction versus depth at 2, 4 and 6 seconds and an illustration of liquid front penetration downwards through the porous layer with steady-state heat transfer from the porous matrix.

52.3 percent of the liquid has vaporised and the front has moved deeper into the porous system to a depth of 490 mm. At 6 seconds, the percentage of the liquid that has vaporized has risen to 82 percent and the front position at this time is at approximately 760 mm. Figures 3.5 (ii)-(iv) provide a better visualisation of the liquid penetration behaviour. It can be seen, by merely looking at Figures. 3.5 (ii)-(iv) that the extent of Zone 1 grows with time and that the magnitude of the front position (i.e. depth of penetration) also increases with time. This is expected since the constantly injected liquid will cause Zone 1 to flow continuously down the porous media while simultaneously vaporising. Again, the knowledge of the front position provides us with information on how deep the LNG has penetrated the porous layer. Since the heat flux supplied to Zone 1 is at a constant rate, vaporization rate will remain constant and the time that will elapse before the interface is generated (or before the vapour mass fraction becomes equal to one) will be very short. When the vapour mass fraction has become equal to one (i.e. interface generation), and at later times after interface generation, Figure 3.5 (ii)-(iv) will now have the new features shown in Figures 3.6 (ii) to (iv). The interfaces are located at the same depth at the later times of 8 and 10 seconds,

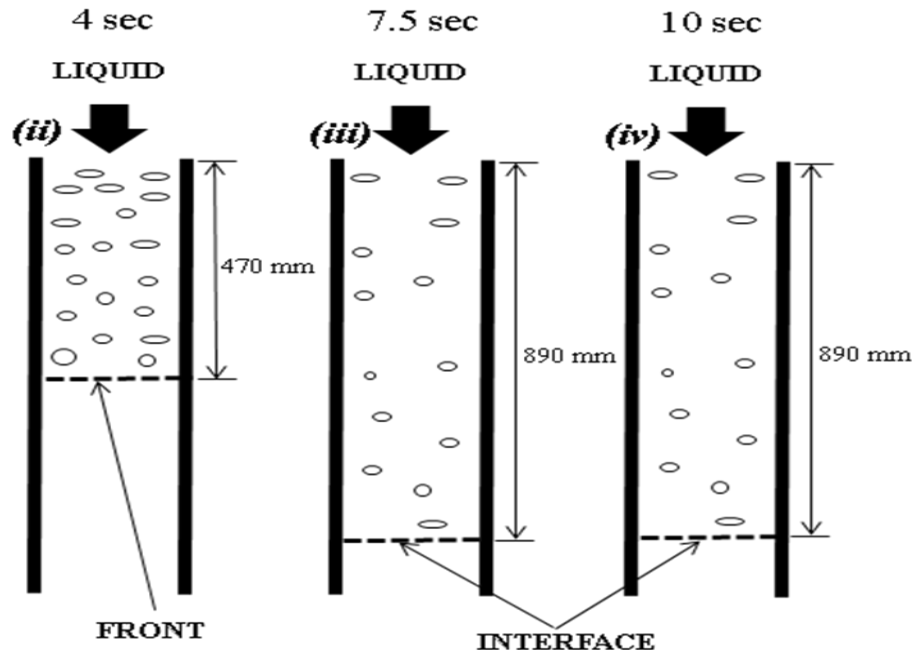


**Figure 3.6:** An illustration of liquid front penetration downwards through the porous layer at 4, 7.5 and 10 seconds, respectively, with steady-state heat transfer from the porous matrix.

respectively. The position of the interface at the later times indicate that the interface, once generated, does not move. This also implies that the liquid has stopped penetrating the porous layer, a behaviour attributable to the fact that the rate of inflow of the liquid at this location is now equal to the rate of outflow of vapour (if steady state solution).

### Front and Interface Movement

It should be intuitively evident that, under current modelling assumptions with respect to heat transfer, the front and interface would behave differently and it is a key interest in the present work to understand their evolution and behaviour. Figure 3.7 shows the position and/or movement of the liquid front and interface

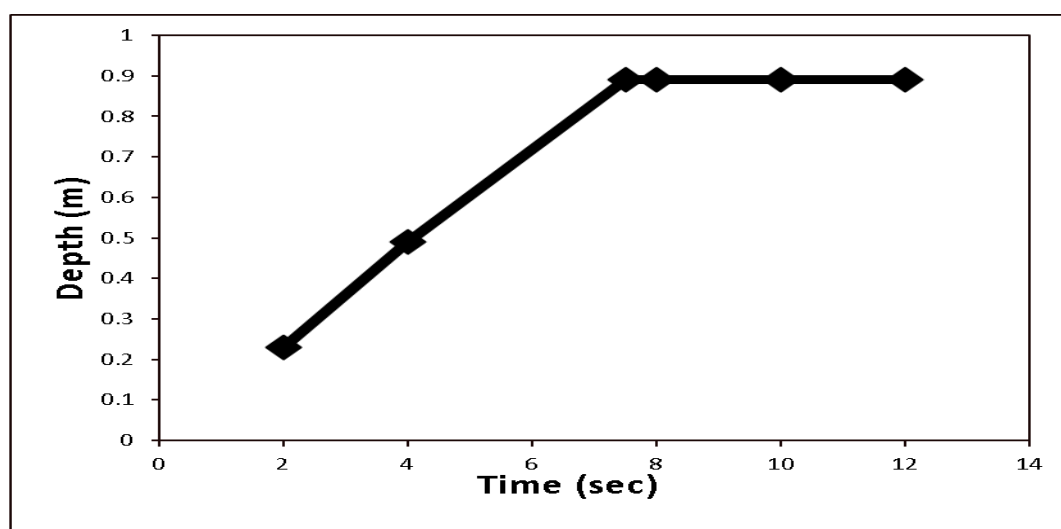


**Figure 3.7:** Position of the liquid front as a function of time showing front movement and stationary interface.

as a function of time for the steady-state case considered in this chapter. It is observed that there is an initial and steady increase in the depth penetrated by the front prior to interface generation (i.e.  $< 7.5$  seconds). At later times following interface generation (i.e.  $> 7.5$  seconds), liquid penetration stops. We have earlier explained that this steady-state behaviour of the interface is a reflection of the equilibrium between the liquid supply rate and the vapour release rate at the considered location. Thus, it can be deduced from Figure 3.7 that the maximum penetration depth of a liquid flowing through porous media under constant heat flux assumptions can be determined from the knowledge of the interface position.

## 3.12 Sensitivity Study

We expect that the variation of the values of the characteristic parameters, such as initial temperature of porous media, porosity, tortuosity, substrate type, and flow velocity can affect the evolution of the zones (Zone 1 and Zone 2) including front and interface positions. We have investigated some of these important effects in this section and it should be mentioned here that the analyses were conducted by varying only one parameter at a time.

### 3.12.1 Initial Temperature of Porous Matrix ( $T_{s0}$ )

Here, the sensitivity to the initial temperature of the porous medium is investigated. The results for two different initial temperatures are shown in Figure 3.8. We note from this figure that, at a relatively higher initial porous material temperature of 293 K, liquid penetration depth is relatively short when compared to the higher depth penetrated when the initial temperature of the porous media is specified as 115 K. A temperature of 115 K is clearly below the leidenfrost point, but we have chosen this arbitrary value to demonstrate how the magnitude of the superheat of the porous matrix can influence liquid vaporization and penetration rates. At this relatively lower initial temperature, the liquid penetrated to a depth of 890 mm, whereas the penetration of the liquid at the 293 K initial temperature is just at a depth of 20 mm. The higher the temperature difference between the flowing liquid and the porous matrix, the higher the amount of heat supplied to the cryogen and the shorter the depth of liquid penetration. This shorter liquid penetration depth can equally be attributed to the relatively higher vaporization effects that tend to reduce liquid penetration rates. On the other hand, a relatively smaller temperature difference between the porous matrix and the cryogen tends, as already indicated, to increase the maximum penetration depth of the liquid.

In examining the behaviour of Zone 2 at the two temperatures of 115 K and 293 K, the extent of Zone 2 is quickly noticed to have a value of 200 mm when the initial temperature is 115 K. When the initial porous material temperature was increased to a value of 293 K, the extent of Zone 2 is shown to have increased to a depth of 1170 mm. The reason for such a marked difference can be attributed,

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again, to the higher superheat of the substrate. A higher superheat implies a higher vapour production rate. The implications of Zone 1 and interface behaviour to the physical system under study is that a lower liquid penetration rate caused by a higher superheat of the substrate has the potential to drastically reduce the maximum depth of penetration of the liquid, hence preventing the liquid from reaching any characteristic threshold depth above which either environmental risks or safety risks may increase. For the infinite heat source assumption discussed in this chapter, the magnitude of the maximum liquid penetration depth is proportional to the quotient of flow (Darcy) velocity and the source factor, as discussed in Section 3.3.

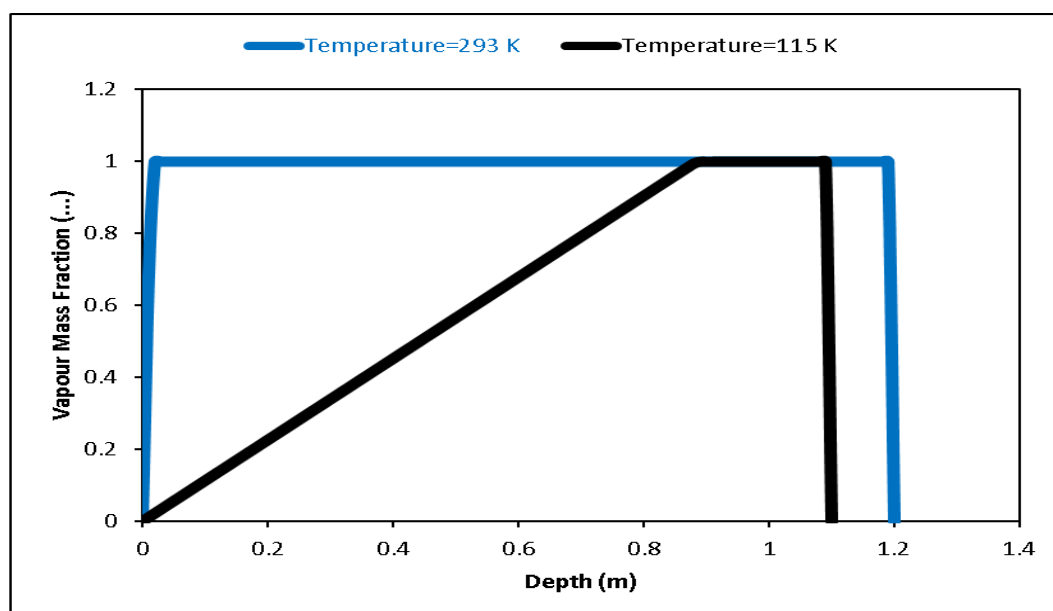


**Figure 3.8:** Effect of initial porous material temperature on vaporizing flow behaviour. Liquid penetration is shorter for relatively higher superheat of the porous media whereas the penetration of the liquid is deeper for relatively lower superheat because the vaporization effects retarding fluid flow is reduced.

### 3.12.2 Porosity ( $\phi$ )

Porosity is a parameter contained in the Kozeny-Carman equation (Bear, 1988) which was used to estimate the specific surface area used in this work. Increasing the porosity decreases the penetration depth just because the interstitial or frontal velocity of the fluid is reduced. This is just because at higher porosity there is more volume for the fluid to occupy. Table 3.4 shows the

model's sensitivity to increase in porosity. The maximum liquid penetration depths (i.e. interface position) decreases as the porosity is increased from 0.80 to 0.93. For a porosity of 0.80, the liquid penetrated to a depth of 890 mm. The porosity effect caused a decrease of liquid penetration depth to 780 mm when a relatively higher porosity value of 0.87 was specified (see Table 3.4). Table 3.4 also indicates that a further increase in porosity to a value of 0.93 results to a further decrease of the liquid penetration depth - to a value of 710 mm. The chosen porosity values, though relatively high, are below the upper limit to the porosity of a fibrous material which is approximately 0.98 (Iberall, 1950). The corresponding lower limit of fibrous material porosities is 0.50 (Iberall, 1950). What is also obvious from the results is the fact that porosity would be of great utility in the design of real systems, especially, for those systems using fibrous materials as they can be easily packed to a desired density and porosity (Iberall, 1950).

**Table 3.4** Values of parameters used for parametric analysis investigating the effects of porosity on liquid penetration and vaporization rates.

| <i>Porosity Value (<math>\phi</math>) [...]</i> | <i>Kozeny Constant (<math>c_0</math>) [...]</i> | <i>Maximum Depth (<math>z_{max}</math>) [m]</i> |
|-------------------------------------------------|-------------------------------------------------|-------------------------------------------------|
| 0.80                                            | 0.17                                            | 0.89                                            |
| 0.87                                            | 0.17                                            | 0.78                                            |
| 0.93                                            | 0.17                                            | 0.71                                            |

### 3.12.3 Substrate Type

Figure 3.9 shows the influence of substrate type on liquid flow behaviour. We plotted results of vapour mass fraction versus depth using three different types of porous materials: glass wool, soil and sandstone. We quickly noticed that the vapour mass fraction when the liquid wets a glass wool porous material is relatively very small relative to the observed profiles for soil and sandstone porous materials, respectively. Liquid flow through the soil reached a maximum depth of 890 mm while the penetration depth of the liquid through sandstone can be located at the shallower depth of 410 mm. We should also notice that, for glass wool, the front has penetrated to a depth of 1030 mm with just 8 percent of the liquid vaporizing at this position. The deeper penetration of the glass wool

substrate when compared to the other two cases can be attributed to the low thermal conductivity of glass wool and the low heat exchange rate at the interface between the flowing liquid and the solid matrix. The relatively low heat transfer rate is reflected in the relatively low vaporization rate experienced by the flowing liquid. The overall effect of substituting the glass wool substrate with the other two substrates (i.e. soil or sandstone) is a shallower penetration of the liquid into the porous medium. Thus, liquid vaporization and penetration rates are strongly influenced by the type of the porous material through which the liquid propagates.

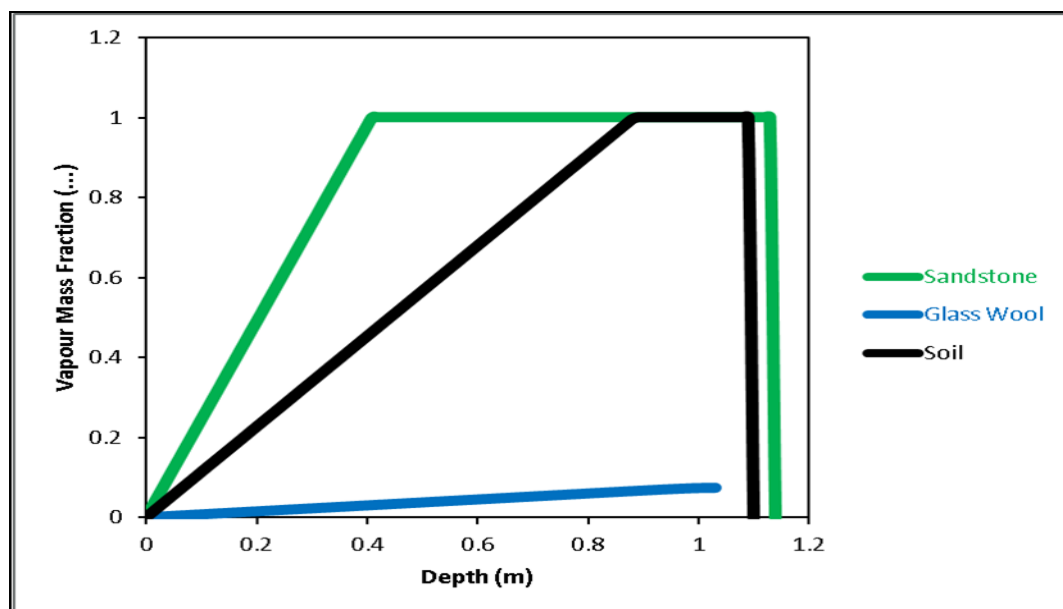


**Figure 3.9:** Effect of substrate type on vaporizing flow behaviour. The liquid penetrated deepest into the glass wool porous material and shallowest into the sandstone porous material. This vaporization and penetration behaviour is essentially influenced by the different thermal properties of the porous materials.

### 3.12.4 Flow Velocity ( $u$ )

Figure 3.10 shows the results of our model's sensitivity to changes in flow velocity, from which the strong impact of decreasing or increasing the injection velocity can be revealed. We observe that the interface position or liquid penetration depth decreased from 890 mm to about 500 mm when the flow velocity was decreased from 0.15 m/s to 0.0825 m/s. This indicates that for a relatively slow flow event, liquid front penetration rate is more affected by the so-called vaporization effect. The reverse is the case for a relatively fast flow event where the liquid front penetration rate is relatively higher at each given time.

Figure 3.10 also shows that decreasing the flow velocity increases the slope of the vapour mass fraction profile allowing the liquid front to reach shallower depth within the porous medium. The extent of Zone 2 at the relatively lower velocity 0.0825 m/s is 70 mm while its value at the relatively higher velocity of 0.15 m/s is 200 mm. It is thus clear from these results that flow/injection velocity is a very important parameter that influences liquid vaporization and penetration rates.

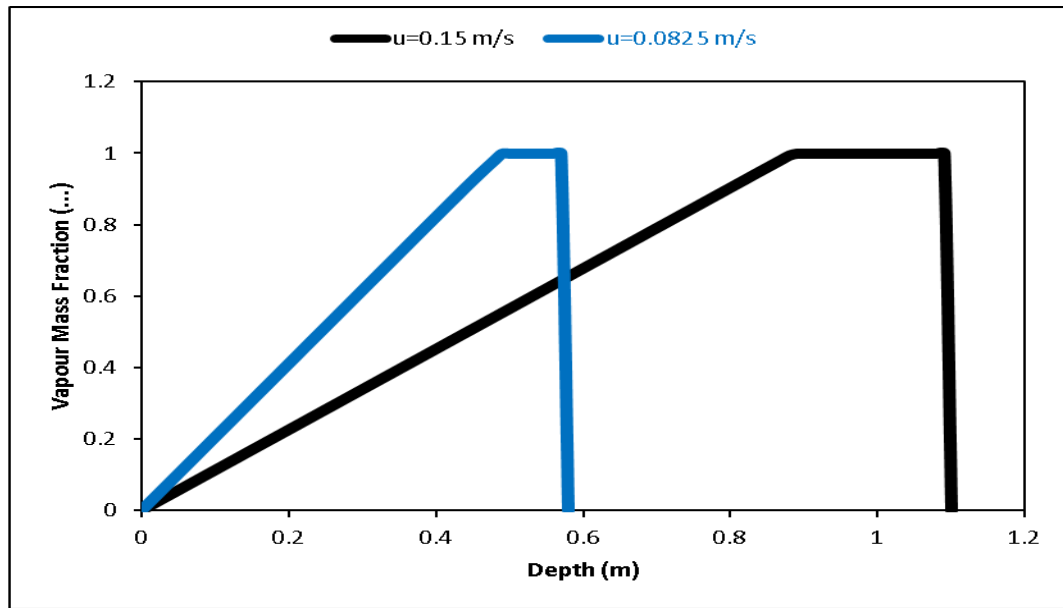


**Figure 3.10:** Effect of flow velocity on vaporizing liquid flow behaviour indicating that increasing the injection velocity of the liquid will increase the maximum penetration depth reached by the flowing cryogen

### 3.13 Discussion and Conclusions

In this chapter, a formulation of a model describing the vaporizing flow of fluids through porous or penetrable media has been presented. For simplicity, an infinite heat source assumption has been imposed. One reason for analysing such an idealised case is to provide useful information and a firm basis for the modelling of a more complicated process. The vaporizing flow model has been used to investigate the importance of different parameters that affect the vaporizing flow phenomena. Such a parametric study helped to gain useful insights that can aid further model development. The parameters studied include initial temperature of porous matrix, porosity, substrate type, and flow (injection) velocity. The model was solved numerically using a finite difference method. The

consistency, stability and convergence of the numerical method was successfully tested and an optimum grid size was selected and used for the simulation of the sample problems presented in this chapter. The model was further verified by comparing its results with that obtained from an analytical solution, with excellent agreement.

Furthermore, it is now understood that when a pressure-driven flow of a liquid flows downwards through a hot porous or penetrable media, a descending two-phase zone containing both liquid and vapour phases is immediately generated. We have identified this two-phase layer as “Zone 1”(see Figure 3.4). We argued (using Figure 3.4) that the extent (vertical depth) of Zone 1 will grow with time and that the leading edge of the growing Zone 1 is called the “front”. As the front and the depth of Zone 1 grows, heat continues to flow into Zone 1 from the porous matrix and new LNG vapour is continuously produced. The increase in the amount of vapour within Zone 1 increases the vapour mass fraction until a time when the vapour mass fraction becomes equal to one. Thereafter, liquid front movement stops and a non-moving liquid front called an “interface” is generated. After interface generation, a new single-phase vapour layer ahead of the interface will start developing. We have referred to this new vapour layer as “Zone 2”. The presence of an interface, implies that steady-state is established and that liquid penetration has stopped. The boundary conditions, geometry, including front and interface positions can influence both penetration and vaporisation rates of the liquid. Describing these in physical terms, the liquid injected from the top of the porous layer produces vapour at an evolving two-phase zone (Zone 1). The evolution of this two-phase layer ceases when a characteristic maximum penetration depth (the interface generation) has been reached. Thereafter, the steady-state and high conductive heat flow into the flowing liquid prevent further penetration of the liquid through the porous medium, leading to a situation in which the maximum penetration depth of the liquid front at a given depth and time may be underpredicted since we have assumed, at this point, that the heat source from the solid matrix is infinite. Further conclusions can be drawn from this study among which are as follows:

- The vaporizing liquid flow phenomena is strongly affected by initial porous material temperature. From a hypothetical analysis conducted using this parameter, it is clear that increasing the initial porous material temperature from 115 K to 293 K decreases the maximum liquid penetration depth (interface) by 98 percent.
  - The maximum penetration depth of the liquid decreases with increasing porosity.
  - The vaporizing liquid flow is strongly affected by the type of porous material. The maximum liquid penetration depth decreased by about 54 percent when soil was substituted with sandstone as the porous material. Decreased vapour production rate and increased liquid penetration rate was observed when glass wool was used as the porous material.
  - The maximum depth of penetration of the liquid is reduced by about 45 percent when the flow (injection) velocity was decreased from a value of 0.15 m/s to a value of 0.0825 m/s. The reasons for this is obvious as a lower flow velocity implies that a slower penetration of the liquid and higher liquid vaporisation rate will ensue. This is because more time is allowed for heat exchange at lower velocities resulting in a more rapid interface generation (or increased vapour production rate) and a shorter interface position (or decreased liquid penetration rate).
  - Finally, we would like to stress at this point again that the models presented in this chapter are highly idealised in the sense that they use assumptions that does not reflect what might be expected in real situations. Specifically, the infinite heat source assumption that invoked a constant temperature boundary condition implies that both models overestimate the vaporisation rate and underestimate the maximum penetration depth. This should not be the case because it is intuitively clear that the liquid upon contact with the solid matrix at
-

a considered location will cool the surface until a time when the temperature of the surface equals that of the liquid. It is also evident that such a time-dependent boundary condition can influence the overall evolution of the liquid inside the porous layer, including its penetration and vaporisation rates. We also expect that the thickness of the vapour film separating the liquid and the solid matrix will decrease with time as the surface temperature of the solid matrix decreases. The disappearance of the vapour film will lead to a change in the boiling regime that can also influence vaporizing liquid flow behaviour. We believe all these issues to be important and they will be addressed in Chapter 4.

# 4

## ***Liquid Flow with One-Dimensional (1-D) Unsteady-State Heat Conduction***

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In Chapter 3, we presented a model that describes the vaporizing flow of liquids through porous or penetrable media. In order to show that the model is fully operational, we first analysed results of a steady-state case where a constant temperature boundary condition is invoked at the interface between the flowing liquid and the solid matrix. This boundary condition ensured that the heat released by the porous matrix for the vaporization of the liquid is constant. We henceforth call this steady-state vaporizing flow model as BoilFlowModel I. In this chapter, we extend the BoilFlowModel I by: (a) incorporating the effects of one-dimensional unsteady-state heat conduction at the interface between the flowing liquid and the solid matrix. This time-dependent boundary condition enables the model to account for the actual time-dependent cooling of the solid matrix and (b) incorporating the effects of boiling regime (to account for the effects of full-boiling heat transfer coefficients) on vaporization and penetration behaviour. Although in the first instance, we assumed in Chapter 3, and for simplicity, that all liquids will boil in the film-boiling regime, this would not always be the case as the type of regime under which a flowing liquid would boil depends on the thermophysical properties of both the liquid and substrate.

This chapter is organized into six sections. Section 4.0 presents the solution of the heat conduction problem including an analysis of the two limiting vaporizing flow cases considered in this chapter. Section 4.1 presents the numerical procedure employed for the solution of these vaporizing flow models. Section 4.2 presents a model verification study in which the results from the most realistic

vaporizing flow model (the so-called BoilFlowModel III) are compared with those obtained from the use of a commercial software (TOUGH2) (Pruess et al., 1999). Section 4.3 presents results from sample simulation problems. In Section 4.4, some parametric trends are presented examining the sensitivity of the vaporizing flow phenomena to changes in initial substrate temperature, flow (injection) velocity, substrate type, thermal conductivity, thermal diffusivity and liquid type. Section 4.5 investigates the effects of boiling regime on the vaporizing flow behaviour. The sixth and final section (Section 4.6) presents the discussion and conclusions from this chapter.

## **4.0 Solution of the Heat Conduction Problem**

The evolution of liquids injected into a porous or penetrable medium will be influenced by the type of heat transfer occurring at the interface between the flowing liquid and the solid matrix. A time-varying temperature boundary condition imposed at the heat exchange surfaces inside the porous medium can provide a good approximation of the amount of liquid vaporized as a function of time and depth. In order to derive simple but realistic estimates of the conductive heat transfer occurring at the heat exchange surfaces of the porous matrix, two limiting vaporizing flow cases are examined in this section. Sections 4.0.1 and 4.0.2 describe these cases and present the heat conduction equations that have been applied and used to solve for the wetted surface temperature of the solid matrix, heat flux, liquid vaporization rate and liquid front penetration rate.

### **4.0.1 Limiting Case #1: Total Wetting of Substrate**

This first limiting case assumes that the vaporizing flow through the porous media is so fast that total available surface areas are instantaneously wetted by the flowing liquid. Since the liquid is assumed to be at its boiling temperature at the point of injection, any heat released by the cooled substrate will be used to vaporize the liquid. Neglecting all other sources of heat, the vaporisation rate can be predicted by solely considering the rate of heat transfer from the substrate into the liquid. We have, for simplicity, also assumed that heat transfer rates are limited by the presence of a stable vapour film. Hence, assuming a dry and homogenous porous media with constant thermal properties, the temperature distribution and heat flux at any vertical depth,  $z$ , within the porous media can

be predicted by solving the following one-dimensional unsteady-state heat conduction equation (Equation 4.1(a)) written in both its partial differential equation (PDE) and finite difference equation (FDE) forms (see Equations 4.1(a) and 4.1 (b)),

$$\frac{\partial T}{\partial t} = \alpha_s \left( \frac{\partial^2 T}{\partial x^2} \right) \quad (4.1a)$$

$$\frac{T_n^{n0+1} - T_n^{n0}}{\Delta t} = \alpha_s \left( \frac{T_{n-1}^{n0} - 2T_n^{n0} + T_{n+1}^{n0}}{(\Delta x)^2} \right) \quad (4.1b)$$

The finite difference approximation of Equation 4.1(a) is based on the nodal points distribution shown in Figure 4.1(a) below. Re-arranging Equation 4.1(b) and making  $T_n^{n0+1}$  subject, gives Equation 4.1(c),

$$T_n^{n0+1} = T_n^{n0} + d_x (T_{n-1}^{n0} - 2T_n^{n0} + T_{n+1}^{n0}) \quad (4.1c)$$

where  $T$  is the PDE version of the temperature distribution in the solid matrix at a lateral depth,  $x$ , and time,  $t$ ,  $T_n^{n0+1}$  is the finite difference form of  $T$ ,  $n$  is the space counter in the horizontal  $x$ -direction, and  $n0$  is the initial time step counter. The diffusion number,  $d_x$ , and thermal diffusivity of the substrate,  $\alpha_s$ , are given by Equations 4.2(a) and 4.2(b), respectively,

$$d_x = \frac{\alpha \Delta t}{(\Delta x)^2} \quad (4.2a)$$

$$\alpha_s = \frac{k_s}{\rho_s C_{ps}} \quad (4.2b)$$

where  $k_s$  is the thermal conductivity of the substrate (W/m/K),  $\rho_s$  is the substrate density (kg/m<sup>3</sup>) and  $C_{ps}$  is the substrate specific heat capacity (J/kg/K).

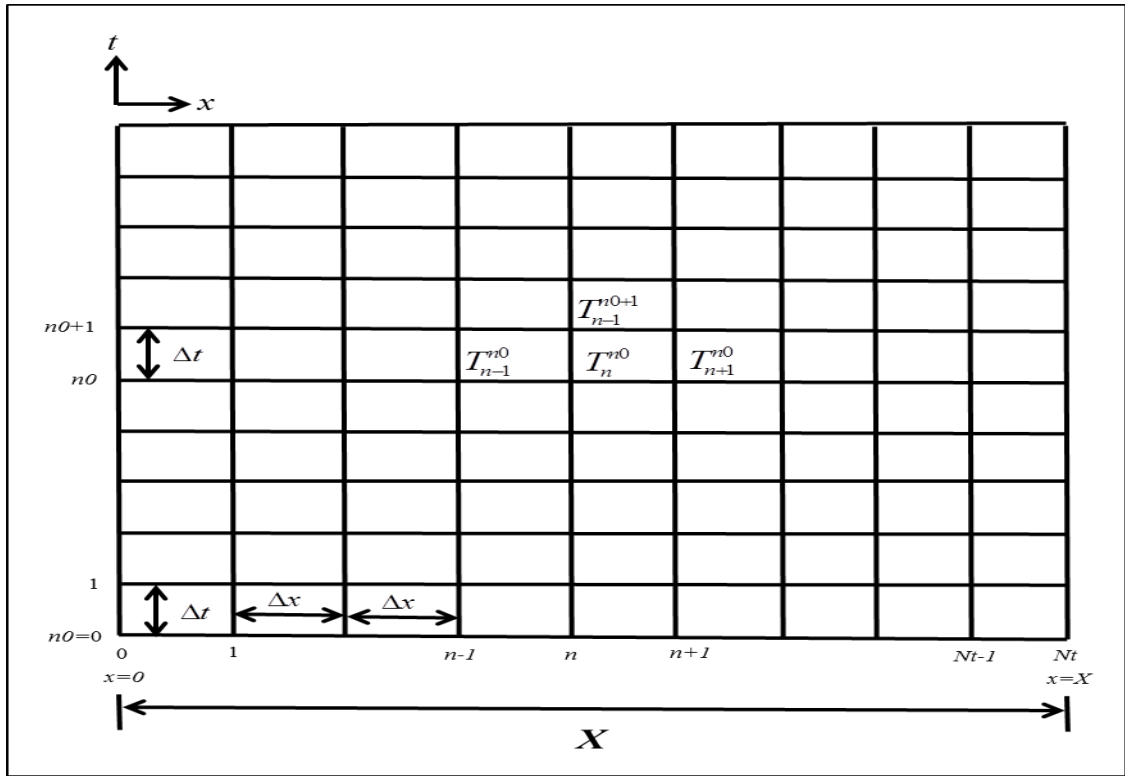


**Figure 4.1(a):** Nodal points distributions showing the discretization of the  $xt$  domain for finite difference approximation of the unsteady-state 1-D heat conduction equation.

The solution of the relevant finite difference approximations of Equations (4.1) and (4.2) is subject to the initial condition given by Equation (4.3),

$$T = T_{s0} \quad (4.3)$$

for all values of  $x$ , where  $T_{s0}$  is the initial uniform temperature of the substrate. Since the heat transfer rate is limited by a stable vapour film, a first boundary condition reflecting this situation is imposed at the heat exchange surface (i.e. at  $x=0$ ) as follows,

$$-k \frac{\partial T}{\partial x} = h(T - T_b) \quad (4.4a)$$

where  $T_b$  is the liquid boiling temperature (K) and  $h$  is the film boiling heat transfer coefficient. In order to calculate the temperatures at the boundary nodes, Equation 4.4a was discretized by introducing the so-called fictitious node

concept (see Ozisik, 1993) and using a central difference formula in approximating the derivative term in Equation 4.4a. In this approach, we assumed that a fictitious node (i.e.  $n=-1$ ) with a fictitious temperature,  $T_{-1}$ , is assigned at a horizontal distance,  $\Delta x$ , from the heat exchange boundary (i.e.  $n=0$ ).  $T_{-1}$  was thereafter eliminated following the evaluation and rearrangement of Equation 4.1c for  $n=0$ , ultimately yielding Equation 4.4b which is the finite difference equation for the boundary nodes at  $x=0$ ,

$$T_0^{n0+1} = (2d_x)T_1^{n0} + \left(1 - \frac{2d_x h \Delta x}{k_s} - 2d_x\right)T_0^{n0} + \left(\frac{2d_x h \Delta x}{k_s}\right)T_b \quad (4.4b)$$

Finite difference equations for a second boundary condition at  $x=X$  are not needed because the temperature,  $T_{s0}$ , is prescribed at the nodes in this boundary,

$$T_{Nt}^{n0+1} = T_{s0} \quad (4.5)$$

This second boundary condition can also be modified by keeping the boundary at  $x=X$  insulated (i.e.  $\partial T / \partial x|_{x=X} = 0$ ), as an option, meaning that at a particular depth no heat flow occurs. Hence, by explicitly solving the finite difference approximation of Equations 4.1a (i.e. Equation 4.1b) subject to the invoked boundary conditions and their relevant finite difference approximations (i.e. Equations 4.3, 4.4 and 4.5), the 1-D temperature distribution in the solid matrix can be predicted. In particular, the wetted surface temperature of the solid matrix at  $x=0$  can be calculated. An analytical solution for the wetted surface temperature has also been provided (Carslaw and Jaeger, 1959a; Vesovic, 2007). In his most recent work on LNG vaporization following spills onto water surfaces, Vesovic (2007) reported the variation of the surface temperature as a function of time as follows,

$$T = T_b + (T_{s0} - T_b) \exp\left(\frac{h^2 \alpha_s t}{k_s^2}\right) \operatorname{erfc}\left(\frac{h^2 \alpha_s t}{k_s^2}\right)^{0.5} \quad (4.6)$$

where  $T$  is the variation of the surface temperature (K),  $k_s$  is the thermal conductivity of the substrate (W/m/K),  $h$  is the heat transfer coefficient (W/m<sup>2</sup>/K),  $t$  is time (s), and  $T_b$  is the liquid boiling temperature (K). A coupling of the solved heat conduction equation (Equation 4.1), which determined the wetted surface temperature of the solid matrix, and the finite difference form of the advection equation (Equations 3.6 or 3.7) of Chapter 3 subject to their relevant initial and boundary conditions yield the amount of liquid vaporized (vapour mass fraction) as a function of time,  $t$ , and depth,  $z$ . We have referred to the vaporizing flow model based on the present time-dependent heat conduction approach as the BoilFlowModel II.

In Figure 4.1, we have plotted the wetted surface temperature and vaporisation rate profiles for a porous medium of height 5 meters and of uniform initial temperature 293 K into which a liquid of boiling temperature 112 K is constantly injected. We assumed that the liquid was injected into the porous layer with an injection velocity of 0.015 m/s. We also assumed that the injected liquid boils in the film boiling regime with a heat transfer coefficient of 155 W/m<sup>2</sup>/K (Conrado and Vesovic, 2000). A consideration of the profiles of temperature and vaporization rate at a depth 0.19 meters down the porous layer shows that the wetted surface temperature and vaporization rate profiles decreases with time.

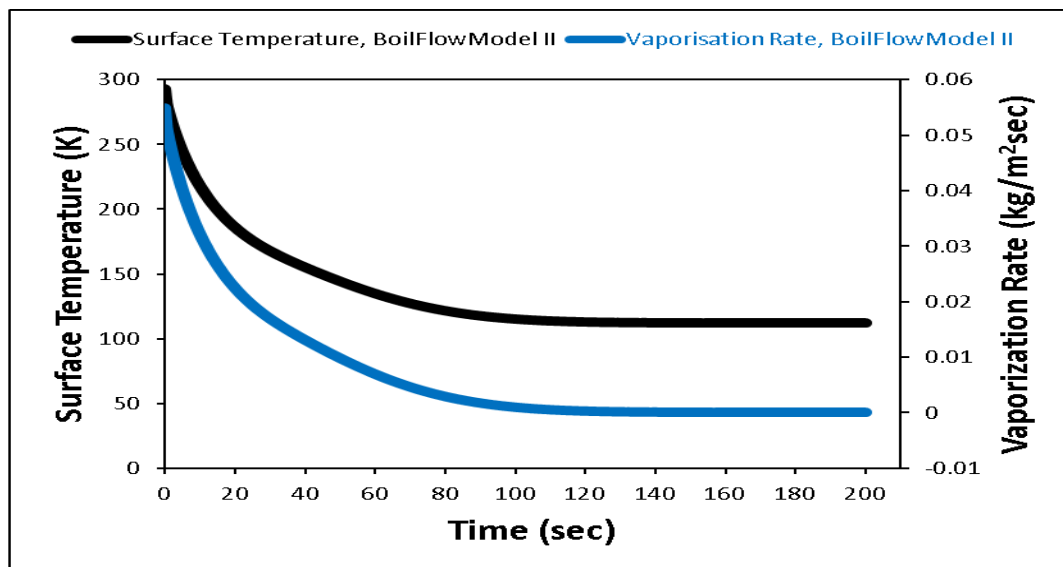


**Figure 4.1(b):** Profiles of wetted surface temperature and vaporization rate based on BoilFlowModel II simulations. The considered location is at  $z=0.19$  meter.

The wetted surface temperature is seen to have cooled towards the liquid temperature of 112 K within approximately 200 seconds of liquid-solid contact at this location. The corresponding decrease in vaporisation rate (blue curve of Figure 4.1) at this location indicates that approximately 185 seconds elapsed before the rate of vaporization could decrease from the initial value of 0.055 kg/m<sup>2</sup>/sec towards zero.

#### 4.0.2 Limiting Case #2: Partial Wetting of Substrate

This second limiting case assumes that the vaporizing flow is sufficiently slow that a partial wetting of the substrate occurs. By “partial wetting”, we imply that the cooling of the substrate is possible only when the flowing liquid has reached a heat exchange surface at a considered location. For this case, the Fourier’s law of heat conduction (Equation 4.1) and its corresponding finite difference equations are also applicable only that, this time, the first boundary condition (Equation 4.4) is replaced with the new forms,

$$-k \frac{\partial T}{\partial x} = h(T - T_b) \quad \text{for } z \leq z(t) \quad (4.7a)$$

$$T = T_{s0} \quad \text{for } z \geq z(t) \quad (4.7b)$$

where  $z(t)$  defines the time-dependent penetration depth of the liquid or rather interface and/or front position in the porous layer. Equations 4.7a and 4.7b ensure that only the portions of a substrate surface which have been wetted by the flowing liquid will undergo a temperature change. Therefore, if the liquid is yet to wet a substrate surface at a particular location,  $z$ , the wetted surface temperature of the substrate at this location(s) ( $z \geq z(t)$ ) remains at its initial value. But as soon as this location is wetted, the cooling of the substrate surface towards the boiling temperature of the liquid immediately ensues. Hence, the wetted surface temperature of the substrate at location  $z(t)$  can be predicted by solving Equation (4.1) subject to the initial and boundary conditions defined by Equations 4.3, 4.5 (or optionally the relaxed version involving the imposition of a constant temperature boundary condition at a finite distance into the matrix) and 4.7. It is important to stress here that the time,  $t$ , is now replaced by the

difference ( $t-t(z)$ ) where  $t(z)$  denotes the period from the time of initial injection across depth,  $z=0$  to the time the front and/or interface has reached a considered depth,  $z(t)$ . The stability criteria for the solution of the two limiting cases discussed above are that  $d_x$ , in Equation 4.2a, must not be used outside the limit:  $0 < d_x \leq 1/2$ . Figure 4.2 shows the profiles of wetted surface temperature and corresponding vaporization rate at a depth of 0.19 m. Initially, the substrate surface is unwetted until after approximately 14 seconds. The wetting of the substrate upon liquid-solid contact results in the immediate cooling of the substrate and it is clearly evident from the curve in Figure 4.2 that the cooling rate and corresponding vaporization rate decreases as a function of time and that the profiles follow the trends observed in Figure 4.1. As the solid matrix surface approaches the liquid boiling temperature, the vaporization rate at this location also approaches zero (since the temperature difference driving force approaches zero as well) signifying that the low vaporization effect at this particular location would tend to increase liquid penetration rates.

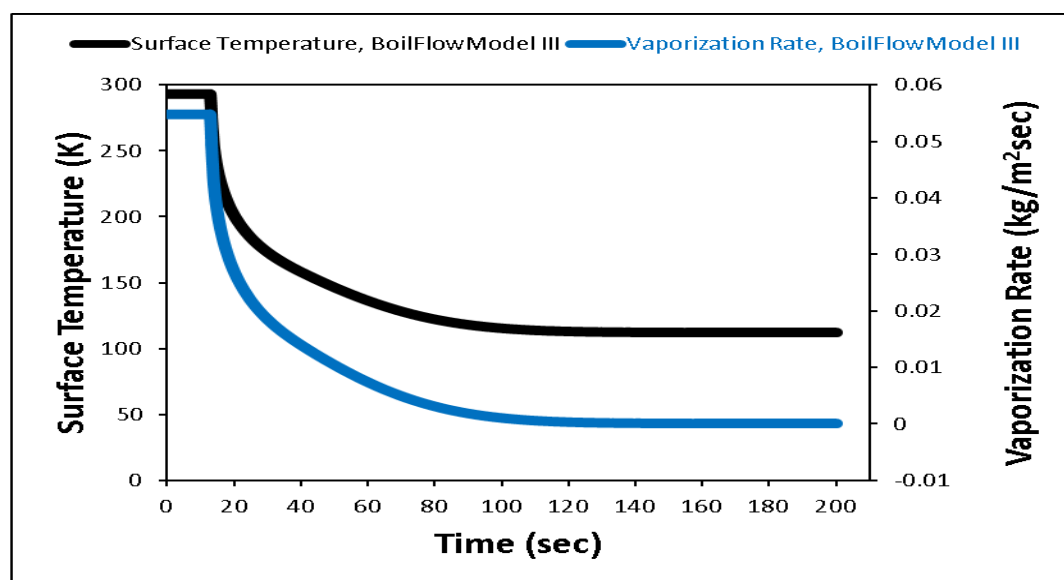


**Figure 4.2:** Profiles of wetted surface temperature and vaporization rate based on BoilFlowModel III simulations. The considered location is at  $z=0.19$  meter.

#### 4.0.3 Distinguishing the Three Vaporizing Flow Models

The BoilFlowModel II and BoilFlowModel III cases presented in Sections 4.0.1 and 4.0.2 and the BoilFlowModel I case discussed in Chapter 3 differ in a number of ways. In the following paragraphs, we distinguish these three vaporizing flow

models in terms of the trends in the profiles of their wetted surface temperature and vapour mass fraction.

#### 4.0.3.1 Wetted Surface Temperature Profiles

Considering the wetted surface temperature profiles at a depth of 0.19 meters, it is shown from Figure 4.3 that no cooling of the substrate occurred for the steady-state case (i.e. BoilFlowModel I). This is in line with the constant temperature boundary condition assumption. For BoilFlowModel II simulations, temperature profile trends indicate that the wetted substrate surface cools with time and the cooling rate is relatively rapid despite the presence of a low thermal conductivity vapour that is immediately generated upon liquid-solid contact. From the temperature profile obtained from BoilFlowModel III simulations, it is clear that a finite time must elapse before a considered location inside the porous medium can be reached and/or wetted by the flowing liquid. Thereafter, the profile of temperature would tend to follow the trend observed using BoilFlowModel II.

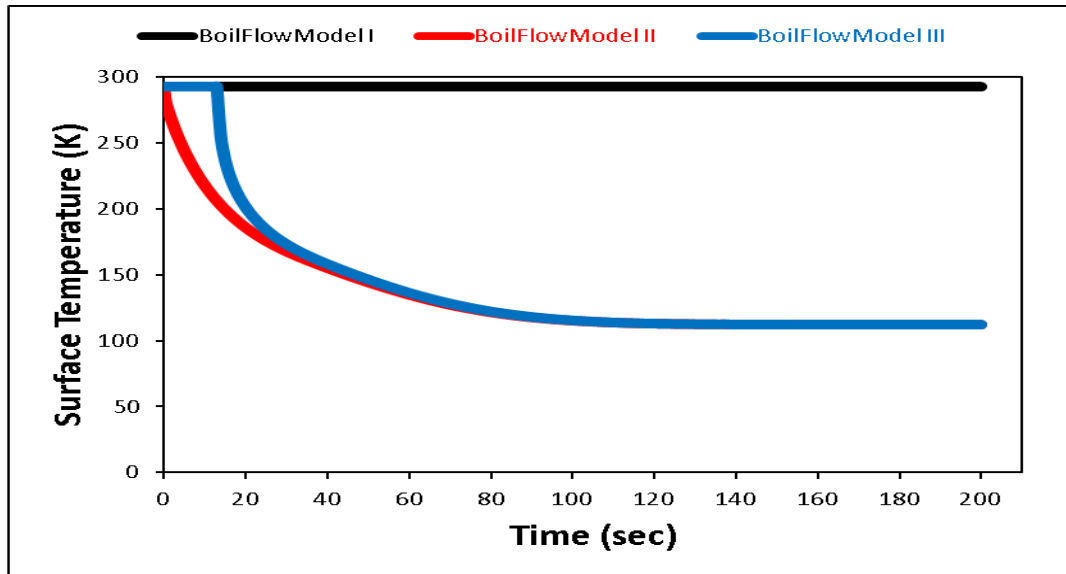


**Figure 4.3:** Comparison of surface temperature profiles of the solid matrix obtained from the BoilFlowModel I, BoilFlowModel II and BoilFlowModel III simulation cases. The considered location is at a depth,  $z=0.19$  meter.

#### Vapour Mass Fraction Profiles

As a vaporizing liquid flows through a porous or penetrable medium, the magnitude of the vapour mass fraction increases until an interface is generated. An interface is generated when the magnitude of vapour mass fraction equals 1.

Figure 4.4 shows that the position of the interface is shortest when simulations are performed using the BoilFlowModel I. For a liquid injected with, say, 0.15 m/s velocity, it took just 6 seconds for the first interface to be generated and this occurred at a depth of 0.67 m. The time for first interface generation when BoilFlowModel II is employed is approximately 12.5 seconds and the location of the interface is at 1.62 m which is almost twice the liquid penetration depth observed for the BoilFlowModel I simulation case. For BoilFlowModel III simulation case, the first interface can be located at a depth of 1.49 meters after about 11 sec. The different elapsed time before first interface generation for the difference models suggests that the evolution of a flowing liquid inside a porous medium would depend on the type of model employed for simulating the vaporizing flow phenomena. Since the position of the interface establishes the depth reached by the liquid within a given duration, Figure 4.4 reveals that vapour production rate is highest using BoilFlowModel I and lowest using BoilFlowModel II. In terms of liquid penetration rates, use of BoilFlowModel 1 shows the liquid becoming least penetrating while the BoilFlowModel II simulation results show the most liquid penetration trend. The BoilFlowModel III is shown to be intermediate between the other vaporizing flow cases, though closer to the trends observed for the BoilFlowModel II case. The reasons for these different trends are discussed in Section 4.0.4.

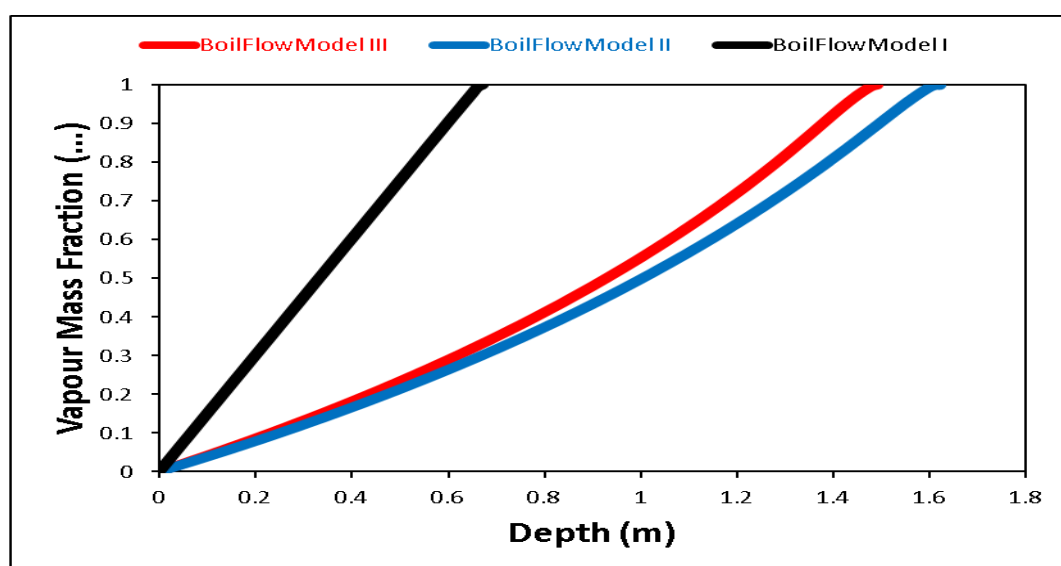


**Figure 4.4:** Comparison of BoilFlowModel I, BoilFlowModel II and BoilFlowModel III cases based on the trends in the profiles of their respective vapour mass fraction and the corresponding times for interface generation.

#### 4.0.4 Discussion

We have so far presented three vaporizing flow models distinguished by the type of heat transfer model applied at the liquid-solid heat exchange boundary. While the BoilFlowModel I simulation case predicts a constant rate of vaporisation; the sample simulation case using BoilFlowModel II predicts a solely time-dependent rate of vaporisation. The third BoilFlowModel III simulation case predicts both a time-and-space-dependent rate of vaporization. Of these three presented models, the BoilFlowModel III is considered the most realistic because it best approximates the actual heat transfer from the substrate into the liquid by not only accounting for the variation of the surface temperature of the substrate upon liquid-solid contact, but also, it keeps the substrate temperature at its initial value prior to liquid-solid contact at a considered location. Therefore, the BoilFlowModel III is hereby chosen for further analysis and use.

### 4.1 Numerical Solution

In Section 3.9 of Chapter 3, we derived the analytical solution for BoilFlowModel I since this is known and used it to verify the numerical model. For the more complicated BoilFlowModel II and BoilFlowModel III cases, respectively, their respective solution by numerical methods is preferred. Hence for the numerical solution of the vaporizing flow models presented in Sections 4.0.1 and 4.0.2, the explicit finite difference method of solution as discussed in Section 3.8 of Chapter 3 is used. We first optimized the grid size by starting with a 100 point grid. Figure 4.5 shows the relative error plots based on the BoilFlowModel II and BoilFlowModel III, respectively. The relative error was based on calculation at different times and depths but we have chosen, in Figure 4.5, to evaluate the relative error at a depth of 50 mm. The relative errors evaluated at other positions were similar to the trends observed in Figure 4.5. In respect of the BoilFlowModel III case, an error of 14 percent, 5.3 percent and 1.8 were estimated upon increasing the number of grid points from 100, 200, 400, respectively. When the number of grid point is increased to 500, a small error is obtained with a grid size of 10 mm. Thus, we believe that 500 grid points would give sufficient accuracy for the BoilFlowModel III solution. Figure 4.5 also indicates that the relative errors obtained for the BoilFlowModel II solution are

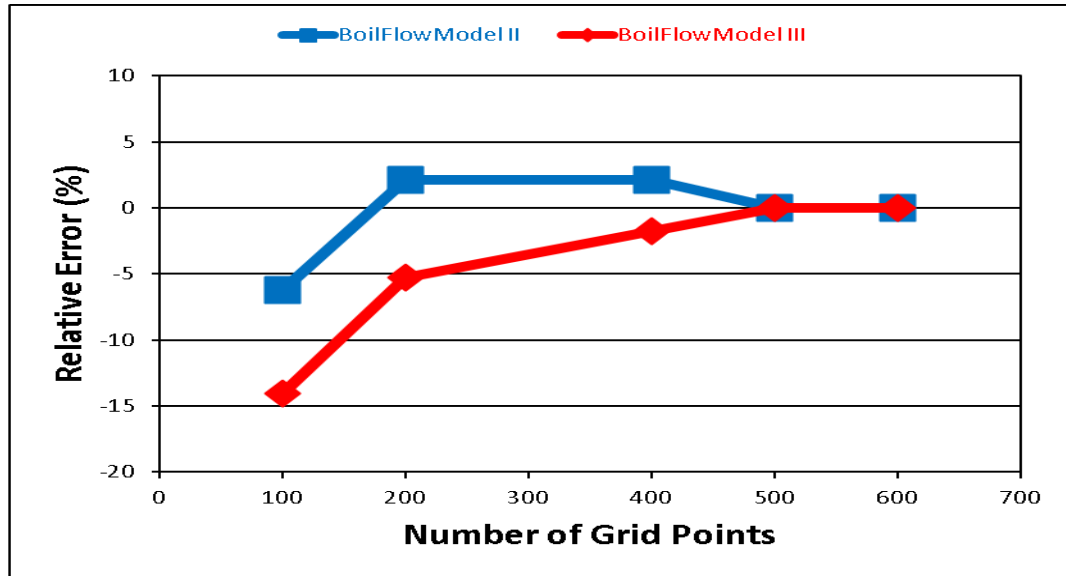


**Figure 4.5:** A graph showing percent error estimates for BoilFlowModel II and BoilFlowModel III, respectively. The error estimation is based on the predicted vapour mass fraction at certain times and depth. The number of grid points was increased from 100 to 500. A grid size of 10 mm (500 grid points) was adjudged to be sufficient for all simulations.

smaller than those obtained for BoilFlowModel III solution with a 6.3 percent relative error observed for the first 100 grid points (grid size=50 mm). The percent error estimation for the BoilFlowModel II simulation case decreased with increasing number of grid points and it is evident from the blue curve of Figure 4.5 that 500 grid points (grid size=10 mm) was, again, sufficient for the BoilFlowModel II simulations. The relative error,  $\zeta$ , estimates were obtained based on a relationship given by Equation (4.8),

$$\zeta_i^{GPT} (\%) = \frac{|\varphi_i^{GPT} - \varphi_i^{A-US}|}{\varphi_i^{A-US}} * 100 \quad (4.8)$$

where  $\zeta$  is the relative error (%),  $\varphi_i^{GPT}$  is the vapour mass fraction using increasing number of grid points between 0 and 500, and  $\varphi_i^{A-US}$  is the solution based on the analytical solution of the unsteady-state heat conduction equation (i.e. Equation 4.11 of Section 4.2.1 is assumed to be the analytical solution to BoilFlowModel II while the analytical solution to BoilFlowModel III is unknown but was assumed in this work to be equivalent to the numerical solution at the chosen 500 point grid). To calculate the wetted surface temperature distribution

in the substrate, the discretized heat conduction equations were solved, then coupled to the advection equation. A successful stability and convergence test under a set time step of 0.0001 seconds yielded transient solutions of the derived partial differential equations (PDE).

## **4.2 Model Comparison with TOUGH2**

To demonstrate the ability of the chosen BoilFlowModel III to accurately represent the derived and implemented mathematical model, a model verification study using a commercial software (TOUGH2) (Pruess, et al., 1999) is presented in this section. This verification study is based on a hypothetical problem involving constant water injection into an air-saturated porous medium. The liquid water was chosen to allow direct comparison with the TOUGH2 (Pruess, et al., 1999) simulation runs. The presentation in this section is split into four sections. Section 4.2.1 gives a brief description of the TOUGH2 software, including a discussion of the trial function heat exchange solution implemented in the code. Section 4.2.2 presents the problem statement and the objectives of the present model comparison study. In Section 4.2.3, the TOUGH2 input specifications and parameter values are presented. The final section (Section 4.2.4) compares the results from our BoilFlowModel III solution with those obtained from a TOUGH2 solution at two different discretization lengths.

### **4.2.1 TOUGH2 Description**

TOUGH2 is a standardized and extensively-validated finite volume simulator that solves for the coupled transport of water, vapour, air, and heat in porous and fractured media (Pruess, et al., 1999; Pruess et al., 1996). The mathematical formulation implemented in TOUGH2 is similar to those used in typical multiphase, multicomponent reservoir simulators as can be found in geothermal and reservoir engineering. The governing fluid flow equations incorporate the effects of gaseous diffusion, Darcy flow, capillary pressure, vaporization, and condensation. The code assumes that the principal phases of water, air and rock are in thermodynamic equilibrium at all times with the flow domain generating liquid-, gas-, and two-phase zones. The software has the capacity to handle both saturated and unsaturated flow problems. The thermophysical properties of fluid mixtures needed for assembling the governing mass and energy balance

equations are provided by “equation of state modules” (EOS) modules (Pruess et al., 1999). The thermophysical properties of liquid and vaporized water are represented by the International Formulation Committee’s steam tables (International Formulation Committee, 1967). An ideal gas approximation is used for air and additivity of partial pressures is assumed for air-water vapour mixtures (Pruess et al., 1999).

The mathematical models in TOUGH2 comprise three nonlinear partial differential equations that are simultaneously solved. These equations are essentially the conservation equations for air, water, and energy. Henry’s law accounts for the dissolution of air in water while Darcy’s law (Bear, 1988) accounts for fluid flow. The software has a capacity to simulate fluid flow in one, two or three dimensions because of the general integrated finite-difference methodology upon which the code is based. The nonlinear difference equations can be linearized, for example, by the Newton-Raphson technique and the linearized equations can be solved by the Harwell MA28 matrix solver (Duff, 1977) that stores only the non-zero elements of a matrix so as to reduce core storage requirements for the code. Another method that can be used is the preconditioned gradient solver (Pruess et al., 1999). Further details of the TOUGH2 software can be found in Pruess et al. (1999).

A semi-analytical heat exchange solution is implemented in TOUGH2 which deserves mention here. This solution enables TOUGH2 to handle coupled problems involving conductive heat transfer. The solid matrix provides the heat source that leads to liquid temperature increase and/or vaporization. The heat exchange approach in TOUGH2 is based on the method of Vinsome and Westerfeld (1980) which is essentially a solution that assumes that the temperatures in the conductive zones would vary smoothly regardless of the strength and speed of temperature changes taken place at the heat exchange boundary. The authors ignored the effect of heat conduction parallel to the solid-liquid interface believing that to be less important than that perpendicular to it. Hence, they expressed the lateral temperature profile in the porous matrix as follows,

$$T(x,t) - T_b = (T_{p0} - T_b + p_1 x + q_1 x^2) \exp(-x/\ell) \quad (4.9)$$

where  $T(x,t)$  is the lateral temperature profile or distribution,  $x$  is the distance from the liquid-solid interface,  $T_b$  is the boiling temperature of the liquid,  $T_{p0}$  is the time-varying temperature at the liquid-solid interface,  $p_1$  and  $p_2$  are time-varying fit-parameters, and  $\ell$  is the penetration length for heat conduction, which the authors (Vinsome and Westerfeld, 1980) defined by Equation 4.10,

$$\ell = \frac{\sqrt{\alpha_s t}}{2} \quad (4.10)$$

where  $\alpha_s$  is the thermal diffusivity of the matrix and  $t$  is the time. Since Vinsome and Westerfeld (1980) assign each grid block at the liquid-solid interface with an associated temperature profile given by Equation (4.9), the fitting coefficients  $p_1$  and  $p_2$  are expected to be different for each grid block and are determined concurrently with the flow simulation subject to two physical constraints (Pruess et al., 1999), namely: (i) continuity of heat flux across the boundary at  $x=0$  and (ii) energy conservation for the entire system consisting of permeable and conductive-only zones (Pruess et al., 1999). With the knowledge of  $p_1$  and  $p_2$  and the heat flux at the heat exchange boundary, the corresponding vaporization rates and liquid penetration depths at different locations can then be readily predicted. We wish to point out here that our analytical solution given as Equation (4.6) only determines the temperature variation at the surface of the solid matrix. The full analytical solution for the temperature distribution as a function of time,  $t$ , and distance,  $x$ , is given by (Carslaw and Jaeger, 1959b),

$$T(x,t) = T_b + (T_{s0} - T_b) \operatorname{erf} \frac{x}{(4\alpha_s t)^{1/2}} + \exp \left( \frac{hx}{k_s} + \frac{h^2 t}{k_s \rho_s C_{ps}} \right) \operatorname{erfc} \left( \frac{x}{(4\alpha_s t)^{1/2}} + \frac{h(t)^{1/2}}{(k_s \rho_s C_{ps})^{1/2}} \right) \quad (4.11)$$

where  $T$  is the temperature distribution (K),  $T_{s0}$  is the initial temperature of the solid matrix,  $k_s$  is the thermal conductivity of the substrate (W/m/K),  $h$  is the heat transfer coefficient (W/m<sup>2</sup>/K),  $C_{ps}$  is the specific heat capacity of the porous matrix,  $\alpha_s$  is the thermal diffusivity,  $x$  is the distance from the liquid-solid boundary,  $t$  is time (sec) and  $T_b$  is the liquid boiling temperature (K). Although Equation 4.9 may seem to have similar features with equation 4.11. However, there are a few advantages which Equation 4.11 has over Equation 4.9 one of which is that they do account for the thermophysical properties of the fluid that flows through the substrate. Another advantage is that trial-function errors are eliminated if Equation 4.11 is used to calculate the temperature distribution.

#### 4.2.2 Problem Statement and Objective

This sample problem is a hypothetical study to test an analogue of LNG. We have examined a 2.6 m length vertical and homogenous sandstone porous layer into which a constant volume of water is injected from the top. The water is assumed to reach its boiling point (373 K) right from the point of injection. The air-saturated porous layer is assumed to be at a uniform initial temperature of 493 K with the saturating air considered a passive bystander and thus not accounted

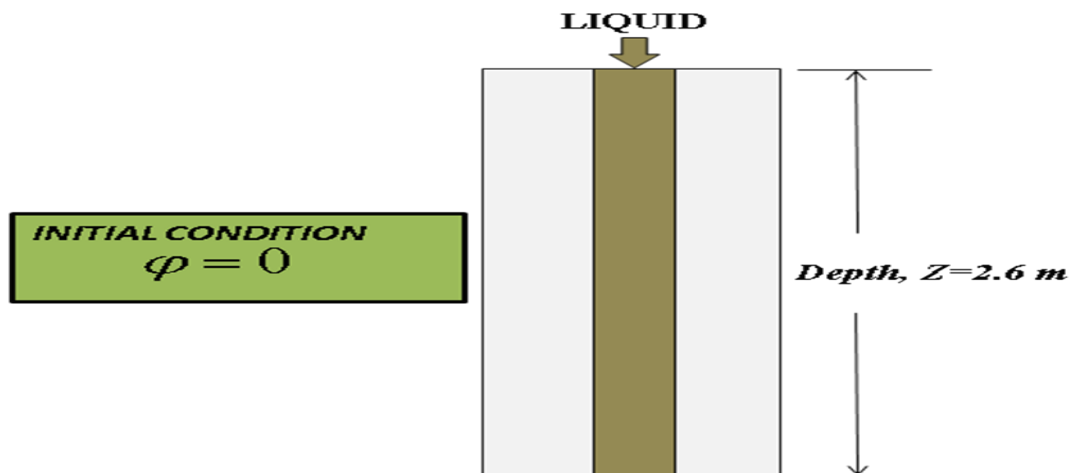


**Figure 4.6:** A simple schematic of the vaporizing flow problem indicating initial condition and the depth of porous layer.

for. At time  $t=0$ , the magnitude of the vapour mass fraction at all considered locations inside the porous layer equals zero since liquid has neither entered the

porous layer nor approached a considered location. At time greater than zero, liquid enters the porous layer and heat transfer from the substrate causes vigorous vaporization of the injected liquid. The problems are to predict the vapour mass fraction as a function time and depth as well as to track a propagating liquid front and/or interface as a function of time and depth. A simple schematic showing the initial condition of the problem and the depth of porous layer is shown in Figure 4.6.

### 4.2.3 Input Specifications and Parameter Values

Two discretization cases using 650 and 2600 number of grid blocks are presented in this section. The 650 grid blocks used for the 4 mm discretization case is shown in Figure 4.7 (a), it consists of one column of grid blocks labeled G1 through G650 and an adjacent column of grid blocks labeled P1 through P650. The attachment of the adjacent column of grid blocks is to prevent pressure build-up in the main model domain that is usually caused by the vaporization of water. The attached grid blocks suck out all produced vapour from the main column of grid blocks thereby preventing the vapour from interfering with the

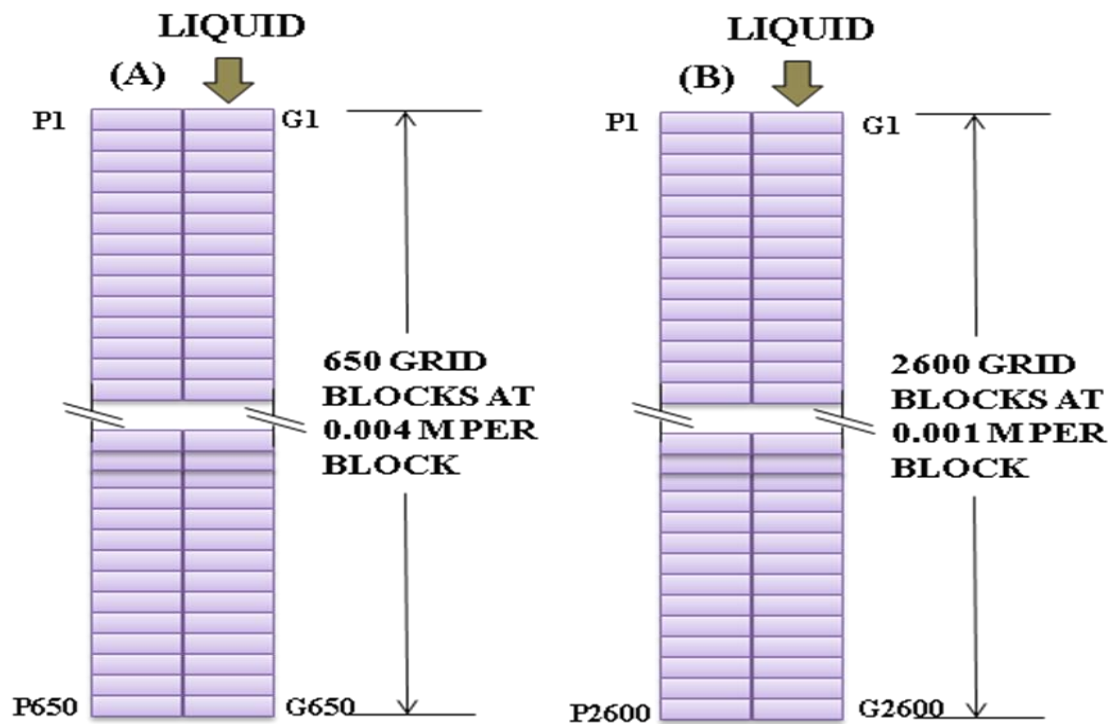


**Figure 4.7:** TOUGH2 discretization used for the comparative study.

flow of liquid in the main column. The 1 mm discretization case involving 2600 grids is illustrated in Figure 4.7 (b). Again, we assign grid blocks G1 through G2600 in a similar fashion as done for the 4 mm discretization. Adjacent grid blocks, P1 through P2600 were each attached to the main column of grids in order to release vapour into it, as already explained during the discussion of the 4 mm discretization case. In addition, such adjacent grid blocks would not be required in our finite difference models having initially assumed that the generated vapour would dissipate sufficiently fast. Table 4.1 details the relevant parameter values used for the present model verification study.

**Table 4.1:** Parameter values used for the comparative study.

| <i>Parameters used for model verification work</i>                | <i>Value</i>          |
|-------------------------------------------------------------------|-----------------------|
| Initial Temperature of Porous Matrix (K)                          | 473                   |
| Liquid Boiling Temperature (K)                                    | 373                   |
| Latent Heat of Vaporization of Liquid (J/kg)                      | 400820                |
| Gravitational Acceleration ( $\text{m/s}^2$ )                     | 9.81                  |
| Injection Velocity (m/s)                                          | 0.023                 |
| Height of Porous Layer (m)                                        | 2.6                   |
| Specific Heat Capacity of Porous Matrix (J/kg/K)                  | 900                   |
| Film Boiling Heat Transfer Coefficient ( $\text{W/m}^2\text{K}$ ) | 155                   |
| Thermal Conductivity of Porous Matrix ( $\text{W/m/K}$ )          | 1.2                   |
| Thermal Diffusivity ( $\text{m}^2/\text{sec}$ )                   | $5.25 \times 10^{-5}$ |
| Permeability (Darcy)                                              | 3.04                  |
| Density of Porous Matrix ( $\text{kg/m}^3$ )                      | 2540                  |

#### 4.2.4 TOUGH2 and BoilFlowModel III Results Comparison

Both the BoilFlowModel III and TOUGH2 successfully ran the boiling flow problem to completion and results of the 4 mm discretization case are presented in Figure 4.8 (a). It is clearly evident from the graph that the depth at which the first interface is generated (or at which the liquid has penetrated) for the BoilFlowModel III compare very well with the TOUGH2 solution results. The interface location for both models are at 38 mm from the injection point which is essentially a very small discrepancy at this location. However, at relatively shorter distances from the injection point, a small discrepancy is evident in the magnitude of the vapour mass fraction. For instance, the magnitude of vapour mass fraction at a shorter depth of 34 mm based on a BoilFlowModel III solution

is greater than the value obtained from a TOUGH2 solution by about 5 percent. But we can also observe that this 5 percent discrepancy became even smaller at a much shorter depth from the injection point, say, 14 mm. A relatively larger discrepancy is also observed close to the injection zone which may be linked to the different boundary conditions applied by the two models. The identified discrepancy may also be related to some numerical dispersion which we have sought to reduce by using a smaller discretization length. In any case, we still believe that despite the identified discrepancy, the result from the BoilFlowModel III solution using the 4 mm discretization length demonstrates a generally acceptable agreement with the TOUGH2 solution.

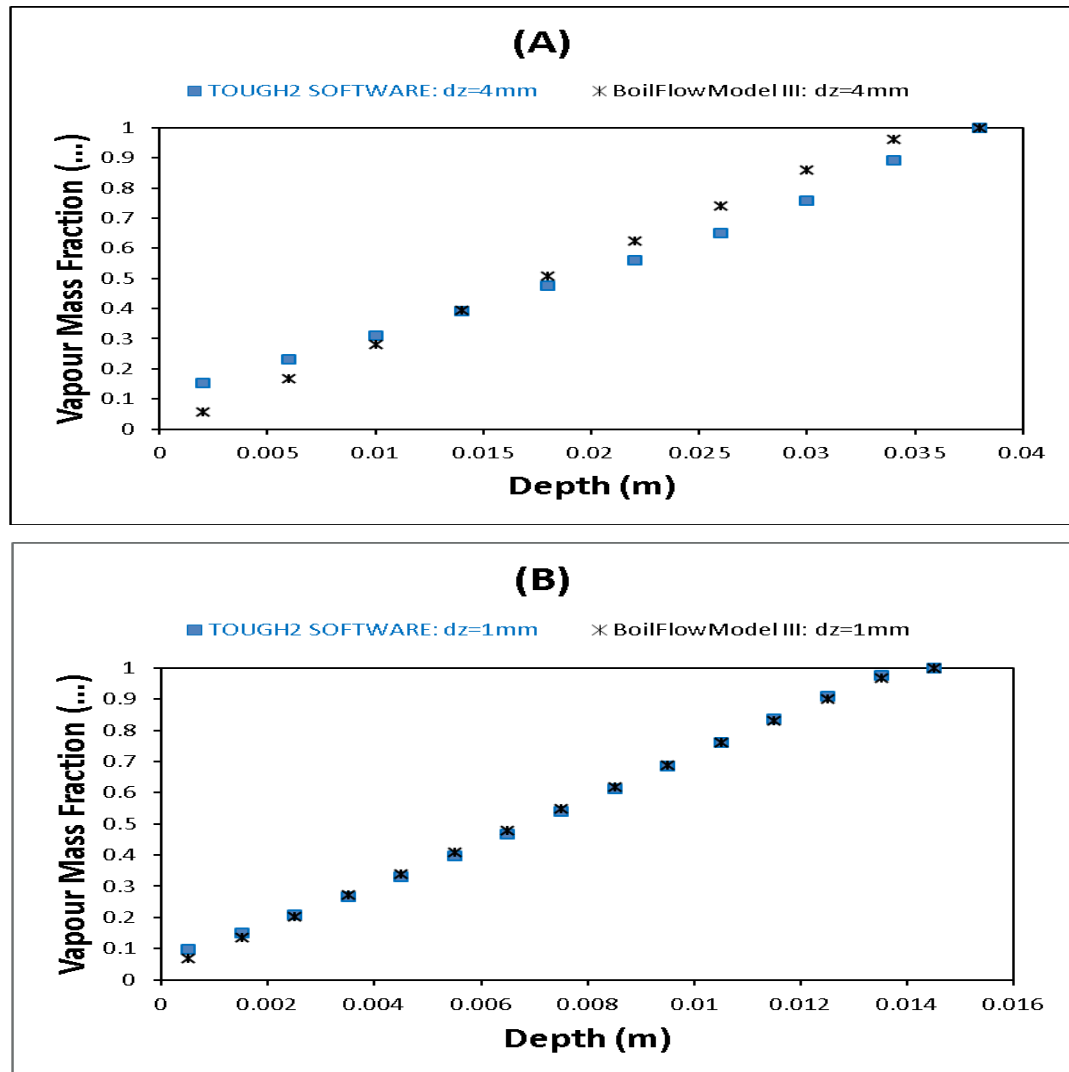


**Figure 4.8:** Vapour mass fraction profiles for the vaporizing flow problem at  $t = 5$  sec. (a) Results using a 4 mm vertical discretization length. (b) Results using a 1 mm vertical discretization length.

The results for the second discretization case is presented in Figure 4.8 (b) where the vapour mass fraction profiles obtained from a TOUGH2 solution is compared with those from the BoilFlowModel III solution using a smaller discretization length of 1 mm. The results indicate that the predicted interface location for both the BoilFlowModel III and TOUGH2 solutions are similar with a very small discrepancy at the interface position. Furthermore, the numerical dispersion earlier observed with the 4 mm discretization length is now significantly reduced. The results demonstrate that the BoilFlowModel III solution better tracked the front propagation and interface locations, suggesting a significant improvement in the prediction of the profiles of vapour mass fraction deriving from the vaporizing flow problem. Overall, we believe the agreement between the TOUGH2 results obtained with the 1 mm discretization case and the BoilFlowModel III results is very good. However it also deserves mentioning here that, in terms of predicting the heat transfer aspects of the vaporizing flow problem, the TOUGH2 heat exchange solution technique may be questionable as already discussed under Section 4.2.1.

### 4.3 Sample BoilFlowModel III Simulation Problem

In this section, we will use the BoilFlowModel III to evaluate the evolution of liquids flowing through porous media of depth 5 meters and with heat transfer from the bounding solid matrix. A tabular listing of liquid and substrate thermophysical properties used in the simulation work discussed in this chapter and the next chapters are listed in Table 4.2, 4.3 and 4.4. In respect of porous or

**Table 4.2:** Properties of LNG, LPG, H<sub>2</sub>O, LH<sub>2</sub>, and LN<sub>2</sub> (NIST WEBBOOK, 2011).

| <i>Liquid Property</i>       | <i>LNG</i>           | <i>LPG</i>         | <i>H2O</i>          | <i>LH2</i>           | <i>LN2</i>           |
|------------------------------|----------------------|--------------------|---------------------|----------------------|----------------------|
| Thermal Conductivity (W/m/K) | 0.193                | 0.017              | 0.58                | 0.099                | 0.138                |
| Density (kg/m <sup>3</sup> ) | 422                  | 580                | 961                 | 70.8                 | 800                  |
| Viscosity (Pas)              | 113x10 <sup>-6</sup> | 8x10 <sup>-5</sup> | 29x10 <sup>-6</sup> | 135x10 <sup>-6</sup> | 158x10 <sup>-6</sup> |
| Heat of Vaporisation (KJ/kg) | 512                  | 428                | 401                 | 454                  | 201                  |
| Specific Heat Capacity (J)   | 3350                 | 1630               | 4210                | 9690                 | 2000                 |
| Surface Tension (N/m)        | 0.014                | 0.016              | 0.059               | 0.002                | 0.009                |

**Table 4.3:** Thermal properties of soil (SO), Glass Wool (GW), Sandstone (SS) and Stainless Steel (SST) used in the present work (Shaw and Briscoe, 1978, NIST WEBBOOK).

| <i>Substrate Property</i>                                | <i>SO</i> | <i>GW</i> | <i>SS</i> | <i>SST</i> |
|----------------------------------------------------------|-----------|-----------|-----------|------------|
| Thermal Conductivity (W/m/K)                             | 5.18      | 0.038     | 1.20      | 15.0       |
| Density of Substrate (kg/m <sup>3</sup> )                | 1600      | 24        | 2540      | 7860       |
| Thermal Diffusivity (10 <sup>-6</sup> m <sup>2</sup> /s) | 3.88      | 2.26      | 0.525     | 3.82       |
| Specific Heat Capacity (J/kg/K)                          | 835       | 700       | 900       | 500        |

**Table 4.4:** Properties of vapour film used for simulations (NIST WEBBOOK, 2011).

| <i>Vapour Property</i>       | <i>LNG</i> | <i>LPG</i> | <i>H2O</i> | <i>LH2</i> | <i>LN2</i> |
|------------------------------|------------|------------|------------|------------|------------|
| Therm. Conductivity (W/m/K)  | 0.021      | 0.012      | 0.025      | 0.099      | 0.017      |
| Density (kg/m <sup>3</sup> ) | 1.02       | 2.16       | 0.517      | 0.165      | 1.93       |
| Specific Heat Capacity (J)   | 2100       | 1440       | 1990       | 12500      | 1050       |
| Critical Temperature (KJ/kg) | 191        | 370        | 647        | 33.1       | 126        |

penetrable substrate properties, we have included those of soil, glass wool, sandstone. In respect of non-penetrable substrate properties, we added those of steel. During film and transition boiling regimes, vapour films are present and an estimation of the properties of the vapour film is required. The vapour film property values are listed in Table 4.4 for the five different liquid types investigated in this chapter.

#### 4.3.1 Vaporizing Flow before Interface Generation

Figure 4.9 (i) presents the evolution of the injected liquid as a function of time and depth. The front location after 2 seconds is 230 mm with approximately 22 percent of the injected liquid vaporizing. Further down the porous layer at a depth of 490 mm, 39 percent of the liquid has already vaporized under 4 seconds of injection. The liquid penetration depth increased within 6 seconds to 760 mm with approximately 52 percent of the liquid vaporizing. The observed trends in Figure 4.9 (i) indicate that the liquid front penetration depth increases with time. This tendency for the front to penetrate deeper into the porous medium can be attributed to a decreasing superheat of the porous system resulting from a decreasing thermal gradient. Figure 4.9 (ii) through 4.9 (iv) show a better visualization of what is happening in Figure 4.9 (i). The interface is clearly absent at the identified times since the magnitude of the vapour mass fraction is still less than unity at these times. Also notice that we have indicated that the

population of the vapour bubbles in the two-phase zone (Zone 2) decreases with time suggesting that less heat is being released for the vaporization of the liquid.

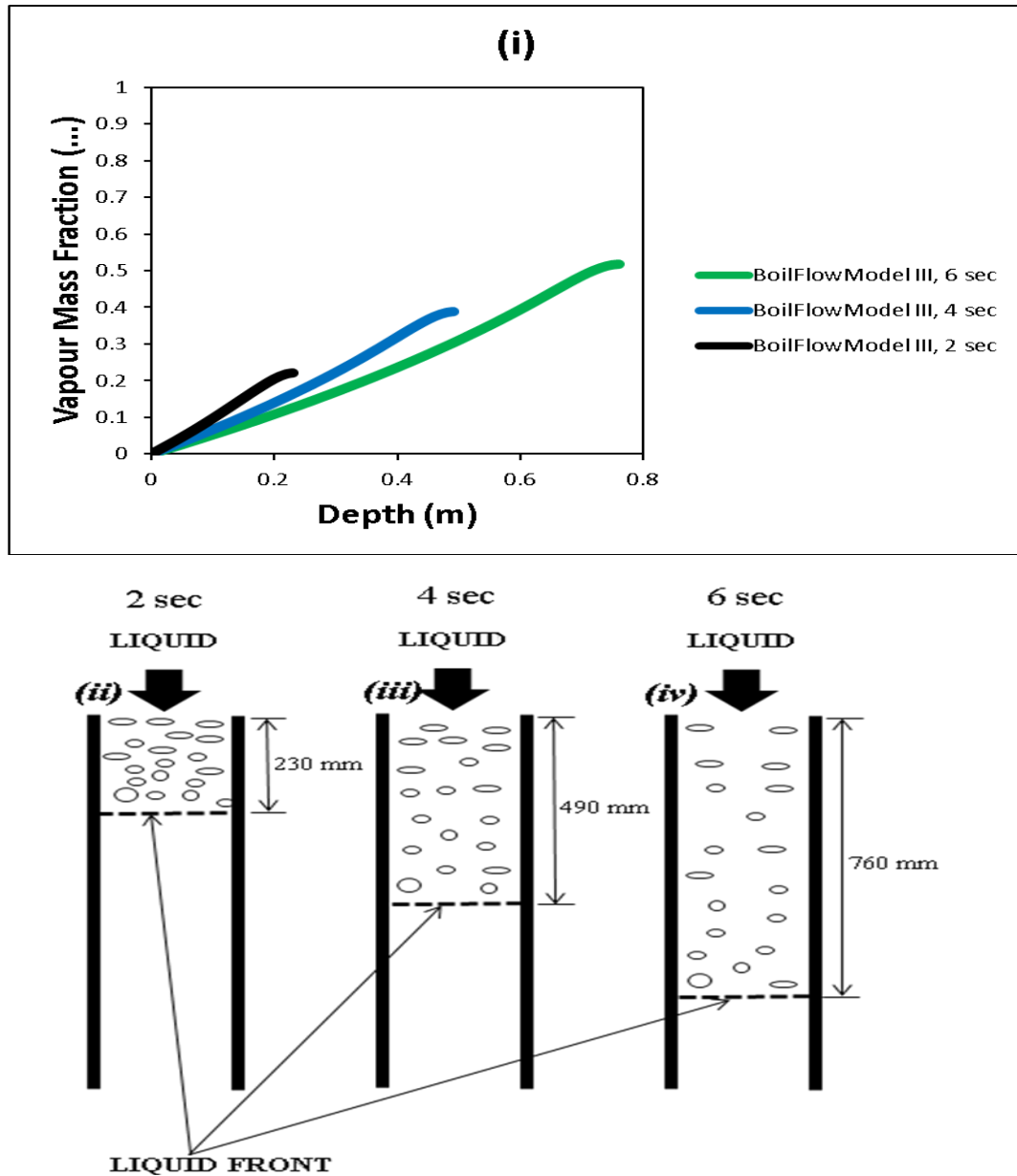


**Figure 4.9:** (i) vapour mass fraction profiles as a function of depth before interface generation; (b) descent of liquid within 2, 4, and 6 seconds of injection.

#### 4.3.2 Vaporizing Flow after Interface Generation

Here, we examine the vaporizing flow behaviour following interface generation. From Figure 4.10, we notice that the interface reached a depth of 20 mm just within 2 seconds of injection. Checking for the interface position at 12 seconds, we find that the interface moved to a new depth of 60 mm. Further descent of the

interface to a new depth of 160 mm was observed after 22 seconds. Like the front movement discussed in Section 4.3.1, the interface also moves with time so long as a constant injection of the liquid into the porous layer is maintained.

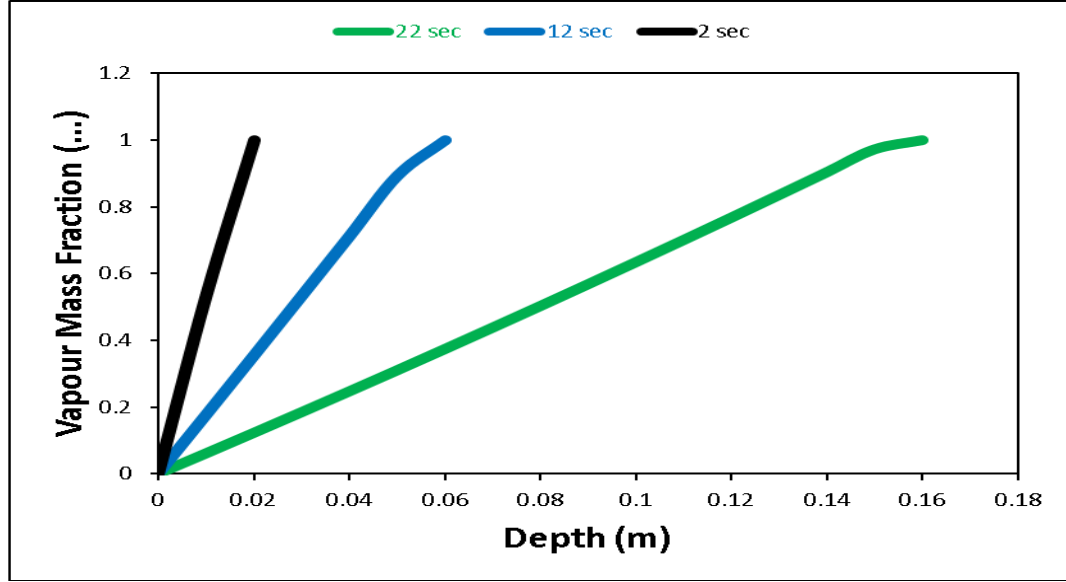


**Figure 4.10:** Vapour mass fraction profiles as a function of depth after interface generation. The interface movement suggests that liquid penetration increases with time.

#### 4.3.3 Vaporizing Flow with Heat Flux decreasing to Zero

The injected liquid will evolve differently when the wetted surface temperature of the substrate has cooled to the boiling temperature of the flowing liquid. Such a scenario implies that heat flux at the designated location will be equal to zero. The simple schematic depicted in Figure 4.1(i) illustrates the evolution of the zones inside the porous layer in the event of heat flux decreasing to zero. Figure 4.11(ii) shows a corresponding schematic plot of vapour mass fraction as a function of depth. The horizontal axis of Figure 4.11(ii) is the axis of vapour mass fraction while the vertical axis is the axis of liquid penetration depth within the porous layer as the liquid flow evolves. Relating the scenario to the present consideration in which the cooled substrate may result in a near zero heat flux at certain depths, we would expect that the evolution of the zones in the porous layer would be slightly different because of the development of a fourth distinct zone. Specifically, a fourth single-phase liquid zone, henceforth referred to as Zone 4, is generated. This Zone 4 is identified as the region h-a-b-g in Figure 4.11(ii). The presence of Zone 4 implies that a direct liquid-solid contact is now

occurring. Ideally, the liquid-vapour two-phase zone would be the first zone to be generated (refer to Section 3.11 of Chapter 3) and we have called it Zone 1. In Figure 4.11(ii), this Zone 1 would be the region  $g-b-b_1-g_1$ . The next zone is the single-phase vapour zone – the so-called Zone 2 (i.e. region  $g_1-b_1-c-f$  of Figure 4.11). We have referred to a third air-filled zone and called it Zone 3, which for the purpose of this work, is considered a passive spectator. Figure 4.11 (ii) identifies Zone 3 as the region  $f-c-d-e$ . The development of Zone 4 can be visualized from results of a sample problem presented in Figure 4.12. We plotted vapour mass fraction as a function of depth at three different times. The interface location after 22 seconds is at 160 mm. After 42 seconds, the interface

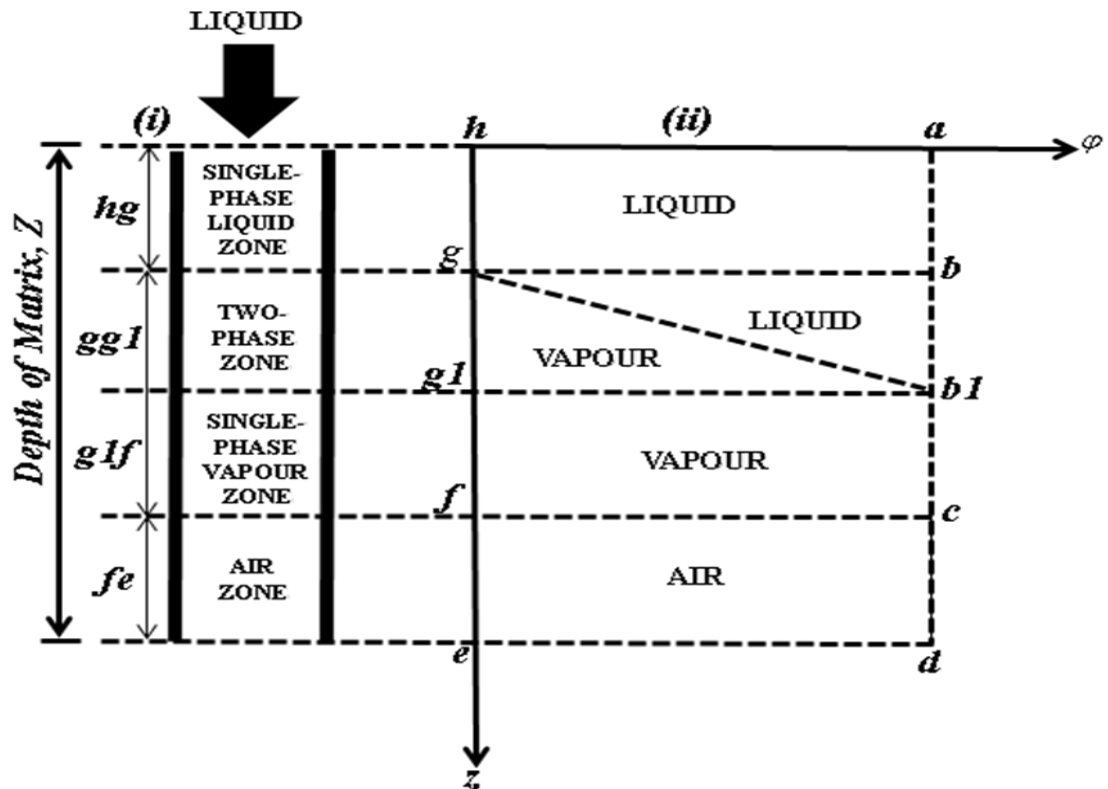


**Figure 4.11:** A schematic of the vaporizing flow problem showing a newly-generated single phase liquid zone – the so-called Zone 4. The development of Zone 4 begins only when the heat flux at a considered location has decreased to zero.

has penetrated to a depth of 1750 mm. A deeper penetration is observed after 62 seconds and the interface location at this time is at 4920 mm, very close to the base of the porous layer. Again, an important observation in Figure 4.12 is the presence of Zone 4 after 62 seconds. For vapour mass fraction profiles at 62 sec,

Zone 4 can be located between the depth  $z=0$  and  $z=1$  mm and the flowing liquid is within this region, in direct contact with the surface of the solid matrix. Vapour is no longer being produced and the velocity of the liquid is expected to increase to magnitude close to the initial injection velocity. We will attempt to demonstrate this in Section 4.3.4. Please note that zones 2 and 3 have been deliberately omitted from Figure 4.12 because our primary objective is to study the propagation of the liquid front and not the dynamics of the generated vapour. Zone 1 falls within the region  $z=1$  mm and  $z=4920$  mm.

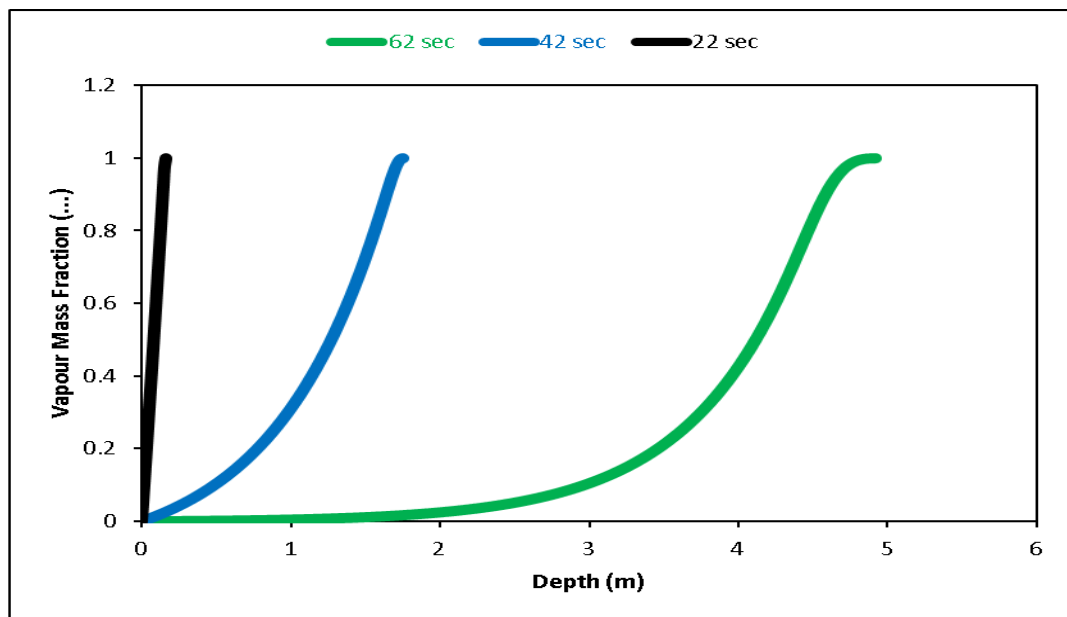


**Figure 4.12:** Vapour mass fraction profiles as a function of depth after interface generation. The presence of Zone 4 can be visualized after 62 seconds and is identified as the region  $z=0$  mm and  $z=1$  mm.

#### 4.3.4 Interface Movement

A better illustration of maximum depth penetrated by the flowing liquid under a given duration can be achieved through the examination of the interface position and movement (see blue curve of Figures 4.13 (a) and (b)). Figure 4.13 (a) also presents a comparison of the liquid penetration depth profiles (interface movement) with or without vaporization. Considering the case of liquid flow without vaporization, it is clear from Figure 4.13 (a) that a flow duration of 62 seconds will result in the front reaching a depth of 9.3 m (a depth far larger than the actual depth of the porous layer), whereas, the liquid will only reach a depth of 4.92 m if the effects of vaporization is included. This impact of vaporization on

liquid penetration rates is clearly evident from Figure 4.13 (a) where it can be seen that the vaporization effect is strongest within the first 20 seconds following injection. Interestingly, we also observe an increase in liquid penetration rates after 20 seconds suggesting that the impact of vaporisation has started to decline as the substrate cools towards the temperature of the liquid and the vaporization rate decreases. It is also evident that the interface penetration rate with vaporization will at some point closely resemble the rates observed for the non-vaporizing liquid flow case (see black of Figure 4.13(a)). Figure 4.13 (b) shows the corresponding interface velocity profiles based on the plot of interface movement

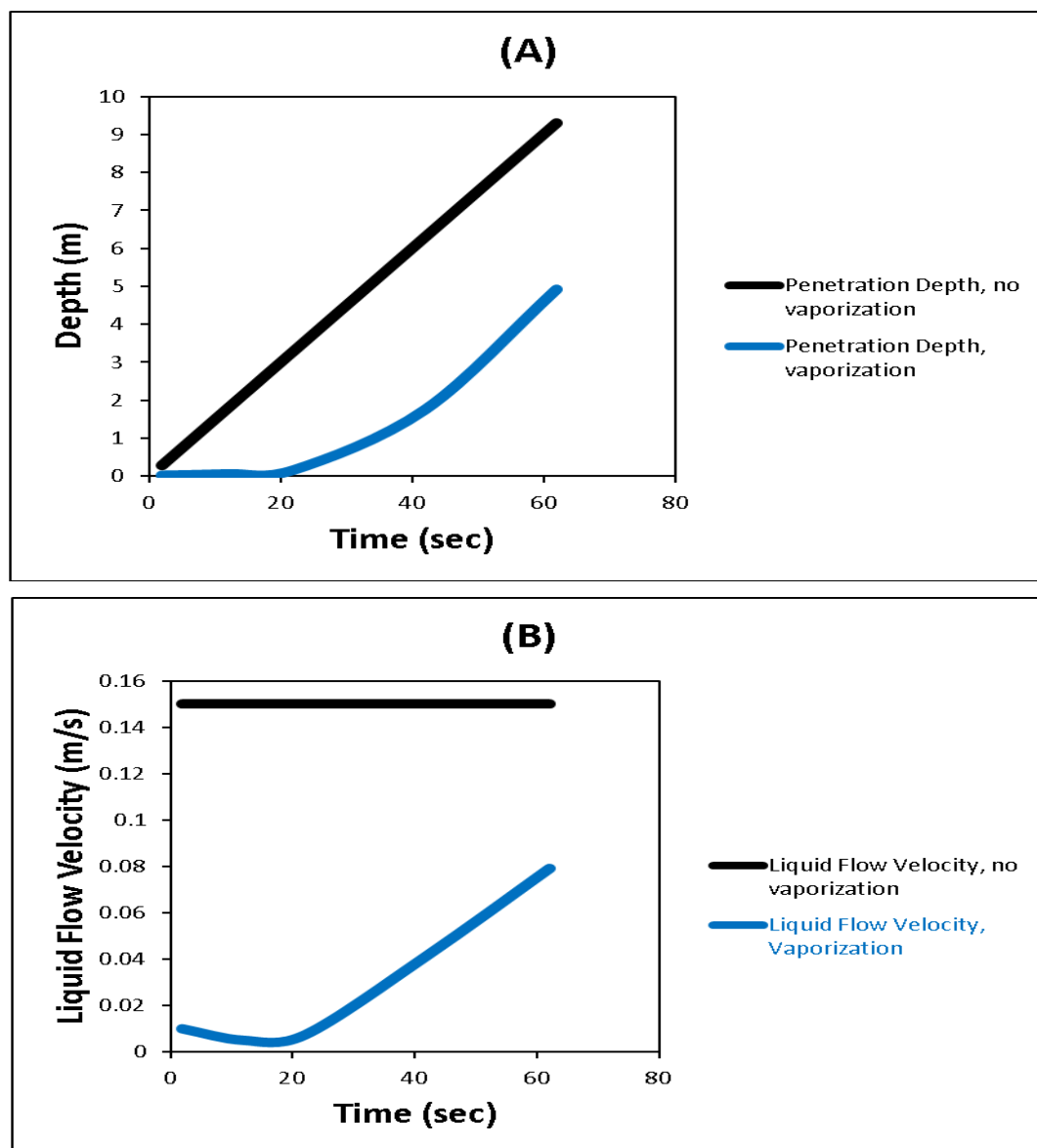


**Figure 4.13:** Vaporizing flow problem: (a) Interface movement with time (b) Flow velocity profiles as a function of time after interface generation.

shown in Figure 4.13 (a). Obviously, the flow velocity profiles in the absence of vaporisation is unchanged while the profiles predicted for a vaporizing flow rapidly decreased from the initial injection velocity 0.15 m/s to a relatively small value 0.01 m/s just under 2 seconds. The reduced velocity of the interface, and in effect, the flowing liquid is caused by the strong vaporization effect which hindered the faster propagation of the liquid interface inside the porous layer. Because of this initially strong vaporization effect, a gradual decrease in liquid front and/or interface velocity is observed in about 20 seconds when a decrease from 0.01 m/s to 0.005 m/s is observed. As the strong vaporization effect begins to weaken after 20 seconds, the liquid flow velocity is observed to rapidly increase ultimately reaching a value of 0.079 m/s after an additional 40 seconds has elapsed. Thus, the velocity of the liquid as it flows inside the porous or penetrable media is clearly influenced by liquid vaporization rate.

#### 4.3.5 Evolution of Zone 2

It might be instructive to also visualise the evolution of Zone 2 at different times following injection. Figure 4.14 shows the interface position after 12 seconds. At this time, the interface is positioned at a depth of 60 mm and the extent of Zone 2 after 12 seconds is approximately 1710 mm. At a later time of 22 seconds, the interface penetrated deeper to a depth of 160 mm with the corresponding extent

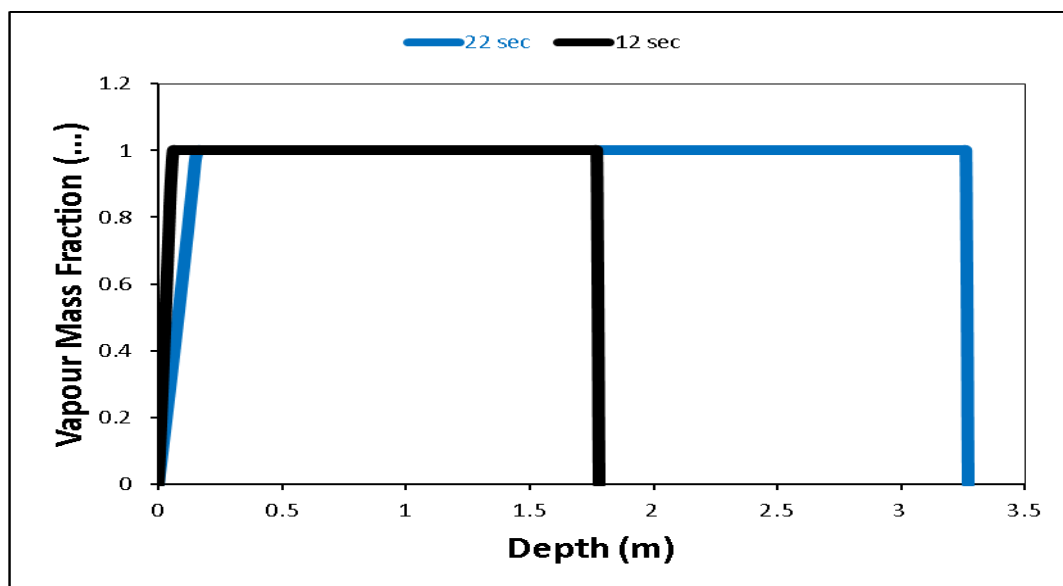


**Figure 4.14:** Vapour mass fraction profiles as a function of depth after interface generation with the evolution of the single-phase vapour zones (Zone 2) shown.

of Zone 2 increasing to 3100 mm. It is evident that the reason for the larger extent of Zone 2 after 22 seconds is because of the relatively larger rate of vapour production where the vapour, once produced, moves ahead of the interface. Moreover, deeper interface penetration also suggests that a larger contact area for heat transfer is available. This explains the reason for the larger amount of vapour produced at the later time of 22 seconds.

## 4.4 Parametric Effects

The influence of some parameters on the vaporizing flow behaviour are investigated in this section. We are particularly interested in understanding how changes in the magnitude of these selected parameters will influence substrate cooling rates, liquid vaporization rates, vapour mass fraction and liquid (interface or front) penetration rates. We will first examine in Section 4.4.1 sensitivity cases of high and low initial substrate temperatures and compare the BoilFlowModel III results with those obtained using the BoilFlowModel I presented in Chapter 3.

### 4.4.1 Influence of Initial Substrate Temperature

Figure 4.15 (a) shows the trends in interface movement when the initial temperature of the substrate is 293 K. From Section 4.3.4, we have already shown that, in the absence of vaporization, the front will reach a depth of 9.3 meters after 62 seconds. When vaporization is accounted for, the liquid penetrated to a depth of 4.92 meters. The blue curve of Figure 4.15 (a) shows the interface behaviour when simulations are performed using the steady-state BoilFlowModel I, where it is seen that the interface did not penetrate beyond 0.2 m. This is expected since, as already explained in Chapter 3, that the interface once generated using BoilFlowModel I does not move. The corresponding interface velocity profiles is presented in Figure 4.15 (b). Firstly, the vaporization process based on BoilFlowModel I is so strong that a rapid decrease in interface velocity is observed. In contrast, the interface velocity profiles based on BoilFlowModel III indicates an initial slow decrease, as earlier discussed, until after approximately 20 seconds when a relatively rapid increase in velocity is observed.

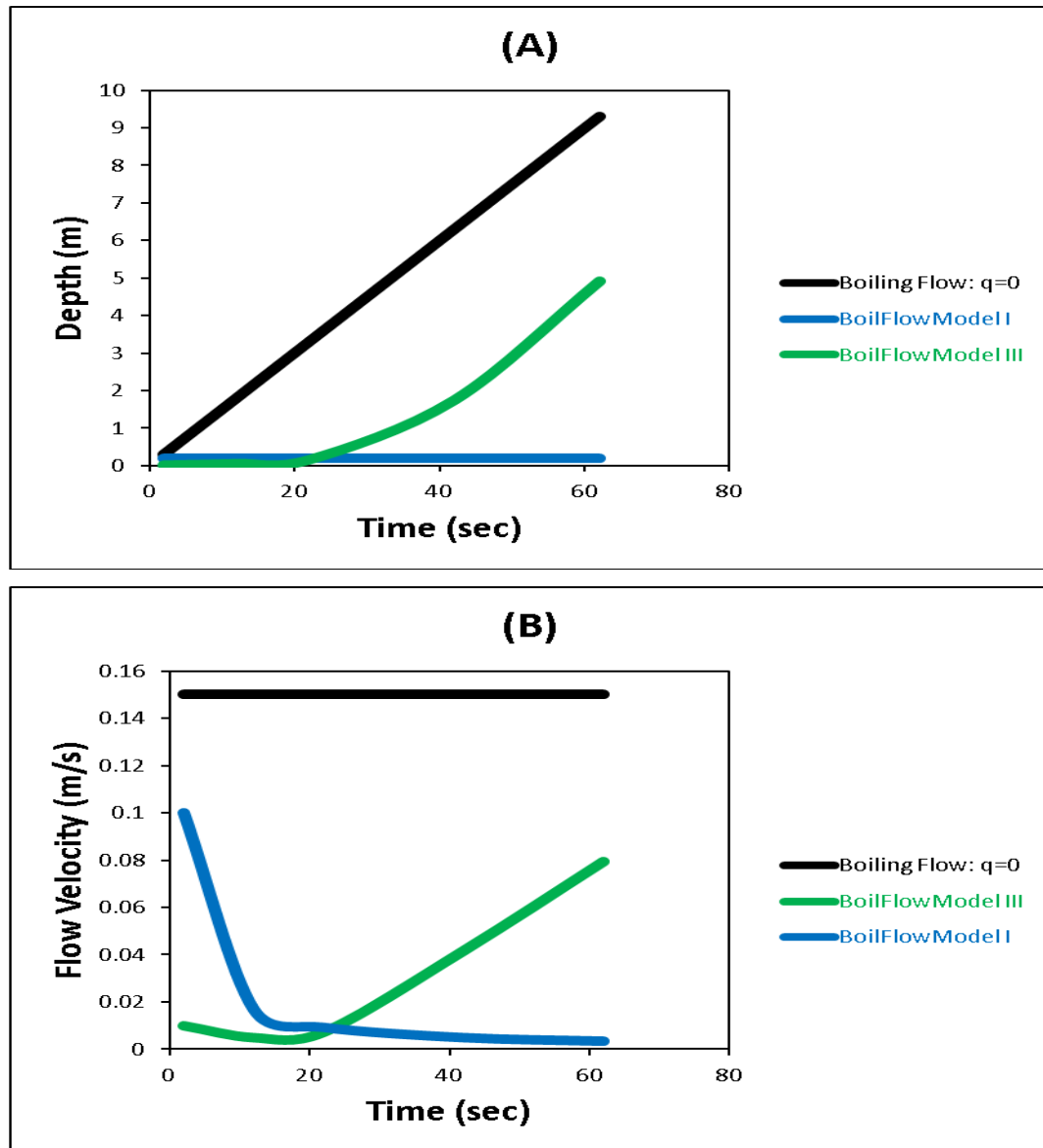


**Figure 4.15:** Evolution of the boiling flow process at higher initial substrate temperature of 293 K: (a) Interface movement as a function of time and (b) flow velocity profiles as a function of time.

The situation is different at a relatively low initial substrate temperature of 115 K as can be seen from Figure 4.16 (a) and 4.16 (b). Considering a total flow duration of 12 seconds, we confirm from Figure 4.16 (a), that the liquid would penetrate to a depth of 1.8 m without the inclusion of vaporization effects (see black curve of Figure 4.16(a)). Examining the flow cases with vaporization, simulation results from using the steady-state BoilFlowModel I reveals a maximum liquid penetration depth of 0.89 m within the considered flow

duration. It was also possible to visualize the front movement at this lower initial temperature case, unlike the 293 K case that led to instant interface generation due to relatively high vaporization rate. The front moved deeper into the porous layer until the generation of the interface limits the advance of the front, suggesting that the liquid has, at this point, reached its maximum penetration depth inside the porous layer. The leading edge of the liquid remained at this depth even though time progressed (see blue curve of Figure 4.16(a)). When the BoilFlowModel III is used, a deeper penetration of the liquid is observed with the

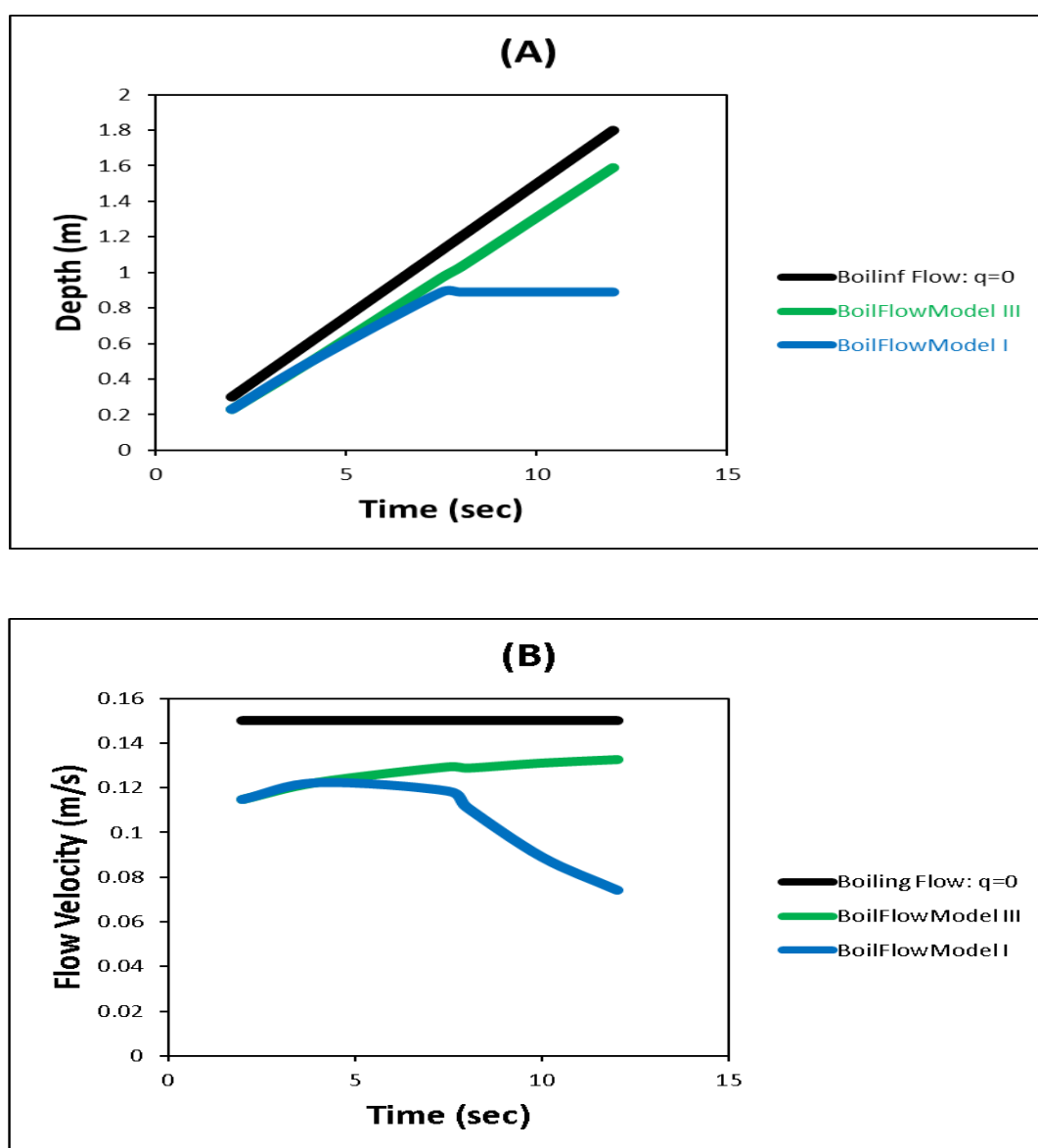


**Figure 4.16:** Evolution of the vaporizing flow process at a relatively lower initial substrate temperature of 115 K: (a) Front and Interface movement as a function of depth and (b) Liquid front/interface velocity profiles as a function of time.

interface located at 1.59 m.

The corresponding front and/or interface velocity profiles are depicted in Figure 4.16 (b). Obviously, the unperturbed flow retains the initial and constant velocity profile of 0.15 m/s throughout the flow duration. Using the BoilFlowModel I, we found that, initially, a rapid reduction in front/interface velocity from the initial injection value of 0.15 m/s to a velocity of 0.115 m/s after 2 second occurred. The velocity was observed to slowly increase before gradually decreasing. A more rapid decrease of the liquid velocity towards a final value of 0.074 m/s with the considered flow duration is observed. For the BoilFlowModel III simulation case, a flow velocity of 0.115 m/s after 2 seconds is revealed at first. Later, a slow increase from 0.1225 m/s to a final value of about 0.133 m/s is observed. The profiles of velocity explains the trends observed in the interface movement plots. These results suggest that liquid penetration rates as well as liquid vaporization rates are affected by the magnitude of the initial temperature of the solid matrix. The lower the initial temperature, the lower the liquid vaporization rate and the higher the liquid penetration rate. For BoilFlowModel I, this behaviour would only occur prior to interface generation. These results are instructive in the sense that they provides an indication of how the rate of vaporization and penetration are affected not only by the choice of vaporizing flow model but also by the initial temperature of the solid matrix in contact with the flowing liquid. The magnitude of the parameter can either increase or decrease liquid front penetration rates.

It is also worthwhile to examine how the variation of the initial substrate temperature influences the cooling rate of the substrate as well as the vaporization rate of the liquid. Figure 4.17 shows the temperature and vaporization rate profiles at two different initial substrate temperatures of 116 K and 293 K, respectively. The profiles were examined at a depth of 190 mm. When the initial substrate temperature was specified as 293 K, the liquid reached the depth of 190 mm within approximately 2 seconds of injection and it required approximately 108 seconds for the wetted surface temperature of the substrate to approach the liquid boiling temperature of 112 K. The corresponding vaporization rate profiles indicate that the vaporization rate approaches zero

from the initial value of  $0.0548 \text{ kg/m}^2/\text{sec}$  after approximately 68.5 seconds. When an initial substrate temperature of 116 K was specified, the profiles of temperature showed a corresponding reduction in the time required for the substrate to approach LNG temperature.

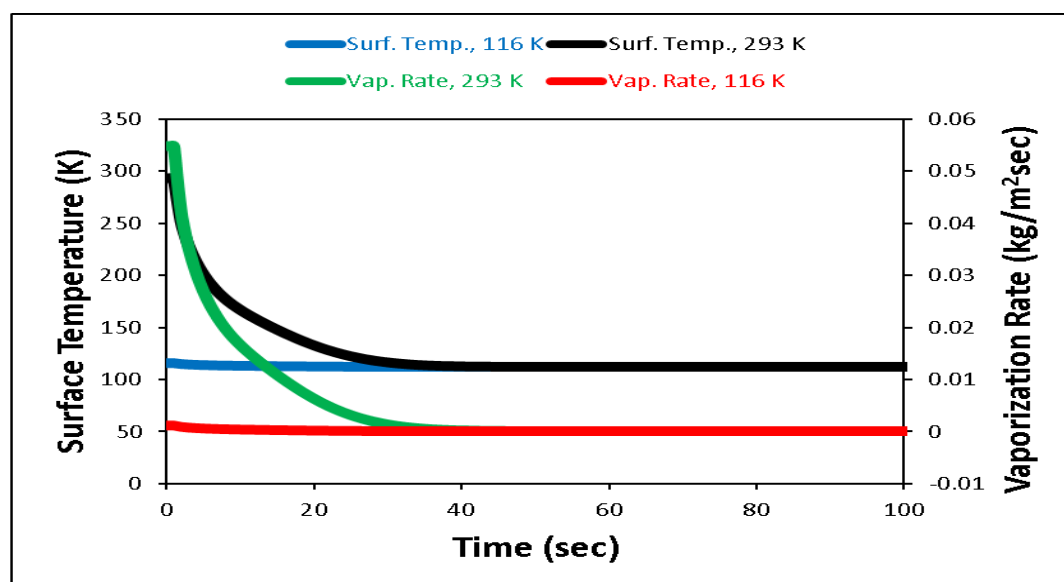


**Figure 4.17:** Surface temperature and vaporization rate profiles at two different initial porous material temperatures of 116 and 293 K, respectively.

We have also examined the influence of initial substrate temperature variations on vapour mass fraction and interface penetration. In Figure 4.18, we present the

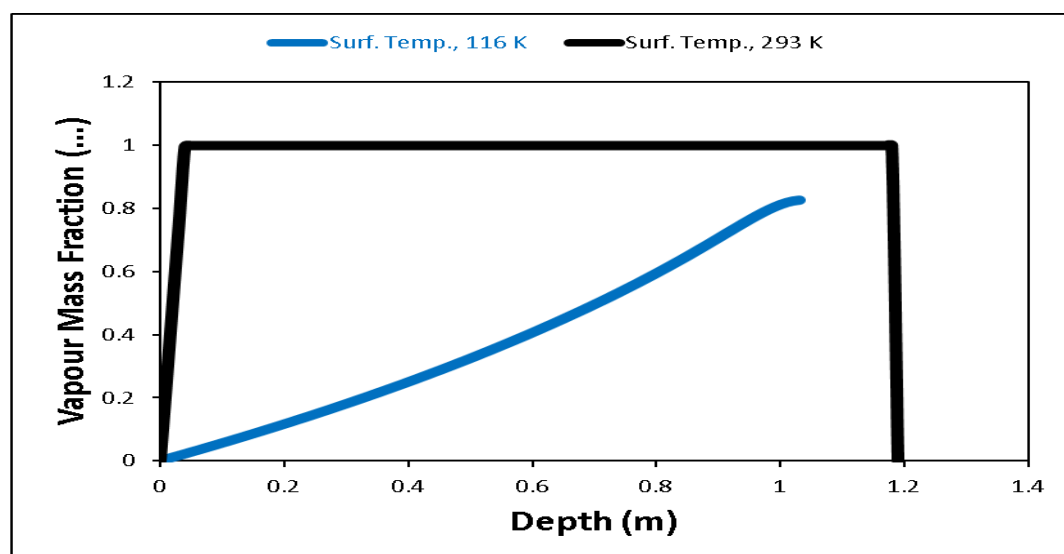


**Figure 4.18:** Vapour mass fraction profiles at two different initial porous material temperatures of 116 K and 293 K, respectively.

vapour mass fraction profiles at 116 K and 293 K, respectively. The interface position for the 293 K specification is shorter, at approximately 50 mm. The extent of Zone 2 is 1130 mm for this higher temperature case. An interface is not generated for the 116 K case rather a liquid (front) reaching a depth of 1030 mm is observed. At this penetration depth, about 83 percent of the injected liquid has vaporized. Let us consider the magnitude of the vapour mass fraction at the 50 mm depth based on the 116 K case, we find that just about 3 percent of the liquid has vaporised at this depth while 100 percent of the liquid vaporized at the same location for the 293 K case. Thus liquid vaporization and penetration rates strongly depend on the initial temperature of the porous layer through which the liquid is flowing.

#### 4.4.2 Influence of Injection Velocity

When the liquid is injected with a velocity of 0.15 m/s, Figure 4.19 suggests that approximately 108 seconds must elapse before the wetted surface temperature of the substrate can cool close to the boiling temperature of the liquid. The corresponding vaporisation rate approached zero at later times. Assuming that

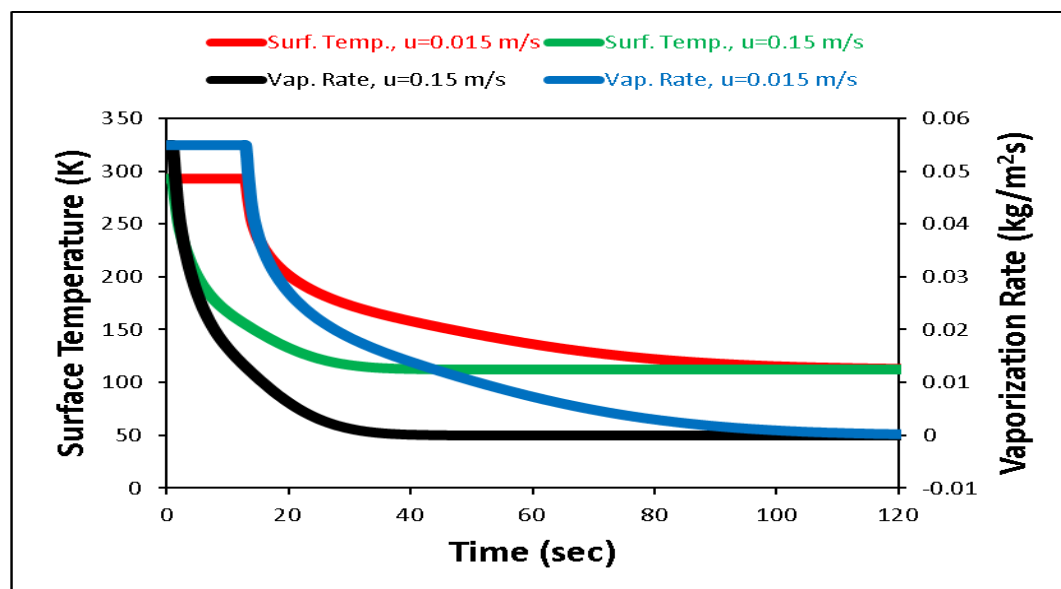


**Figure 4.19:** Surface temperature and vaporization rate profiles at two different injection velocities of 0.15 m/s and 0.015 m/s, respectively.

the flowing liquid contacted the solid matrix at the considered location of 190 mm, a different trend is observed when the liquid was injected with a relatively

lower velocity of 0.015 m/s. In this particular case, it took approximately 14 seconds for the liquid to contact the considered location unlike the shorter time of 2 seconds that was observed when the liquid was injected with a velocity of 0.15 m/s. Figure 4.19 also reveals that a relatively larger time must elapse before the wetted substrate can cool close to liquid temperature, based on the relatively lower velocity value. Figure 4.20 examines the influence of injection velocity in terms of vapour mass fraction and interface penetration. For the relatively higher injection velocity of 0.15 m/s, the interface is located at 50 mm with Zone 2 extending to a value of 1130 mm. The relatively lower injection velocity of 0.015 m/s results in the movement of the interface to a depth of 10 mm from the injection point. The extent of Zone 2 is found to be approximately 10 mm. The shorter interface at the relatively lower injection velocity is caused by higher vaporization rate. The shorter Zone 2 extent at this lower velocity can be related to the smaller area of contact for heat transfer. The reverse could be explained for the relatively higher injection velocity case.

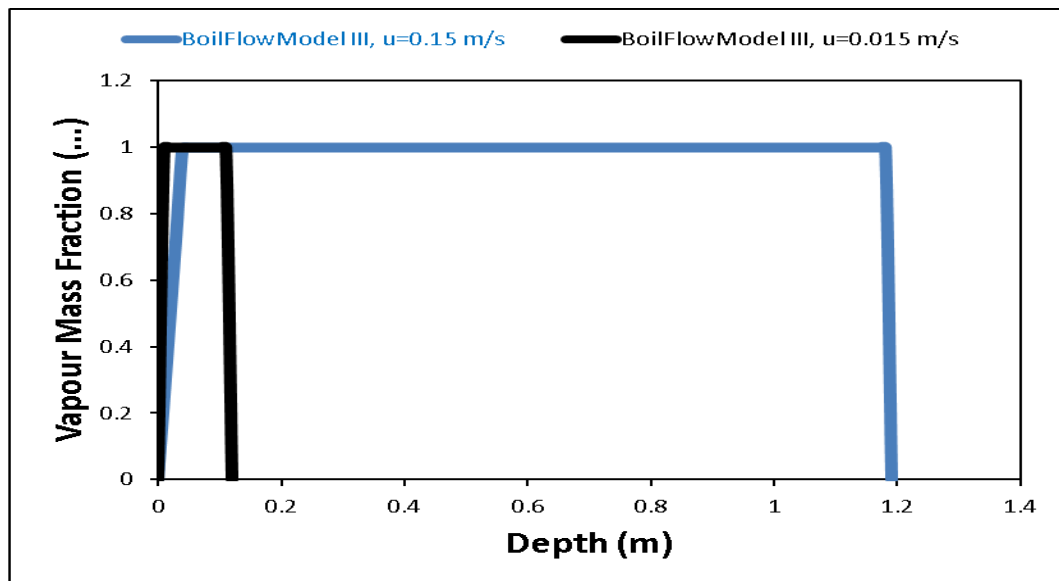


**Figure 4.20:** Vapour mass fraction profiles at two different injection velocities of 0.15 m/s and 0.015 m/s, respectively.

#### 4.4.3 Influence of Substrate Type

The cooling rate and corresponding vaporisation rate will depend on the type of porous materials (i.e. substrate type) through which the liquid is flowing. Figure 4.21 shows the temperature and vaporization rate profiles for soil and glass wool porous materials. The differences in the substrate cooling rate can be noticed

from the temperature values at the first liquid-solid contact time of approximately 2 sec for the soil and glass wool porous materials. While the soil substrate cooled towards 271.8 K at this time, the glass wool substrate cooled to a much lower temperature of 120.4 K. It is clear from these results that the glass wool cooling rate is extremely rapid compared to the cooling rate observed by soil. Figure 4.22 examines the trends in vapour mass fraction profiles using soil and glass wool porous materials. The maximum liquid penetration depth is shorter at

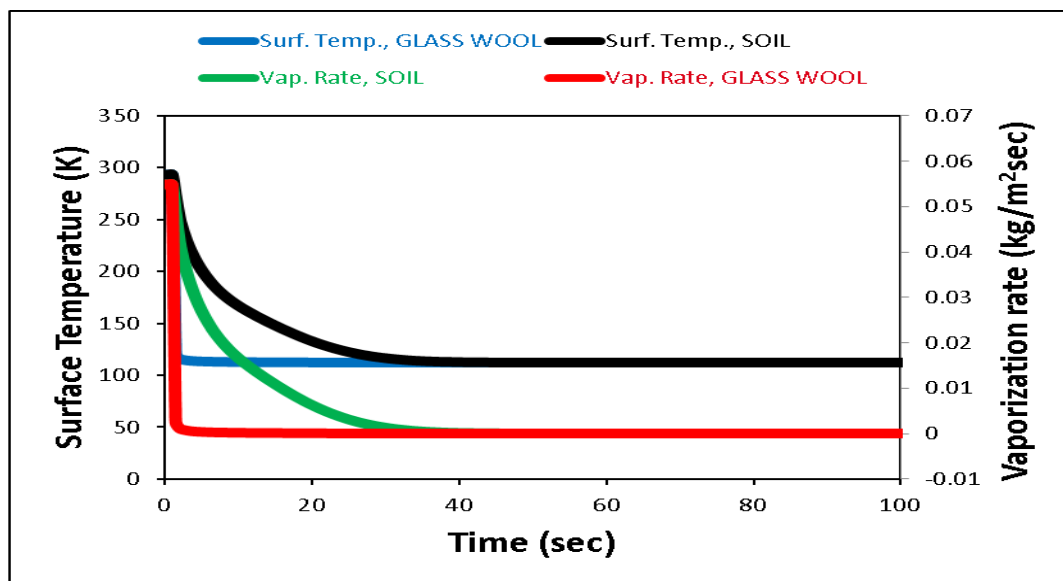


**Figure 4.21:** Surface temperature and vaporization rate profiles using two different types of substrates.

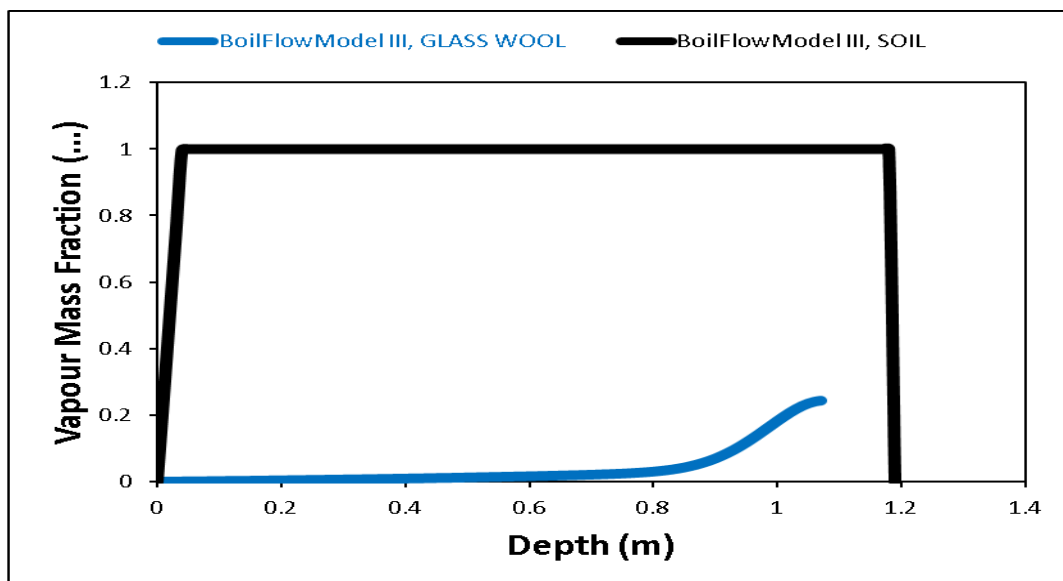


**Figure 4.22:** Vapour mass fraction profiles for two different types of substrates.

50 mm for soil because of its higher thermal conductivity (thermal conductivity effects is discussed in Section 4.4.4) which results in very fast vapour production rate. Thus, substituting glass wool for soil leads to a higher vaporization rate at the considered location. On the other hand, the glass wool porous material leads to liquid vaporization rates so small that no interface is generated at the considered time and location. The front position is at 1070 mm where it is seen that approximately 24 percent of the liquid has vaporized. This implies a tendency for deeper liquid penetration for glass wool substrates than soil substrates.

#### 4.4.4 Influence of Thermal Conductivity

We have discussed in Section 4.4.3 above that the substrate type influences the vaporizing flow behaviour. One parameter governing the amount of heat transported to the surface is the thermal conductivity of the substrate. Here, we tested the sensitivity of the vaporizing flow process by varying the magnitude of the thermal conductivity of the solid matrix. Figure 4.23 presents the profiles of temperature and vaporization rate at a depth of 190 mm. The time it took for the temperature of the porous material to decrease towards 112 K reduced from 108 seconds to 76.5 seconds when the substrate thermal conductivity was decreased from a value of 5.18 W/m/K to a value of 1.18 W/m/K. This trend can be explained

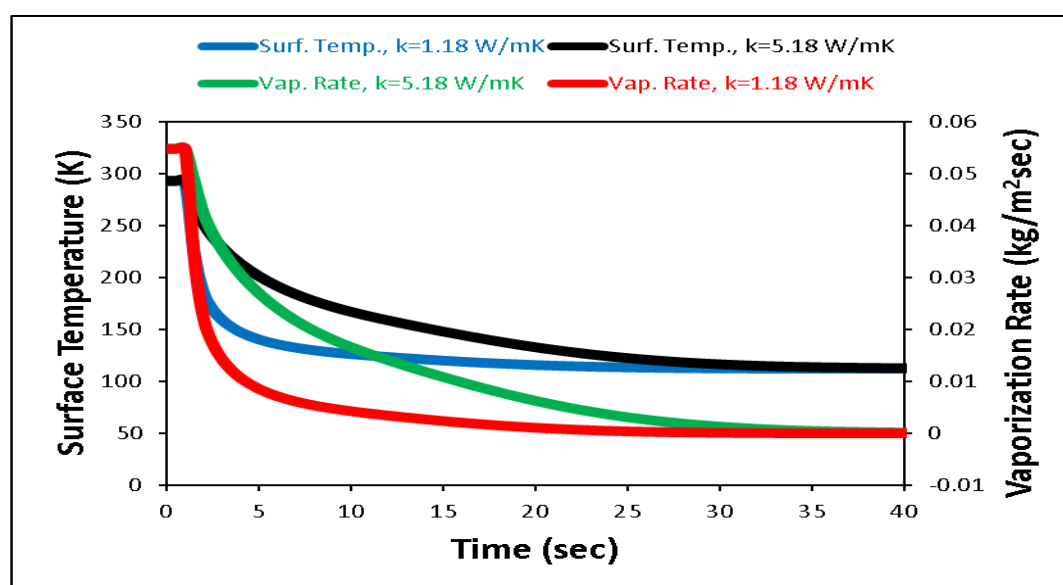


**Figure 4.23:** Surface temperature and vaporization rate profiles for two different thermal conductivities of the porous material.

from the understanding that a higher thermal conductivity of the substrate implies a lower cooling rate occasioned by the fast conduction of heat towards the surface of the solid matrix with attendant decrease in the cooling rate. The variation of thermal conductivity will also influence vapour mass fraction and interface penetration depth as Figure 4.24 would show. Liquid penetration is deeper with the relatively lower value of thermal conductivity because the strength of the vaporisation effect reduces with less vapour interfering with liquid penetration. The shorter extent of Zone 2 (101 mm) at the lower thermal conductivity value is a direct reflection of the low vapour production rate at considered locations when the magnitude of the thermal conductivity is lowered. A reverse explanation can be given for the longer extent of Zone 2 (1130 mm) at the higher thermal conductivity value.

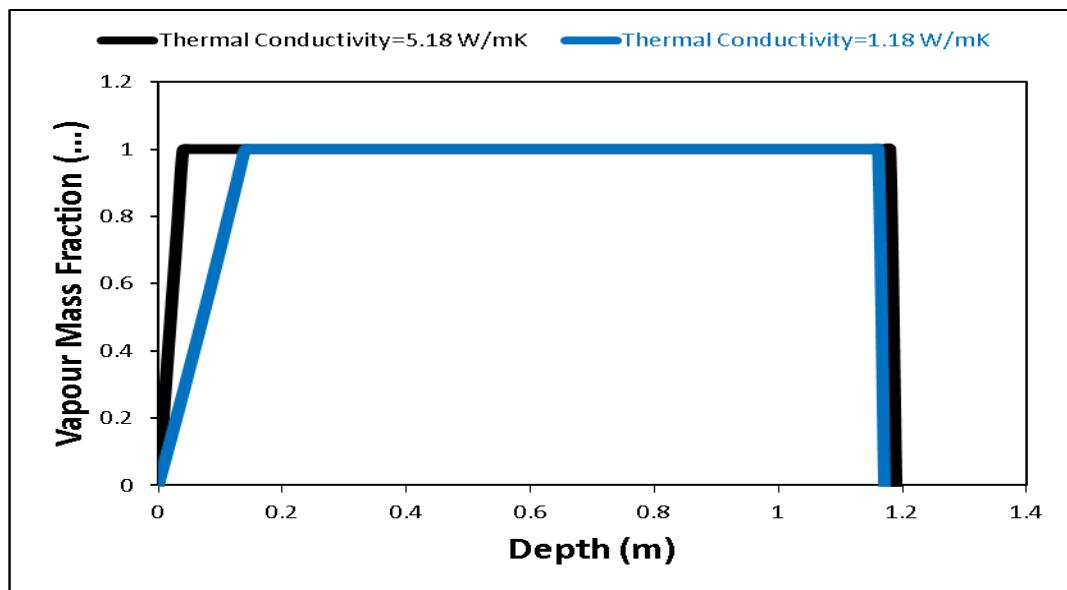


**Figure 4.24:** Vapour mass fraction profiles at two different thermal conductivities of the porous materials.

#### 4.4.5 Influence of Thermal Diffusivity

Specifying a thermal diffusivity of  $4.0 \times 10^{-6} \text{ m}^2/\text{sec}$ , Figure 4.25 shows that the wetted surface temperature will cool towards the liquid boiling temperature within 108 seconds following liquid-solid contact and that the vaporization rate will approach zero under 68.5 seconds. We observe a different cooling and vaporisation behaviour when a relatively higher value of thermal diffusivity ( $4.0 \times 10^{-5} \text{ m}^2/\text{sec}$ ) is specified. Here, the substrate cooled towards the boiling

temperature of the liquid within just 25 seconds of injection with the corresponding vaporization rate approaching zero under 19 seconds of liquid-solid contact. Figure 4.26 shows how penetrable the liquid will become when the

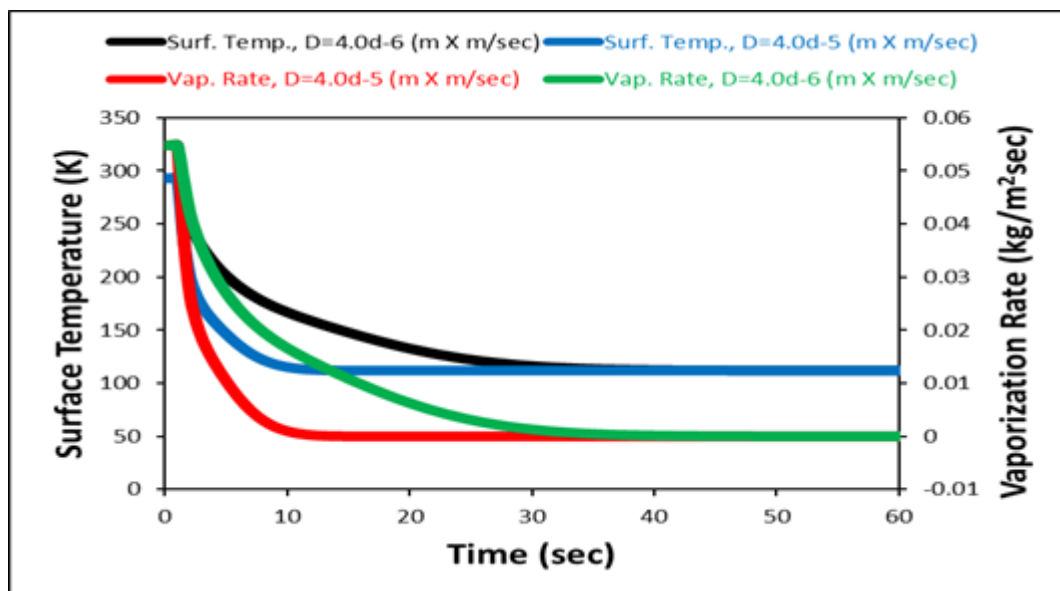


**Figure 4.25:** Surface temperature and vaporization rate profiles based on two different thermal diffusivities of the porous material.

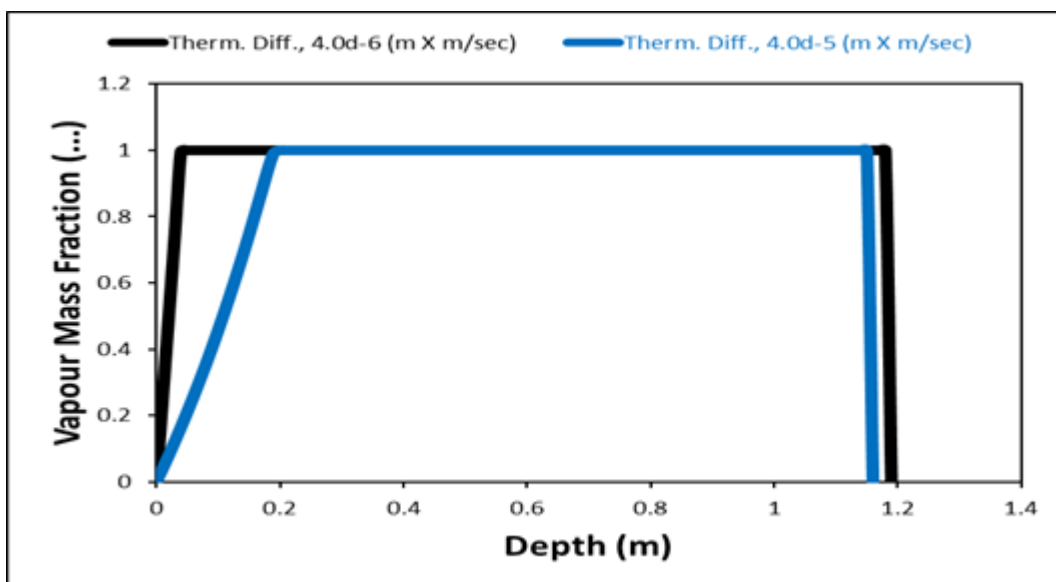


**Figure 4.26:** Vapour mass fraction profiles based on two different thermal diffusivity cases of  $4.0 \times 10^{-5} \text{ m}^2/\text{sec}$  and  $4.0 \times 10^{-6} \text{ m}^2/\text{sec}$ , respectively.

substrate thermal diffusivity is increased. Higher liquid penetration is observed when the higher thermal diffusivity of  $4.0 \times 10^{-5} \text{ m}^2/\text{sec}$  is specified. Again, at this

relatively higher value of thermal diffusivity, liquid will flow to the substrate much slower leading to a more rapid cooling of the substrate. As a result, a slower vaporisation and deeper penetration will ensue. The interfaces are located at 50 and 200 mm for the lower and higher thermal diffusivity values, respectively. The corresponding extent of Zone 2 at the lower and higher thermal diffusivities are equally affected and are at penetration depths of 1130 and 230 mm, respectively.

#### 4.4.6 Influence of Liquid Type

The generic nature of the BoilFlowModel III allows for the simulation of different types of liquids, both cryogenic and non-cryogenic. We illustrate in Figure 4.27

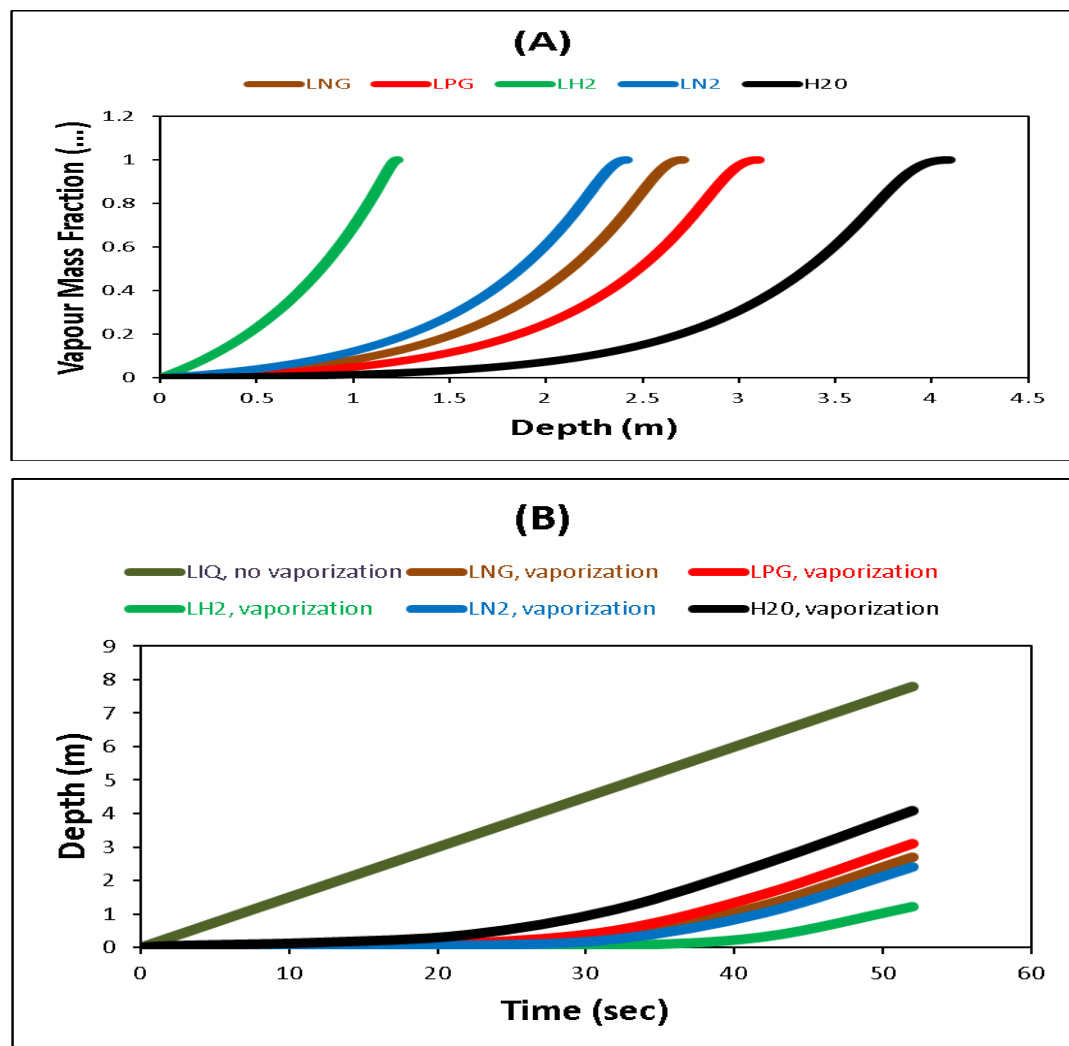


**Figure 4.27:** Effect of liquid type on boiling flow behaviour: (a) Vapour mass fraction profiles as a function of depth; (b) interface movement with time.

the trends observed from simulating the vaporizing flow of LNG, LPG (Liquefied Petroleum Gas),  $\text{LN}_2$  (Liquid Nitrogen),  $\text{LH}_2$  (Liquid Hydrogen) and water ( $\text{H}_2\text{O}$ ) through porous or penetrable media. Figure 4.27 (a) shows the profiles of vapour mass fraction as a function of depth for the five different liquid types. Considering a flow duration of 52 seconds, it is clearly evident that the interface penetration for water is the highest at a depth of 4.09 m. The maximum interface penetration for LPG is 3.1 m while LNG penetrated to a maximum depth of 2.71 m. The maximum interface positions for  $\text{LN}_2$  and  $\text{LH}_2$  are 2.42 m and 1.23 m, respectively. Thus, the penetration rates depends on the type of liquid injected, even though the presence of vapour reduced the impact of these different liquids on liquid penetration rates. Figure 4.27 (b) provides a better visualization of the movement of the interface for the different liquid types as time progresses. Here, the respective interface movement is compared with the front movement in the absence of vaporization. In the absence of vaporization, a front will reach a depth of 7.8 m (assuming the porous layer is extended to, say, 10 m) within the considered flow duration of 52 seconds.

## 4.5 Boiling Regime Effects

We have so far assumed that the vaporizing flow phenomena is limited by film boiling only. As already pointed out, this is not always the case since not all liquids will boil in the film boiling regime. The thermophysical properties of the substrate through which the liquid is flowing can cause the flowing liquid to boil under a different boiling regime or rather a combination of boiling regimes. In order to account for these possibilities, we have incorporated the effect of boiling regime or boiling heat transfer coefficient in the model. We first examined the case of three-regime boiling before investigating how a solely nucleate boiling process can influence the vaporizing flow behaviour.

### 4.5.1 Three-Regime Boiling

As already explained in Chapter 2, three-regime boiling describes a boiling phenomenon where a boiling liquid undergoes film, transition and nucleate boiling regimes, in this order. Also in Chapter 2, we presented a detailed analysis of the mechanism of three-regime boiling as well as identified various correlations that could be employed for the prediction of the different boiling

regimes. For the purposes of this work, the correlations of Klimenko (1981) for film boiling and those of Kalinin (Kalinin et al., 1975; Opschoor, 1980) for transition boiling were employed for the estimation of boiling heat transfer coefficients. The full details of both Klimenko's (Klimenko, 1981) and Kalinin's (Kalinin et al., 1975) correlations are presented in Section 2.0.3 of Chapter 2. The following paragraphs will examine the profiles of temperature, vaporization rate, vapour mass fraction and interface movement under three-regime boiling.

### **Temperature and Vaporization Rate Profiles**

Figure 4.28 (a) shows the temperature and vaporization rate profiles obtained for the case of a flowing liquid undergoing three-regime boiling by considering a liquid-solid contact depth of 190 mm from the injection point. A plot of the flow behaviour under film boiling is also added for comparative analysis. Figure 4.28 (a) indicates that, under three-regime boiling, the wetted substrate would cool to the boiling temperature of the liquid within just 8 seconds following liquid contact as against the higher cooling duration noted under film boiling. Though initially, trends in the cooling rates are similar until after approximately 4.5 seconds following which transition boiling regime is initiated. This causes the profile to deviate from the film boiling curve. Moreover, substrate cooling rates are more rapid during the transition boiling regime as the blue curve of Figure 4.28(a) would indicate. The wetted surface temperature at the onset of transition boiling is about 208 K with the vaporization rate having a value of 0.027 kg/m<sup>2</sup>/sec.

Figure 4.28 (b) shows the trend for the corresponding vaporization rates. The curves of film and three-regime boiling follow a similar trend during film boiling. After the onset of transition boiling which, again, initiates after approximately 4.5 seconds, a rapid rise in vaporisation rate is clearly evident. This is caused by more cooling of the solid matrix as more liquid comes in direct contact with the substrate, thus partially removing the heat transfer limitations caused by the earlier presence of the low thermal conductivity vapour film. After the onset of nucleate boiling, the rate of vaporization decreases with time. Between the onset of transition boiling and the onset of nucleate, the vaporization rate increased from 0.027 kg/m<sup>2</sup>/sec to 0.073 kg/m<sup>2</sup>/sec.

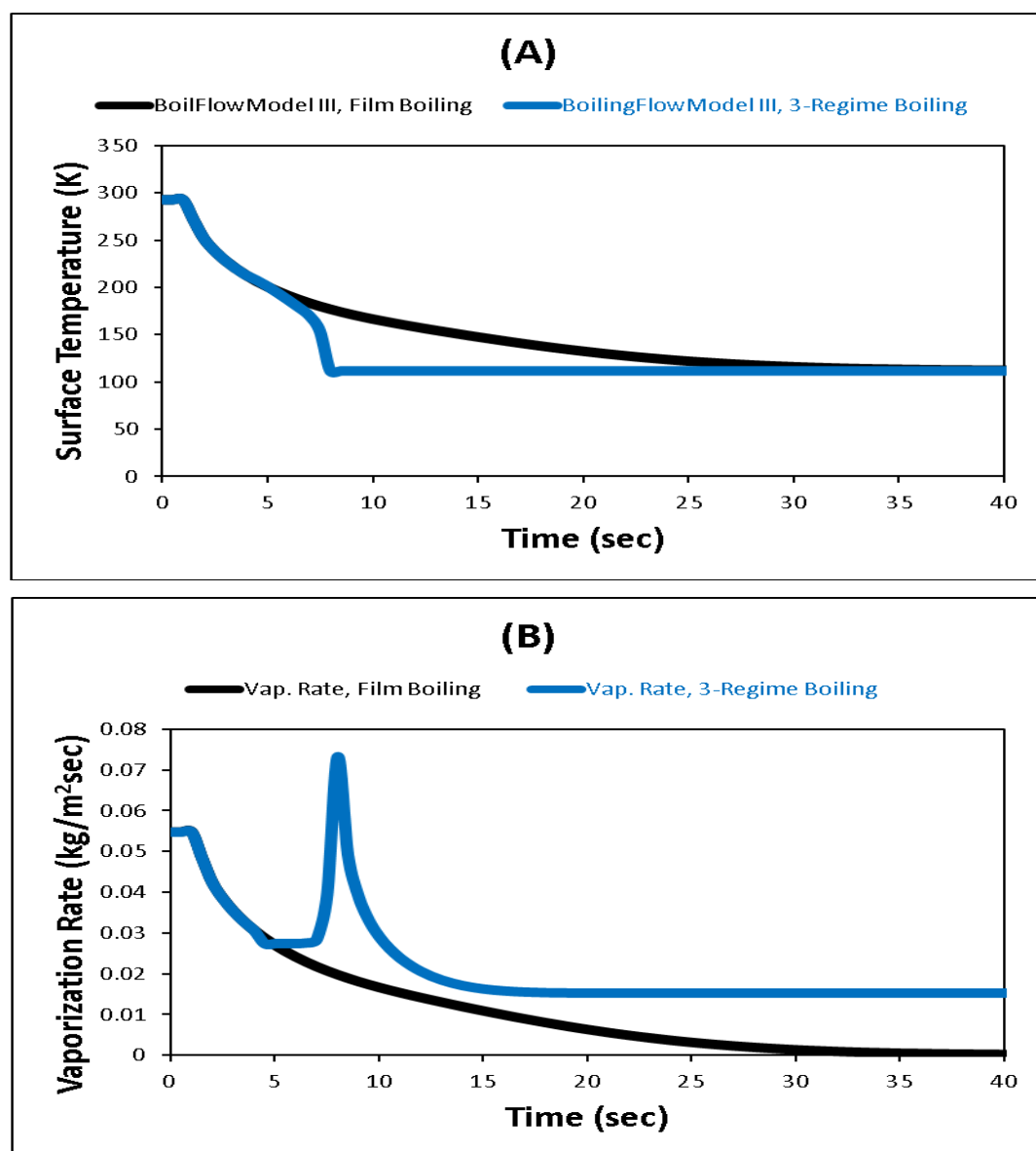


**Figure 4.28:** Temperature and vaporization rate profiles for film and three-regime boiling.

### Vapour Mass Fraction Profiles

We now examine how the influence of three-regime boiling reflects on vapour production and interface penetration rates. Comparing the vapour mass fraction profiles as a function of depth under film and three-regime boiling, Figure 4.29 reveals that the interface position under three-regime boiling is at 60 mm after 32 seconds. On the other hand, the interface position under film boiling is found to be at a depth of 600 mm which is an order of magnitude increase in the maximum liquid penetration depth. The reason for the decrease in penetration

depth under three-regime boiling can be attributed to the effects of higher heat transfer and vaporization rates experienced during transition and nucleate boiling regimes.

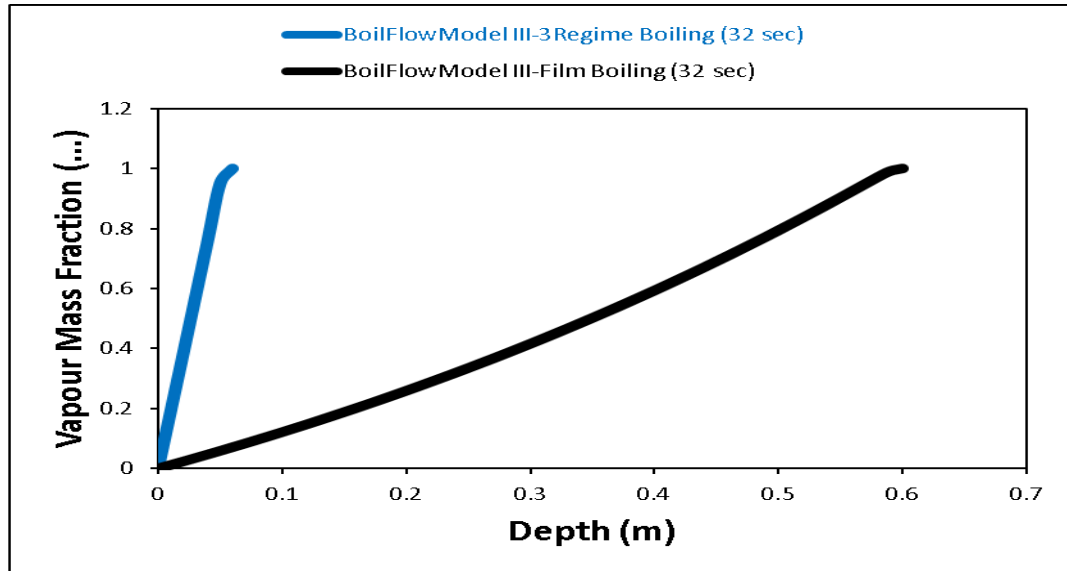


**Figure 4.29:** Vapour mass fraction profiles for vaporizing liquid flows undergoing film and three-regime boiling.

### Interface Movement and Velocity Profiles

We have plotted the interface movement and the corresponding flow velocities in Figure 4.30 (a) and 4.30 (b), respectively. Figure 4.30 (a) shows the movement of the interface under film and three regime boiling. We observe that the interface position under film boiling moved from 20 mm after 2 seconds to 600 mm after 32 seconds. The interface position for the three-regime boiling case moved from 20 mm at 2 seconds to 60 mm within the 32 seconds flow duration. Excluding the effect of vaporization on the flowing liquid, we would expect the liquid to reach a depth of 4.8 m. Figure 4.30 (b) shows the corresponding interface velocity profiles under film and three-regime boiling. Initially, the vaporization rate is so strong under both boiling regimes that the liquid flow velocity decreased from 0.15 m/s to 0.01 m/s within 2 seconds of injection. The flow velocity under film boiling regime ultimately increased with increasing time. In contrast, the flow velocity under three-regime boiling ultimately decreased with increasing time. Thus, more liquid penetration is expected under film boiling than under three-regime

boiling since the vaporization effect during film boiling is weaker than during three-regime boiling.

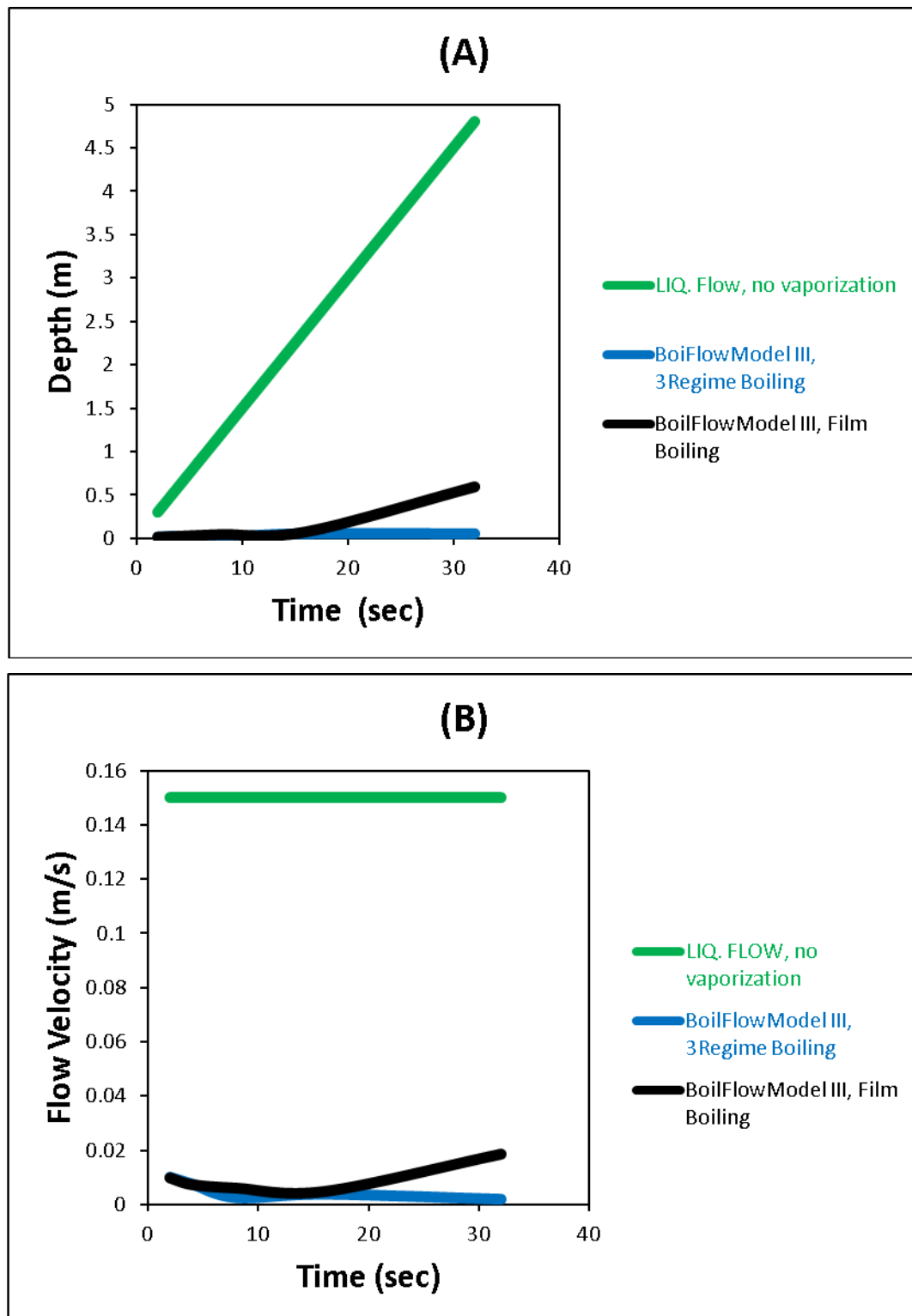


**Figure 4.30:** Interface position and flow velocity profiles for vaporizing flow under film and three-regime boiling.

### 4.5.2 Nucleate Boiling

This section investigates the influence of nucleate boiling regime on vaporizing flow behaviour and compares its results with those obtained for film and three-regime boiling. The establishment of nucleate boiling regime implies that the vapour films that are normally present under film and transition boiling regimes would completely collapse, allowing direct contact between the flowing liquid and the solid matrix. Since the heat transfer is no longer limited by the low thermal conductivity vapour, the temperature distribution and heat flux at the wetted substrate surface can be calculated by numerically solving a heat conduction problem as presented in Section 4.0.2. But this time, the first boundary conditions (Equations 4.7 (a) and 4.7 (b)) which characterizes the heat exchange at the liquid-solid interface is now replaced by Equations 4.12 (a) and 4.12 (b),

$$T = T_{s0} \quad \text{for } z > z(t) \quad (4.12a)$$

$$T = T_b \quad \text{for } z \leq z(t) \quad (4.12b)$$

where  $T_{s0}$ ,  $T_b$ , and  $z(t)$  are as defined in Section 4.0.2. Equation 4.12 (b) implies that, under nucleate boiling the substrate that is in direct contact with the liquid at a considered location is at the liquid boiling temperature. Nucleate boiling heat flux at the liquid-solid contact point is estimated using Equation (3.11) of Chapter 3.

The plot in Figure 4.31 depicts the temperature and vaporization rate profiles for a vaporizing liquid undergoing nucleate boiling. First, the liquid contacts the substrate after about 1.5 seconds, then rapidly cools the wetted surface to liquid boiling temperature. This rapid cooling rate is reflected in the corresponding rapid rise in the vaporization rate curve which ultimately reached a peak value of 0.268 kg/m<sup>2</sup>/sec. Thereafter, the vaporisation rate decreases with time at the considered location. In terms of vapour mass fraction, Figure 4.32 indicates that liquid penetration under nucleate boiling will increase with time. From the profiles of vapour mass fraction as a function depth, the first interface location after 4 seconds is at a depth of 20 mm from injection point. At 32 seconds, the interface moved deeper into the porous media to a new depth of 60 mm from

injection point. In Section 4.5.3, we compare the results from the various boiling regimes.

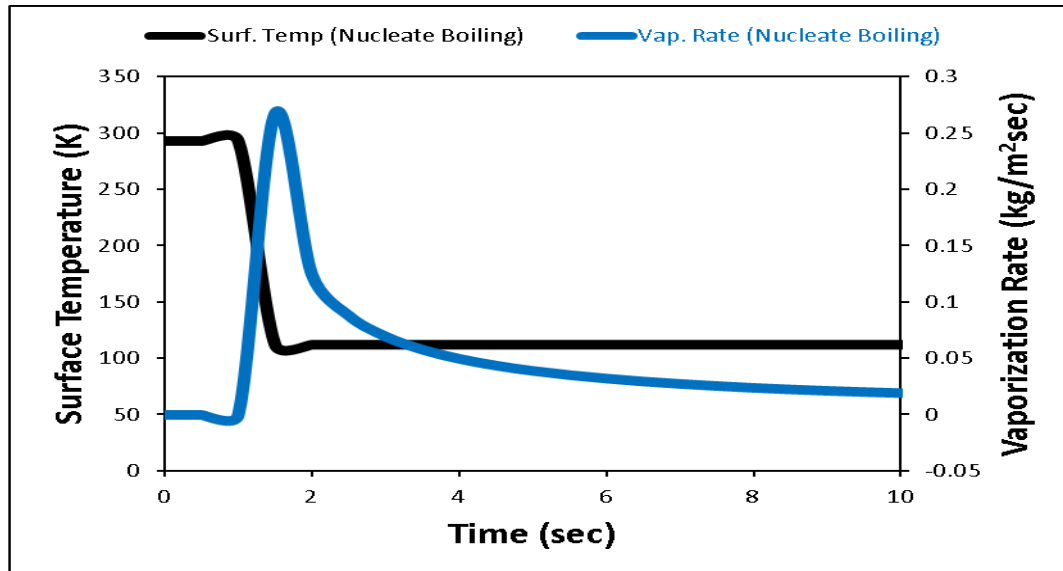


**Figure 4.31:** Temperature and vaporization rate profiles for a vaporizing liquid flow under nucleate boiling regime.

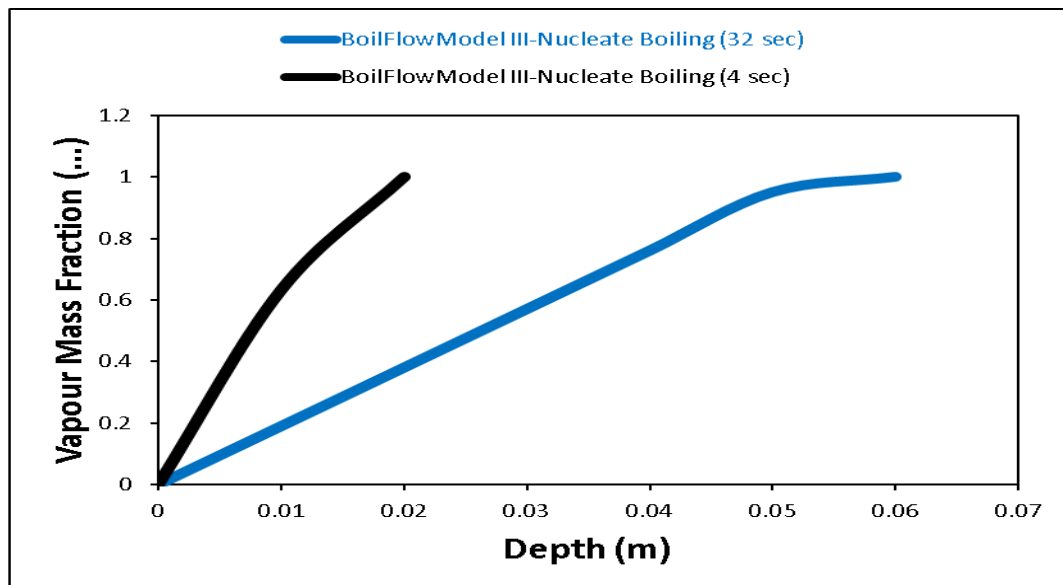


**Figure 4.32:** Vapour mass fraction profiles for a vaporizing liquid flow under nucleate boiling regime.

### 4.5.3 Comparison of the Different Boiling Regimes

An overall comparison of the vaporizing flow of liquids undergoing film boiling, three-regime boiling and nucleate boiling is presented in this section in order to give a better closure on the influence of the different boiling regimes on the

vaporizing flow process. The temperature profiles obtained from simulating a sample problem are presented in Figure 4.33. As earlier indicated, the wetted substrate surface under film boiling approached liquid temperature within 108 seconds following liquid-solid contact. For the three-regime boiling case, the cooling of the substrate to the liquid boiling temperature required approximately 8 seconds while the cooling rate under nucleate boiling is extremely rapid and was completed almost at the time of liquid-solid contact. The vaporization rate profiles for the different boiling regimes are also shown in Figure 4.34. The maximum vaporization rate under film boiling is  $0.055 \text{ kg/m}^2/\text{sec}$ . Under three-regime boiling, the maximum vaporization rate increased from  $0.055 \text{ kg/m}^2/\text{sec}$  to  $0.073 \text{ kg/m}^2/\text{sec}$ . The vaporisation rate under nucleate boiling is, at early times, the highest of the three with a value of  $0.268 \text{ kg/m}^2/\text{sec}$  which can be explained

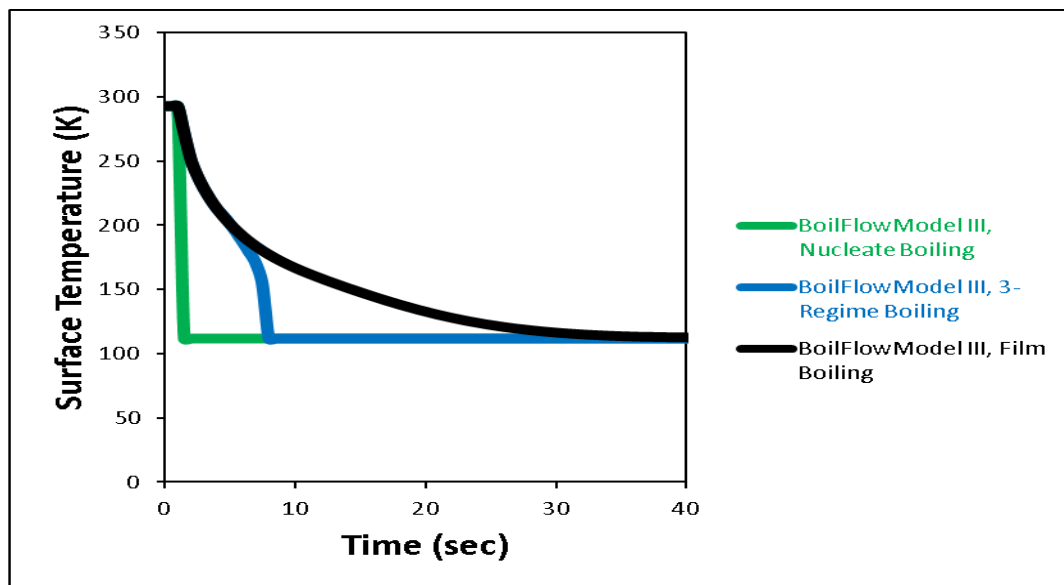


**Figure 4.33:** Temperature profiles under film boiling, three-regime boiling and nucleate boiling.

from the fact the limiting vapour is absent during nucleate boiling, thus allowing a larger release of heat. The result also show that the surface of the solid matrix is less rapidly cooled during film boiling and most rapidly cooled during nucleate boiling. The three-regime boiling case also showed very rapid cooling after the boiling regime has moved away from film boiling regime. The high vaporization rate observed at the onset of nucleate boiling can be attributed to the very small

time interval used at the start of the simulation which resulted in the extremely large initial vaporization rate.

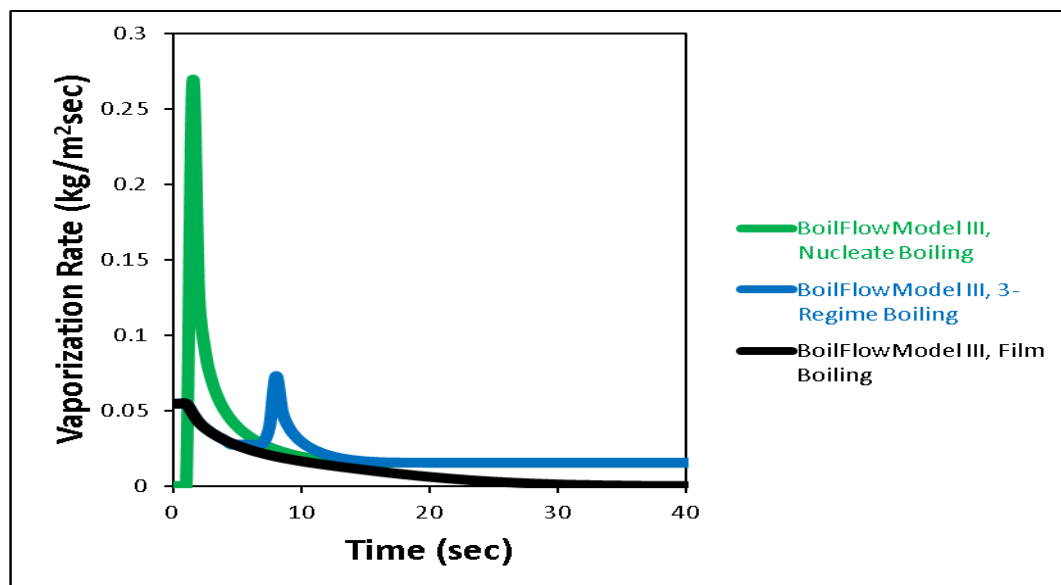


**Figure 4.34:** Vaporization rate profiles for film, three-regime and nucleate boiling regime.

Figure 4.35 views the results in terms of interface penetration depth. The interface moved from 20 mm to 600 mm under film boiling within a 32 seconds flow duration. The three-regime boiling flow case revealed a shorter interface penetration depth of 60 mm from injection point. The same magnitude of interface penetration depth is observed under nucleate boiling flow regime (i.e. 60 mm). We might relate the similarity between nucleate and three-regime boiling cases by first understanding that the last regime of a three-regime boiling flow is the nucleate boiling and the equivalence observed in interface penetration depth suggests that the closeness of the two curves can be related to the “nucleate boiling effect” under three-regime boiling.

Figure 4.36 is the corresponding interface velocity profiles for a vaporizing flow undergoing the three identified boiling regimes. While interface flow velocity under film boiling ultimately increased, that of three-regime boiling ultimately decreased. Under nucleate boiling, a relatively steady value of interface velocity is maintained between 2 seconds and 8 seconds. Thereafter, a decrease in the interface flow velocity is observed and this trend closely follows that of the three-

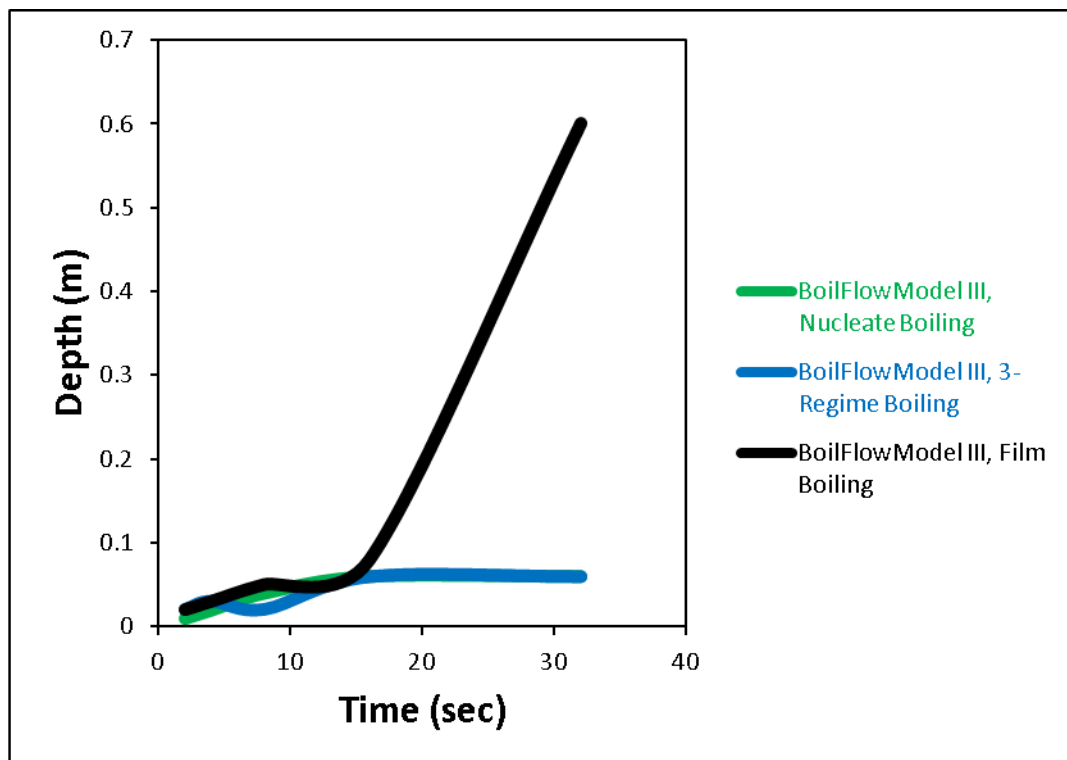


**Figure 4.35:** Interface position for a vaporizing flow under film boiling, three-regime and nucleate boiling regimes.

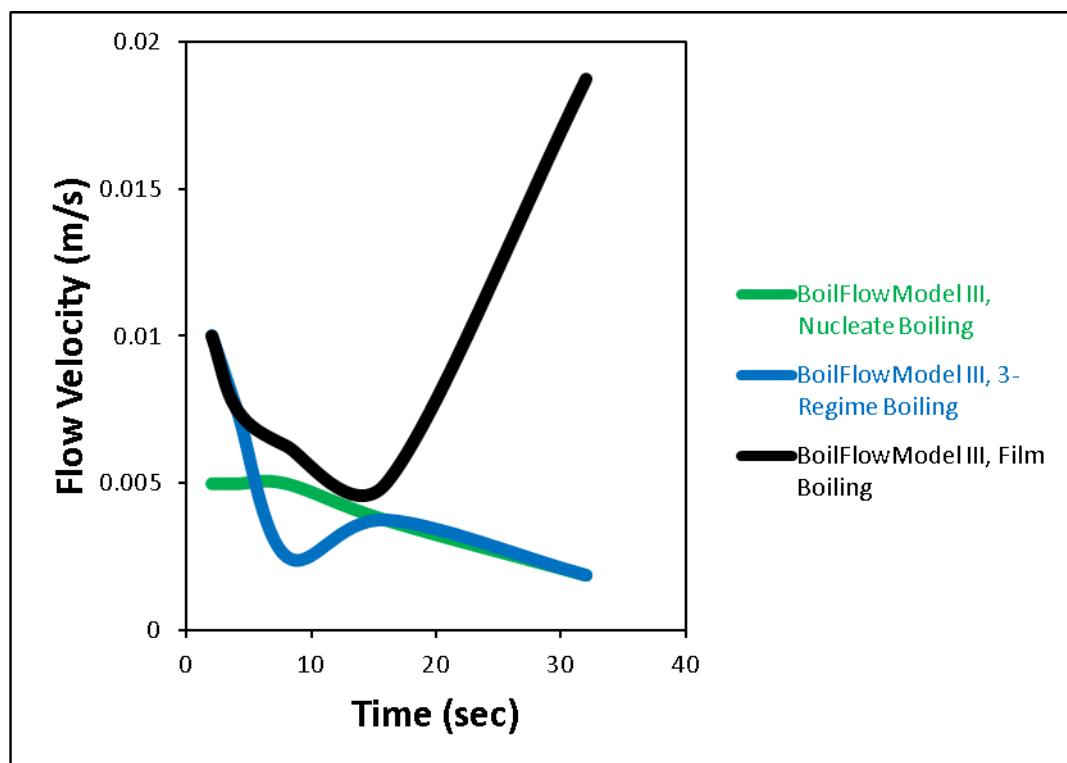


**Figure 4.36:** Interface velocity profiles for film, three-regime and nucleate boiling.

regime boiling case. What these results show is that the rate of liquid vaporization and the rate of penetration is affected by the type of boiling regime occurring at the interface between the flowing liquid and the solid matrix. Vaporization rate is relatively low during film boiling and relatively high during nucleate and three-regime boiling. Correspondingly, liquid penetration rates under film boiling is relatively high with the reverse being the case for either nucleate or three-regime boiling.

## 4.6 Discussion and Conclusions

A general model for predicting the vaporizing flow of liquids through porous or penetrable media has been extended to account for the effect of time-dependent heat conduction at the interface between the flowing liquid and the solid matrix. A finite difference solution coded in FORTRAN enables a prediction of the vapour mass fraction as a function of time and depth as well as the tracking of a propagating front and/or interface inside a porous layer. The position of the front and/or interface inside the porous layer signposts the depth which the injected liquid has reached. The extended model has been successfully verified using a commercial finite-volume based software (TOUGH2). Several sample simulation cases were examined in order to demonstrate the models' potential and generic features. The model was further extended to account for the influence of boiling regime on the vaporizing flow behaviour having understood that, for real systems, incorporating these effects are important. The following conclusions are drawn from the results presented in this chapter:

- 1) For a vaporizing flow through porous media, the effect of a decreasing one-dimensional unsteady-state heat flux at the interface between the flowing liquid and the solid matrix results in a decrease in the vaporization rate and vapour mass fraction as well as a corresponding increase in the maximum liquid penetration depths.
- 2) The rate of cooling of the wetted substrate surface depends on the type of boundary condition applied at the heat exchange interfaces. Two limiting cases of heat conduction are applied and used to examine the vaporizing

flow of liquids through porous media. A comparative study of the three vaporizing flow models reveals that the selection of the most realistic of the three (i.e. BoilFlowModel III) is appropriate.

- 3) Simulation results from the selected vaporizing flow model was compared to results of simulation runs obtained from a commercial finite volume simulator (TOUGH2) and it was found that the predictions by the BoilFlowModel III agreed well with TOUGH2 predictions for water in film boiling regime , both qualitatively and quantitatively. It was also found that the BoilFlowModel III has advantages over the TOUGH2 model in how it (i.e. BoilFlowModel III) models the temperature distribution in the solid matrix
- 4) Results from sample simulation problems reveal that liquids injected into porous media evolves with a time-dependent propagating front and/or interface whose location determines the depth of liquid penetration into the porous media. In addition to the presence of a two-phase liquid-vapour zone (Zone 1), a single-phase vapour zone (Zone 2) and an air-saturated zone (Zone 3), a fourth single-phase liquid zone (Zone 4) can also developed as soon as the heat flux at a considered liquid-solid contact point decreases to zero.
- 5) A number of input parameters are varied in order to analyse how they will influence the amount of liquid vaporized (vapour mass fraction) at a considered location and the maximum depth at which the liquid penetrates under a given flow duration. Results indicate that: (i) as initial substrate temperature decreases, vapour mass fraction decreases and liquid penetration increases because the amount of heat released by the substrate decreases, thus reducing vaporization rate and its attendant effects on liquid flow; and (b) as injection velocity decreases, vapour mass fraction increases and liquid penetration rate decreases since liquid-solid contact time increases, allowing more time for heat transfer and vaporization.

- 6) A sensitivity study on thermal conductivity and thermal diffusivity makes it clear that as (a) thermal conductivity decreases, vapour mass fraction decreases and liquid penetration rate increases because the conductive heat flux moves to the wetted surface less rapidly, thus increasing the cooling rate and decreasing the heat flux needed for liquid vaporization and (b) as thermal diffusivity increases, vapour mass fraction decreases and penetration rate increases because the conductive heat transfer is faster resulting in a faster decrease in heat flux and vaporization rate.
- 7) Parametric effects on substrate and liquid type are investigated and it was found that replacing, for example, a soil substrate with a glass wool substrate will significantly reduce the vapour mass fraction and increase the liquid penetration rates. It was also demonstrated that injecting five different liquid types with varied thermophysical properties, namely LNG, LPG, LH<sub>2</sub>, LN<sub>2</sub> and H<sub>2</sub>O leads to the liquids reaching different depths under a considered flow duration. The results shows that H<sub>2</sub>O would be, for example, more penetrable into soil than would LH<sub>2</sub>.
- 8) The effect of a liquid flowing through porous or penetrable media and vaporizing under nucleate or three-regime boiling results in an increase in vapour mass fraction and a decrease in liquid penetration rates. In contrast, observations made from the results obtained for a film-boiling regime case reveal that a relatively lower vapour mass fraction profiles and a deeper liquid (interface) penetration rates would result. Hence, film boiling underpredicts the vapour mass fraction and overpredicts the liquid penetration rates. A more realistic vaporizing flow model should incorporate the effects of boiling regime.
- 9) In this chapter, we applied a one-dimensional unsteady-state heat conduction equation with various boundary conditions in order to obtain solutions for substrate temperatures, heat flux into the flowing liquid, liquid vaporization and penetration rates. We found that heat conduction and liquid vaporization affect liquid penetration into the porous medium. We assumed in the respective models that heat transfer into the liquid is

strictly horizontal, thus ignoring vertical heat conduction effects. In Chapter 5, we investigate vertical heat conduction effects through the implementation of a two-dimensional unsteady-state heat conduction model and examine its influence on the vaporizing flow behaviour.

# 5

## ***Liquid Flow with Two-Dimensional (2-D) Unsteady-State Heat Conduction***

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In Chapter 4, the BoilFlowModel III model was chosen and successfully used to study the vaporizing flow of liquids through porous or penetrable media. The influence of the boiling regime on the vaporizing flow behaviour was also investigated in Chapter 4. The BoilFlowModel III (hereafter referred to as the 1-D Model) formulation describes a model of liquid flow with one-dimensional unsteady-state heat conduction from the confining solid matrix. The influence of vertical heat transfer within the solid matrix was neglected. In this chapter, the effect of vertical heat transfer was included through the extension of the 1-D model to a 2-D model. The 2-D model simply describes a model of liquid flow with two-dimensional unsteady-state heat conduction from the confining solid matrix. Furthermore, the 1-D model and the 2-D model are used to simulate some sample problems and their respective solutions are compared in order to understand how differently they would predict vaporizing fluid flow processes under the same flow conditions. This chapter was split into three sections. In Section 5.0, the mathematical formulation and solution of the 2-D model was presented. In Section 5.1, we presented a comparative study of the 1-D model and the 2-D model based on sample simulation cases involving vaporizing flow of LNG through soil and glass wool porous materials. In the final section (Section 5.2), the discussion and conclusions drawn from comparing the 1-D and the 2-D solutions are presented.

## 5.0 Two-Dimensional Heat Conduction Model

From the presentations in Chapters 3 and 4, it is clear that a description of the vaporizing flow of liquids through porous or penetrable media can be achieved by a coupling between an advection equation and a one-dimensional heat conduction equation. This section describes the formulation of a two-dimensional heat conduction equation that was coupled to an advection equation to form a 2-D model of vaporizing fluid flow through porous media. Figure 5.1 is a schematic diagram describing the 2-D model. The porous layer consists of two regions – Region A and Region B as shown in Figure 5.1. Region A indicates the vertical extent of a two-phase liquid-vapour zone (the so-called Zone 1) or a single phase liquid zone (the so-called Zone 4). Region B is the zone comprising of both a single phase vapour zone (the so-called Zone 2) and an air-saturated zone (the so-called Zone 3). More details on the various regions (or zones) are provided in Section 3.11 and Section 4.3.3 of Chapter 3 and 4, respectively. The 2-D problem geometry shows the liquid entering the porous medium from the top and then flowing downwards between the bounding solid matrices. The front or interface describes the leading edge of the liquid as it moves downwards through the porous layer. Heat conduction is assumed to occur within the matrix in both horizontal ( $x$ ) and vertical ( $y$ ) directions. The heat flux into the flowing LNG would, as we might expect, lead to the vaporization of the cryogen.

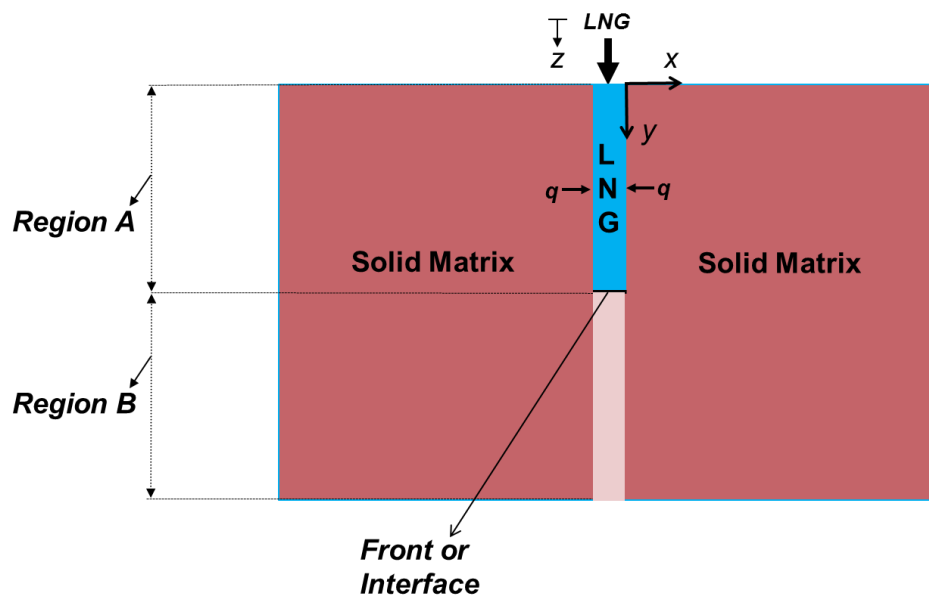


**Figure 5.1:** A schematic diagram of the 2-D model showing the injection of LNG into a porous layer with two-dimensional heat transfer from the solid matrices into the liquid.

### 5.0.1 Formulation of the Problem

In Figure 5.2, we considered just the right half of the problem geometry shown in Figure 5.1, due to symmetry. The maximum lateral (horizontal) distance of the solid matrix is  $X$ , and the maximum vertical distance of the solid matrix is  $Z$ . Once introduced into the porous layer, the cryogen then flows downwards in the  $z$ -direction as shown in Figures 5.1 and 5.2. The nodal network used for the 2-D analysis and based on the 2-D slab of Figure 5.2 is shown in Figure 5.3. We have arbitrarily designated 11 planes in Figure 5.3 in order to help in the easier visualization of the two-dimensional (2-D) nodal network. In the nodal network, the temperature within the 2-D slab has the bottom left corner as the origin of the  $x$ - and  $y$ -coordinate system (rather than the top left corner as indicated in Figures 5.1 and 5.2). The horizontal  $x$ -coordinate goes from  $x=0$  to  $x=X$  in the solid matrix while the vertical  $y$ -coordinate goes from  $y=0$  to  $y=Z$  in the solid matrix.

The aim is to calculate the heat transfer rates into the liquid. To achieve this, we calculated the temperature distribution in the solid matrix including the temperatures in all nodal points in contact with the liquid. From Figure 5.3, the nodal points  $T_{0,0}$  through  $T_{10,0}$  are expected to experience a decreasing temperature upon liquid-solid contact. It may be worthwhile to point out that the quantity of heat that can be released from a given horizontal plane within the solid matrix, say  $v$ , to vaporize the liquid at a time,  $t$ , can simply be calculated from the knowledge of the rate of change in the heat content,  $Q_v$  and the heat conduction from the neighbouring planes at time,  $t$ .

For the purpose of this work, we analysed the heat losses using a finite difference approach (Smith, 1985). We assumed that the solid matrix is isotropic and at a specified initial (time  $t=0$ ) temperature,  $T_0$ . Then, for time  $t>0$ , the liquid (whose temperature,  $T_b$  is assumed constant) contacts the solid matrix with accompanying transfer of heat from the solid matrix to the flowing liquid. We also assumed that there is a heat transfer coefficient and the magnitude of this coefficient is dependent on the type of boiling regime occurring at the interface between the flowing liquid and the solid matrix.

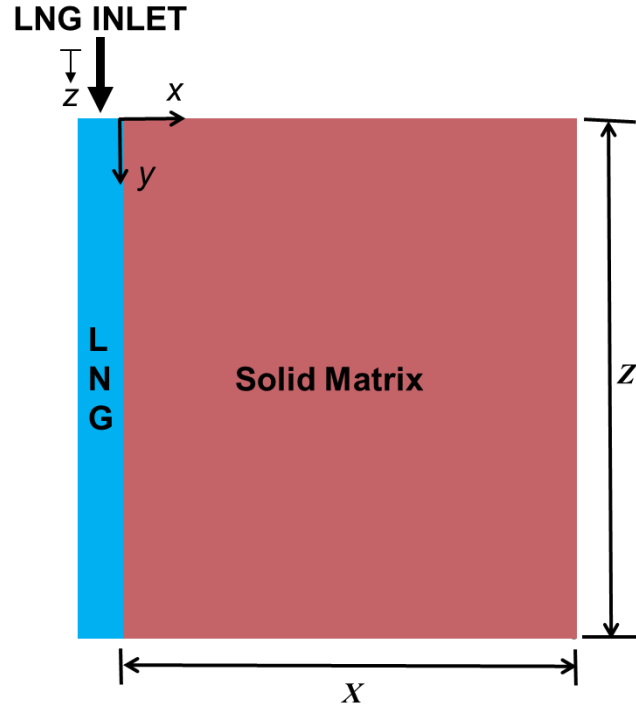


**Figure 5.2:** A schematic diagram showing the right half of Figure 5.1, which is the computational domain upon considering the left half of the problem geometry as the axis of symmetry.

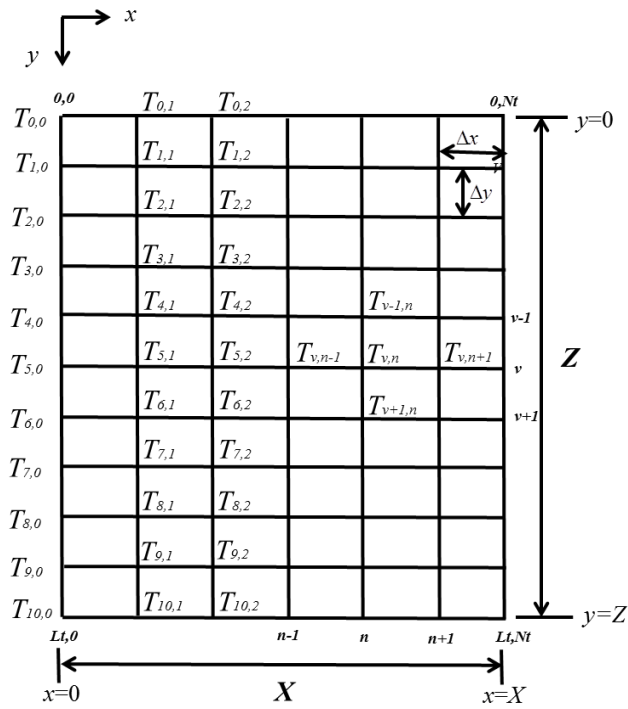


**Figure 5.3:** A schematic diagram of the nodal network for the two-dimensional (2-D) model. The nodal network is constructed based on Figure 5.2.

Having already indicated above that there is no internal heat sources and that the material thermal properties are uniform and independent of temperature, the two-dimensional heat conduction in rectangular coordinates is given by the Fourier's Law of heat conduction (Ozisik, 1993; Carslaw and Jaeger, 1959b),

$$\frac{\partial T}{\partial t} = \alpha_s \left( \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right) \quad (5.1)$$

The two-dimensional temperature distribution function  $T(x,y,t)$  within the solid matrix is obtained when Equation (5.1) is solved subject to the following initial and boundary conditions:

$$(a) \quad T(x,y,0) = T_0 \quad \dots T_0 \text{ is the initial temperature.} \quad (5.2)$$

$$(b) \quad T(0,y,t) = T_0 \quad \dots \text{for } z > z(t), (0 \leq v \leq Lt) \quad (5.3)$$

$$(c) \quad \left. \frac{\partial T}{\partial x} \right|_{x=0} = \frac{h}{k_s} (T - T_b) \quad \dots \text{for } z \leq z(t), (0 \leq v \leq Lt) \quad (5.4)$$

$$(d) \quad \left. \frac{\partial T}{\partial x} \right|_{x=X} = 0 \quad \dots \text{for all } t, (0 \leq v \leq Lt) \quad (5.5)$$

$$(e) \quad \left. \frac{\partial T}{\partial y} \right|_{y=0} = 0 \quad \dots \text{for all } t, (1 \leq n \leq Nt-1) \quad (5.6)$$

$$(f) \quad \left. \frac{\partial T}{\partial y} \right|_{y=Z} = 0 \quad \dots \text{for all } t, (1 \leq n \leq Nt-1) \quad (5.7)$$

where  $z(t)$  is the time-dependent penetration depth of the liquid front and/or interface,  $z$  is a considered location down the porous layer,  $h$  is the heat transfer coefficient,  $k_s$  is the thermal conductivity of the substrate,  $Z$  is the vertical extent of the solid matrix,  $X$  is the horizontal extent of the solid matrix,  $v$  is the vertical space counter,  $n$  is the horizontal space counter,  $Nt$  is the number of horizontal nodes,  $Lt$  is the number of vertical nodes, and  $T_b$  is the boiling temperature of the liquid. Equation 5.3 with its inequality condition means that

the temperature at the liquid-solid interface remains at its initial value so long as the liquid front,  $z(t)$ , is yet to reach a considered location,  $z$ .

### 5.0.2 Numerical Solution Procedure

A finite difference (Smith, 1985) computer program written in FORTRAN was developed and used to solve the partial differential equations presented in Section 5.0.1. Specifically, the forward-time centred-space (FTCS) explicit finite difference method was used to solve for the temperature distribution in the solid matrix. The terms in Equation 5.1 are discretized along both  $x$ -direction and  $y$ -direction as shown in Figure 5.3 using the following finite difference approximations:

$$\frac{\partial^2 T_{v,n}}{\partial x^2} = \frac{T_{v,n-1}^{n1} - 2T_{v,n}^{n1} + T_{v,n+1}^{n1}}{(\Delta x)^2} \quad (5.8)$$

$$\frac{\partial^2 T_{v,n}}{\partial y^2} = \frac{T_{v-1,n}^{n1} - 2T_{v,n}^{n1} + T_{v+1,n}^{n1}}{(\Delta y)^2} \quad (5.9)$$

$$\frac{\partial T_{v,n}}{\partial t} = \frac{T_{v,n}^{n1+1} - T_{v,n}^{n1}}{\Delta t} \quad (5.10)$$

Again, the subscript,  $v$ , and the subscript,  $n$ , are the space counters in vertical ( $y$ ) and horizontal ( $x$ ) coordinates, respectively. By replacing the terms in Equation 5.1 with their corresponding finite difference approximations (Equations 5.8 through 5.10), and at a certain grid point, say  $(v,n)$ , Equation 5.11 can easily be inferred:

$$\frac{T_{v,n}^{n1+1} - T_{v,n}^{n1}}{\Delta t} = \alpha_s \left( \frac{T_{v,n-1}^{n1} - 2T_{v,n}^{n1} + T_{v,n+1}^{n1}}{(\Delta x)^2} + \frac{T_{v-1,n}^{n1} - 2T_{v,n}^{n1} + T_{v+1,n}^{n1}}{(\Delta y)^2} \right) \quad (5.11)$$

Rearranging Equation 5.11 gives the following expression for the determination of the temperature distribution in the solid matrix,

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$$T_{v,n}^{n1+1} = T_{v,n}^{n1} + C_x (T_{v,n-1}^{n1} - 2T_{v,n}^{n1} + T_{v,n+1}^{n1}) + C_y (T_{v-1,n}^{n1} - 2T_{v,n}^{n1} + T_{v+1,n}^{n1}) \quad (5.12)$$

where  $C_x$  and  $C_y$  are given by,

$$C_x = \frac{\alpha_s \Delta t}{(\Delta x)^2} \quad (5.13a)$$

$$C_y = \frac{\alpha_s \Delta t}{(\Delta y)^2} \quad (5.13b)$$

Equation 5.12 is the explicit finite difference equation that solved for the interior (or internal) grid points (or nodes).  $T_{v,n}^{n1+1}$  is determined at the time level  $n1+1$  from the knowledge of grid point temperatures at the previous time level  $n1$ . The stability criterion associated with Equation 5.12 is given by (Ozisik, 1993),

$$(C_x + C_y) \leq 0.5 \quad (5.14)$$

Once  $T_{v,n}^{n1+1}$  was solved, the temperatures at the interface between the flowing liquid and the solid matrix,  $T_{v,0}$ , for  $v=0$  to  $Lt$  were calculated by first discretizing the convective boundary condition (Equation 5.4) using a second-order accurate discretization scheme (Ozisik, 1993). Obviously, the use of this convection boundary condition (Equation 5.4) is only applicable during film and transition boiling. In the discretization of Equation (5.4), we first applied a fictitious node,  $(v,-1)$ , at a horizontal distance,  $\Delta x$ , from the liquid-solid boundary nodes and assigned a fictitious temperature,  $T_{v,-1}$ . By applying a similar fictitious node approach discussed in Section 4.0.1 of Chapter 4 and evaluating Equation 5.12 for  $n=0$ , the finite difference equations for the boundary nodes,  $T_{v,0}^{n1+1}$  becomes,

$$T_{v,0}^{n1+1} = \left( 1 - 2C_x - \left( \frac{2h\Delta x C_x}{k_s} \right) \right) T_{v,0}^{n1} + (2C_x) T_{v,1}^{n1} + \left( \frac{2h\Delta x C_x}{k_s} \right) T_b + C_y (T_{v-1}^{n1} - 2T_{v,0}^{n1} + T_{v+1,0}^{n1}) \quad (5.15)$$

Furthermore, the right boundary nodes was determined by invoking a Dirichlet boundary condition which had the initial temperature of the solid matrix prescribed at these nodes. Similarly, the initial solid matrix temperature was prescribed for the top boundary nodes (for all columns 1 to  $Nt-1$ ) and the bottom boundary nodes (for all columns 1 to  $Nt-1$ ), respectively. For nucleate boiling, the determination of temperatures at the liquid-solid boundary was based on the application of a non-derivative boundary condition which is essentially a prescription of the boiling temperature of the flowing liquid at these boundary nodes.

Thus, with the knowledge of the temperatures at the liquid-solid boundary nodes, the boiling temperature of the liquid and the heat transfer coefficient at the liquid-solid boundary, the heat flux,  $q$ , from the solid matrix which resulted in the vaporization of the flowing liquid was calculated from Equation 3.10 of Chapter 3, during film and transition boiling, or from Equation 3.11 of Chapter 3, during nucleate boiling. Ultimately, by use of a rearranged finite difference approximation of Equation 3.7 of Chapter 3 (i.e. Equation 3.19), the magnitude of the vapour mass fraction,  $\varphi_i^{k+1}$  as a function of time,  $t$ , and depth,  $z$ , was predicted from Equations 5.16a and 5.16b,

$$\varphi_i^{k+1} = \varphi_i^k - \frac{u\Delta t}{\Delta z}(\varphi_i^k - \varphi_{i-1}^k) + S\Delta t \quad (5.16a)$$

$$S = \frac{qA_{s,specific}}{\rho L} \quad (5.16b)$$

where all variables and parameters in Equations 5.16a and 5.16b are as defined previously. Successful convergence tests were carried out on the 2-D model based on the stability and convergence criteria explained for the 1-D case discussed in Sections 3.8 and 4.1 of Chapters 3 and 4, respectively. The stability of the finite difference method was checked using Equation 3.22 of Chapter 3 and Equation 5.14 of the present chapter. The horizontal grid was chosen by dividing the total horizontal lengths of the 2-D slab by  $Nt$ . Likewise, the vertical grid was chosen

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by dividing the total vertical length of the 2-D slab into  $Lt$  nodes. The final 2-D grid was chosen for the numerical simulation by first selecting  $\Delta t = 0.0001$  sec,  $Lt = 5$  and  $Nt = 5$ . Further numerical simulations were performed with successively finer grid sizes (or larger number of  $Nt$  and  $Lt$  nodes) and their respective convergence checked, based on the assumption that the solution obtained using the finest grid is equivalent to the analytical solution. We ensured that as the grid is refined, the discretization error approaches zero. Based on the conditions and simulation cases considered in this section (i.e. Section 5.0), it was found that the chosen final grids ( $Nt = 51$ ,  $Lt = 51$ ) and a selected time-step of the order of 0.001 sec used in the FORTRAN code is sufficient for all simulation results presented in Section 5.1 of this chapter. The necessary cpu time for the entire model ranged between 1 min to 1 hr on a Pentium 2.10 GHz workstation.

## 5.1 Simulation Results

The 1-D and 2-D solutions compared in this chapter predict the vaporizing flow of liquids through two different types of porous materials. In both simulations, the liquid is assumed to enter the porous layer from the top with the magnitude of the injection velocity specified as 0.45 m/s. The problem geometry and dimensions considered are similar to those presented in Chapter 4. The chosen dimensions are representative of some of the typical dimensions found in analogous systems where the models can be applied.

The substrates investigated are soil and glass wool while the liquid considered is LNG. The thermophysical properties of the respective substrates and liquids are listed in Table 4.1-4.4 of Chapter 4. For sample problems involving full boiling heat transfer coefficients (i.e. three-regime boiling), a constant heat transfer coefficient obtained using Klimenko's correlation (Klimenko, 1981) and a time-varying heat transfer coefficient obtained using the correlations of Kalinin et al. (1975) were used. Equation (2.11) of Chapter 2 is the heat transfer coefficient based on Klimenko's correlation (Klimenko, 1981) while Equation (2.10) is the heat transfer coefficient based on the correlations of Kalinin et al. (1975). Model outputs for the present analysis include temperature distributions of the solid

matrix, vertical and lateral temperature profiles, local lateral heat flux, front/interface movement and front/interface velocity.

### **5.1.1 Simulation Case #1: LNG Flow through Soil**

This sample simulation case involving LNG flow through soil includes the use of both a constant and a time-varying heat transfer coefficient. The initial temperature of the substrate is 293 K and the boiling temperature of LNG is 112 K. The LNG immediately vaporizes upon entry into the porous layer with the rate of vaporization dependent on the type of boiling regime occurring at the interface between the flowing liquid and the solid matrix.

#### Soil Temperature Distributions

The 2-D colour-shaded temperature distributions for cooling times of 5, 20 and 39 seconds are shown in Figures 5.4 (a-c) for soil. The temperature contours clearly indicate that the regions closest to the wetted surface boundary show the sharpest decrease in temperature. It is also obvious from the three plots that, upon prolonged liquid-solid contact, the cooling process penetrated deeper into the solid matrix with regions near the liquid-solid boundary eventually getting cooled close to LNG temperature. The top and bottom rows of Figures (a-c), which are, at a constant temperature,  $T_0$ , act as heat sources.

Examining Figure 5.4 (a) more closely, we observe that, at 5 sec, the cooling has reached a lateral distance of approximately 0.025 m. Figures 5.2 (b) and 5.2 (c) also show that at the later times of 20 and 39 sec, respectively, the lateral distance reached by the cooled region within the solid matrix further increased, suggesting that the lateral region (or distance) that can be cooled by the flowing liquid increases with time so long as the solid matrix is continuously wetted by the colder liquid. Although vertical variation of temperature at a given lateral distance was not easily noticeable at early times, say 5 sec, the variation of temperature in the vertical direction became more noticeable as time progressed to, say 39 sec (see Figure 5.2(c)). The vertical variability that can be observed at 39 sec may be attributable to the 2-D effect produced by the top and bottom boundaries of the solid matrix. These top and bottom boundaries at,  $n=1$  to  $Nt$ , are constant because a constant temperature boundary condition is prescribed at

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these boundaries, with each boundary producing the 2-D effect and acting as a heat source.

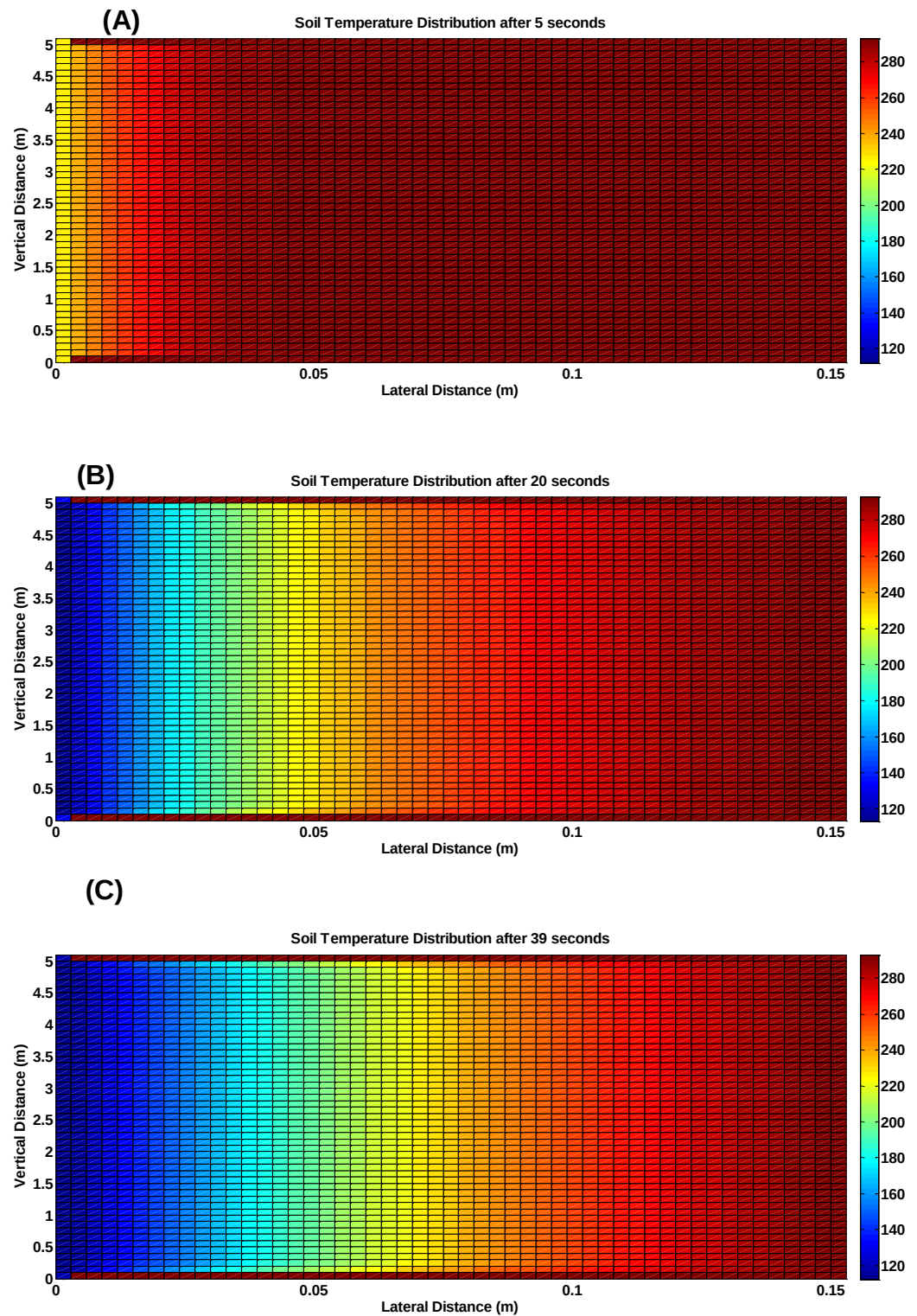
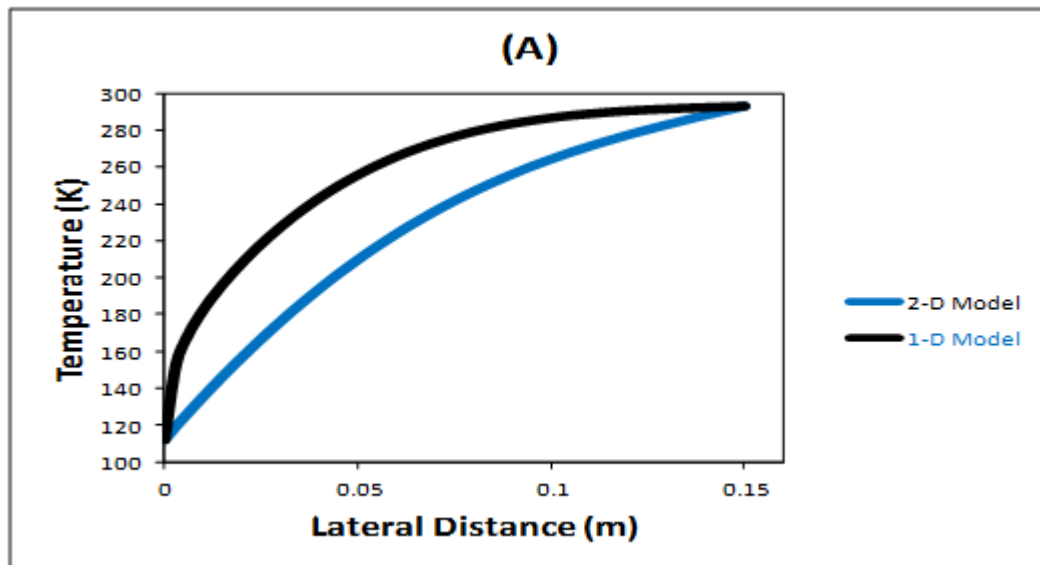


**Figure 5.4:** Temperature distribution in the soil substrate: (a) 5 seconds, (b) 20 seconds; and (c) 39 seconds.

### Lateral and Vertical Temperature Profiles

The lateral temperature profiles predicted by the 1-D and 2-D solutions after 39 sec based on considering the layer below the constant temperature boundary layer (see Figure 5.4 (c)) is shown in Figure 5.5 (a). Clearly, the two models gave differing results for lateral temperatures at the considered vertical location and at the considered time. It was found that, after 39 sec, the boiling process at the LNG boundary has already shifted to the nucleate boiling regime for both the 1-D and 2-D solutions. We also found that, in the 1-D solution, nucleate boiling occurred at relatively shorter duration than in the 2-D solution. This suggests that the amount of heat released by the solid matrix based on the 1-D model would be relatively smaller than the 2-D model after 39 sec . This ultimately results in the 1-D solution producing results that suggests a less rapid cooling of the solid matrix relative to the 2-D solution which had the onset of nucleate boiling occurring at a much longer time and which had the vertical heat conduction effects also accounted for. The deviation plot shown in Figures 5.5 (b) indicates that the lateral temperature profiles obtained based on the 1-D and 2-D models deviated by about 35 %. At first, the deviation profiles rapidly increased to the maximum 35 % value nearer the liquid-solid boundary with increasingly smaller deviation observed at increasing lateral distance into the solid matrix.



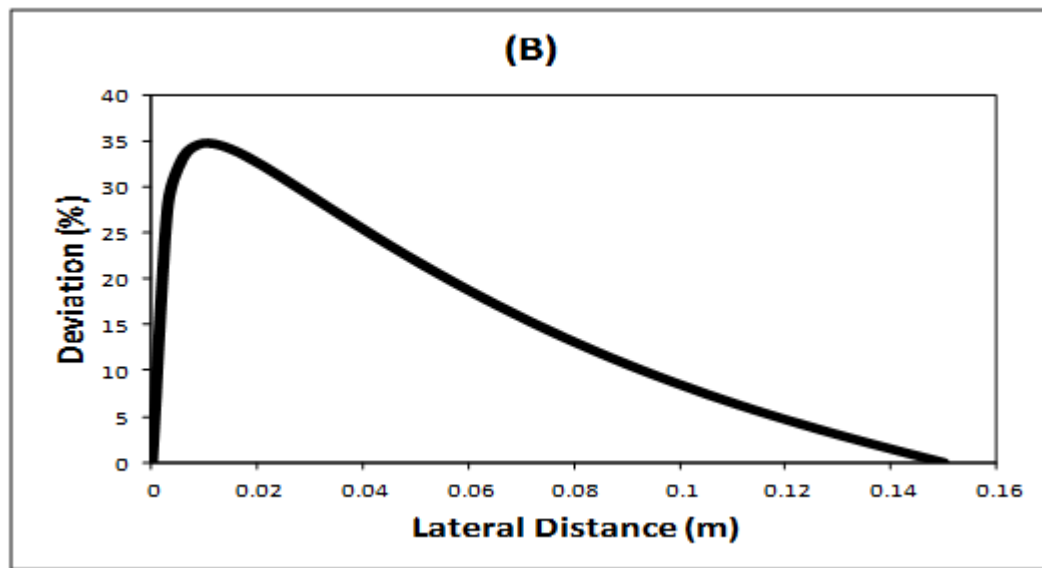


**Figure 5.5:** Comparison of the lateral temperature profiles of the one-dimensional (1-D) finite difference solution with those of the two-dimensional (2-D) finite difference solution at 39 sec: (a) Soil lateral temperature profiles; (b) percentage deviation based on the considered soil lateral temperature profiles.

A comparison of the cooling rates of the solid matrix at  $x=0$  and at location,  $z=200$  mm, based on the 1-D and 2-D solutions is shown in Figure 5.6. The cooling rate of the soil at the considered location based on the 2-D solution was found to be relatively slower than that of the 1-D solution. Initially, the difference in the two solution is small which may be attributed to film boiling, where direct contact of the flowing cryogen with the soil is not possible since a stable vapour film separating the liquid and solid phases is in place. In such a situation, the heat loss is mainly controlled by the superheat of the substrate, rather than substrate thermal properties. We can discern from Figure 5.6 that after approximately 8 sec (which also signifies the onset of transition boiling), the difference between the 1-D and 2-D solutions increases as vertical heat conduction effects becomes increasingly significant. For the 2-D solution, the duration of the transition boiling regime is found to be longer than the 1-D solution. We can attribute this longer transition boiling duration to the larger increase in the amount of heat reaching the surface as well as an increase in the amount of produced vapour. The relatively large increase in vapour production during transition boiling leads to the observed decrease in cooling rate profiles obtained from the 2-D solution. Furthermore, the difference between the 1-D and 2-D solutions in terms of the respective times for the onset of nucleate boiling regime can be observed from

Figure 5.6. For the 2-D cooling process, the onset of nucleate boiling occurred at a longer time of 28 sec whereas the onset of nucleate boiling for the 1-D solution occurred after 14 sec. We can attribute this difference to the prolonged transition boiling (i.e. longer time allowed for the heat exchange process during transition boiling) duration observed for the 2-D simulation case.

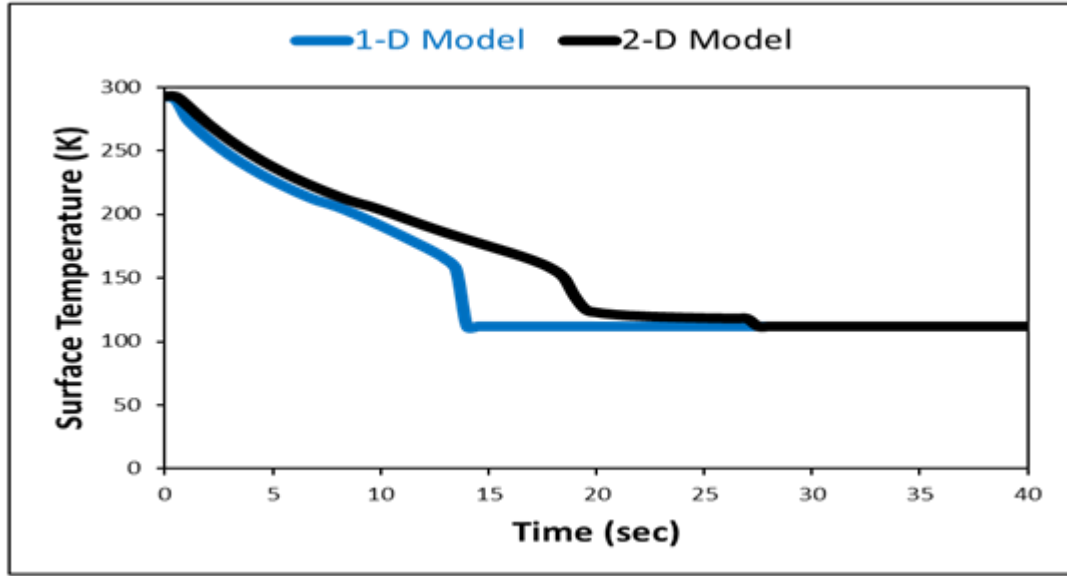


**Figure 5.6:** Comparison of temperature profiles at the LNG-soil boundary (at  $x=0$ ) obtained from the 1-D and 2-D solutions.

The 1-D and 2-D solutions can also be compared in terms of their respective vertical temperature profiles at specified lateral distances into the solid matrix. For the 1-D solution at 3 mm, it is seen from Figure 5.7 (a) that the temperature profile decreases rapidly from 293 K to 153 K after 15 sec. The corresponding 2-D solution decreased from 293 K to 177 K. These results show that the 1-D and 2-D temperature profiles are similar at first until after approximately 8 sec when they began to produce unidentical temperature profiles, thereby showing that the two models are different. A comparison of the temperature profiles at the lateral distance of 42 mm (not shown) suggested a relatively smaller difference between the 1-D and 2-D solutions. This finding indicates that a longer time would be required for the temperature field at the lateral distance of 42 mm to be perturbed. The deviation of the temperature profiles of the 1-D and 2-D models for lateral distances of 3 and 42 mm are shown in Figure 5.7 (b), where it is again evident that the difference between the 1-D and 2-D solutions increases more in

regions of the slab that are nearer the liquid-solid interface (see black curve of Figure 5.7 (b)). These lateral temperature profile and deviation plots suggest that the 1-D and 2-D solutions will start to differ after a characteristic time that is mainly dependent on the thermophysical properties (including boiling regime at the liquid-solid boundary) of the substrate and that the difference between the two solutions, for the considered duration, would decrease with increasing lateral distance into the solid matrix.

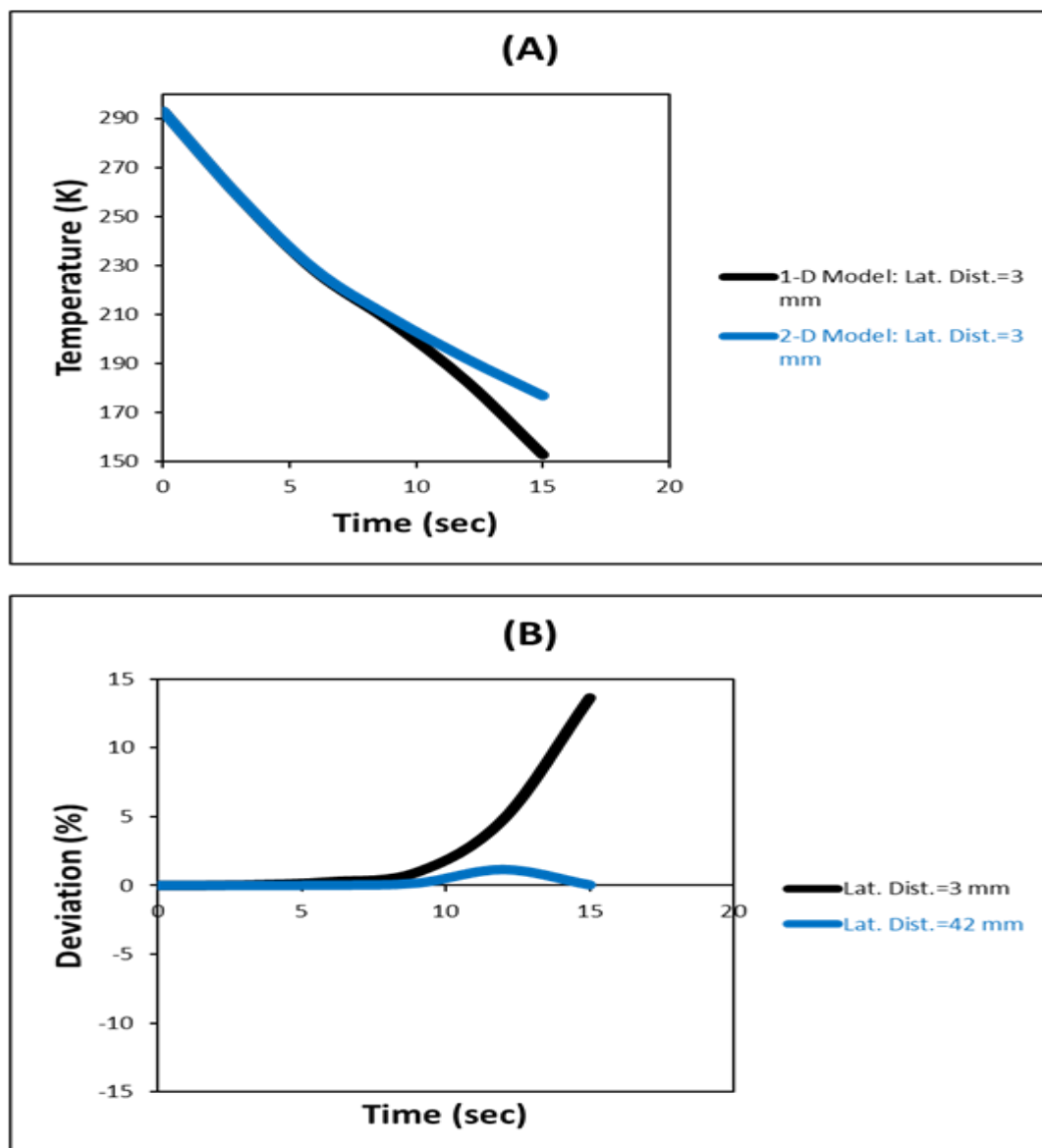


**Figure 5.7:** Comparison of the temperature profiles of the 1-D solution with those of the 2-D solution at lateral distances of 3 mm and 42 mm, respectively: (a) Soil temperature profile at a lateral distance of 3 mm; (b) Deviation between the 1-D and 2-D solutions at lateral distances of 3 and 42 mm.

To gain further insight into the differences between the two-models, the local lateral heat flux at a lateral distance coinciding with the nodal points ( $v$ ), for the 1-D model, and ( $v,n$ ) for the 2-D model, are compared. From Figure 5.3 above, it can be deduced that the local lateral heat flux at nodal point ( $v,n$ ) from plane  $v+1$  to plane  $v$  can be expressed as,

$$q_{v+1,v(n)} = -k_s \frac{T_{v+1,n} - T_{v,n}}{\Delta y} \quad (5.17)$$

while the local lateral heat flux from plane  $v$  to plane  $v-1$ ,  $q_{v,v-1(n)}$ , becomes,

$$q_{v,v-1(n)} = -k_s \frac{T_{v,n} - T_{v,n-1}}{\Delta y} \quad (5.18)$$

We present, in Figure 5.8, the local lateral heat flux profiles at nodal points (6) and (6,6) for the 1-D and 2-D models, respectively. In respect of the 1-D solution, we can observe that as the liquid film-boils at the heat exchange surface, the initial local lateral heat flux profiles based on the 1-D and 2-D solutions are approximately the same. However, within 13 seconds following the onset of transition boiling, these profiles began to differ. Again, the negligible

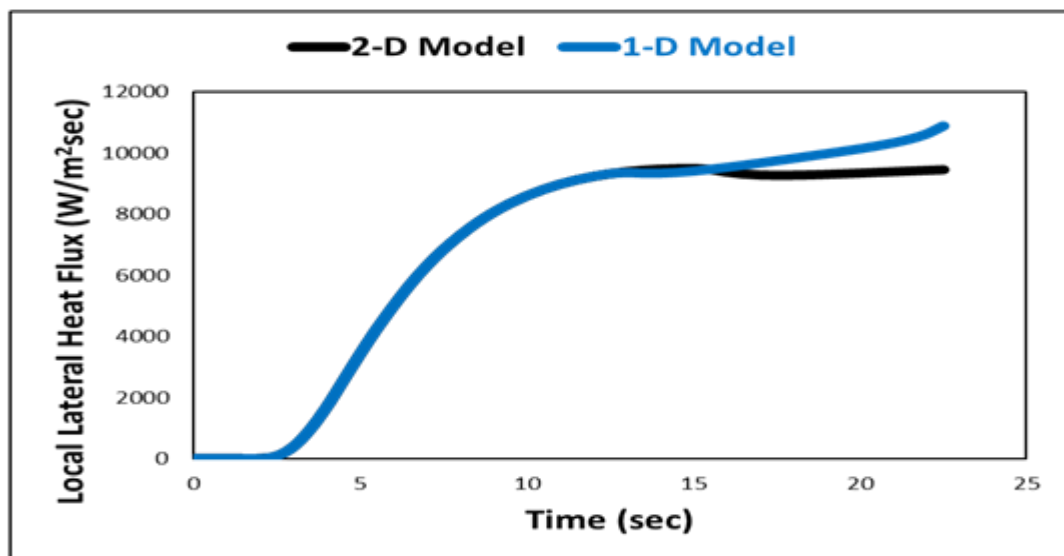


**Figure 5.8:** Comparison of the local lateral heat flux predicted by the 1-D solution to local lateral heat flux predicted by the 2-D solution.

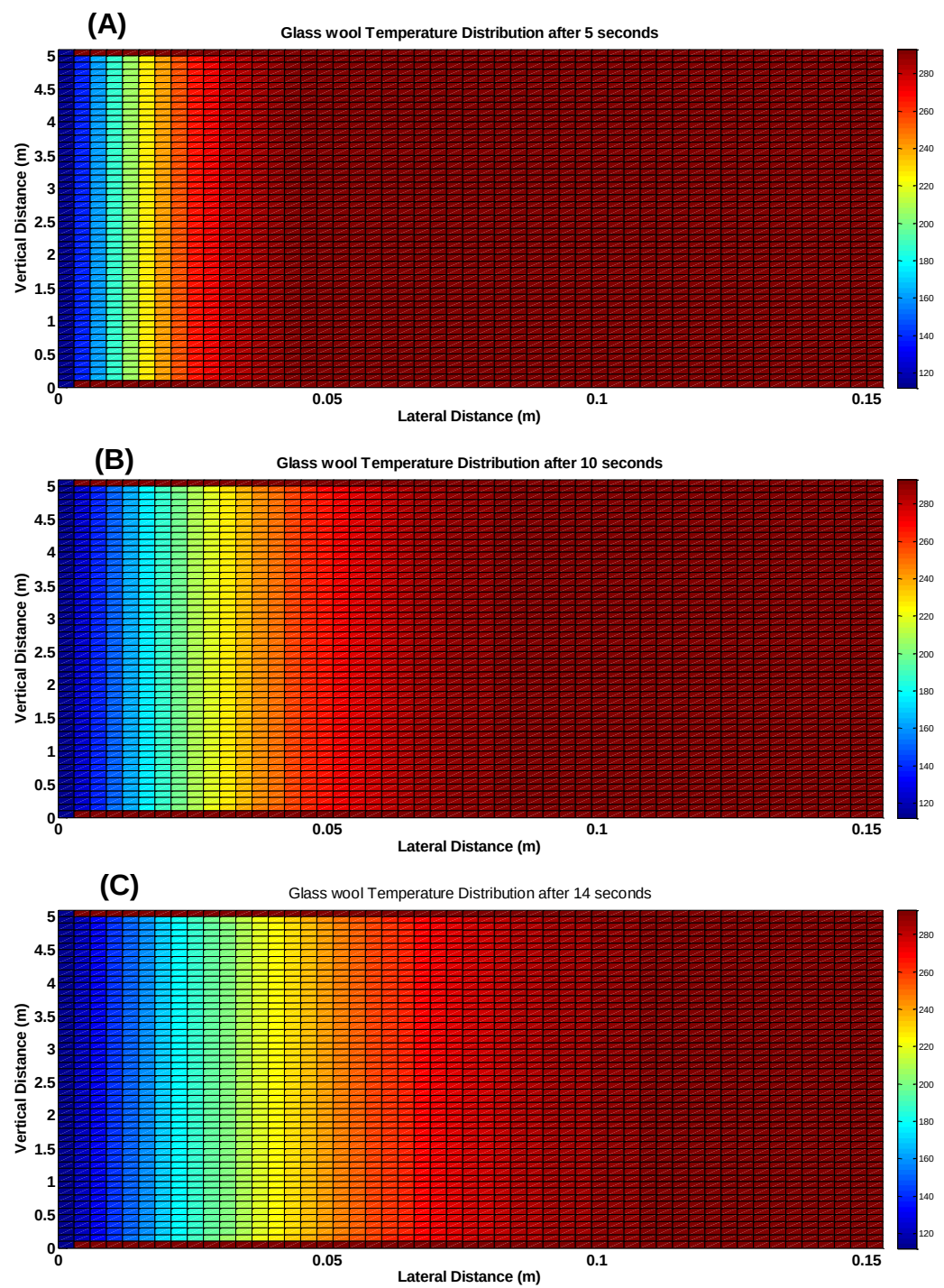
influence of substrate thermal properties during film boiling contributed to the initial similarities in the heat flux profiles of the two models. As time progresses, the difference between the 1-D and 2-D models is expected to increase and it may then be necessary to consider if the use of the 2-D model should be appropriate at larger times. In essence, a pragmatic approach to estimating heat transfer effects is to establish a characteristic time above which the use of a 1-D model becomes inadequate. A number of authors have proposed different time criterion for both LNG and other analogous processes including Webber (1987) and Phillips (1996).

### **5.1.2 Simulation Case #2: LNG Flow through Glass Wool**

The 1-D and 2-D models were also used to simulate the vaporizing flow of LNG through glass wool – an important porous material used in the design and construction of the cargo tanks used in membrane LNG ships. Because of the thermal properties of this substrate, the heat transfer mode would be solely nucleate boiling. We have taken the dimension of the problem to be the same as those discussed in Section 5.1.1. The 1-D and 2-D model comparison in this section is limited to analysing trends in temperature profiles, interface movement and interface velocity. However, it might be useful to first assess the trends in temperature distributions within the glass wool substrate.

#### Glass wool Temperature Distributions

Figures 5.9 (a-f) show the temperature distributions within the glass wool matrix at six different cooling times of 5, 10, 14, 20, 40 and 150 sec. A decrease in temperature is immediately observed close to the wetted side of the solid matrix after 5 sec. As time progresses, the varying temperature field within the solid matrix increases (see Figures 5.9 (b-f)), making it clear that the temperature distribution and heat penetration is strongly dependent on time. The variability of the temperature in the vertical direction, though not clearly noticeable at earlier times, became more noticeable at longer times of, say 150 sec. With such time-dependent temperature variation in both the horizontal and vertical directions, the validity of the 1-D model at late times may be doubtful because of the increasing influence of vertical heat conduction effects. Again, the top and bottom rows in Figures 5.9 (a-f) act as heat sources and produces the 2-D effect.



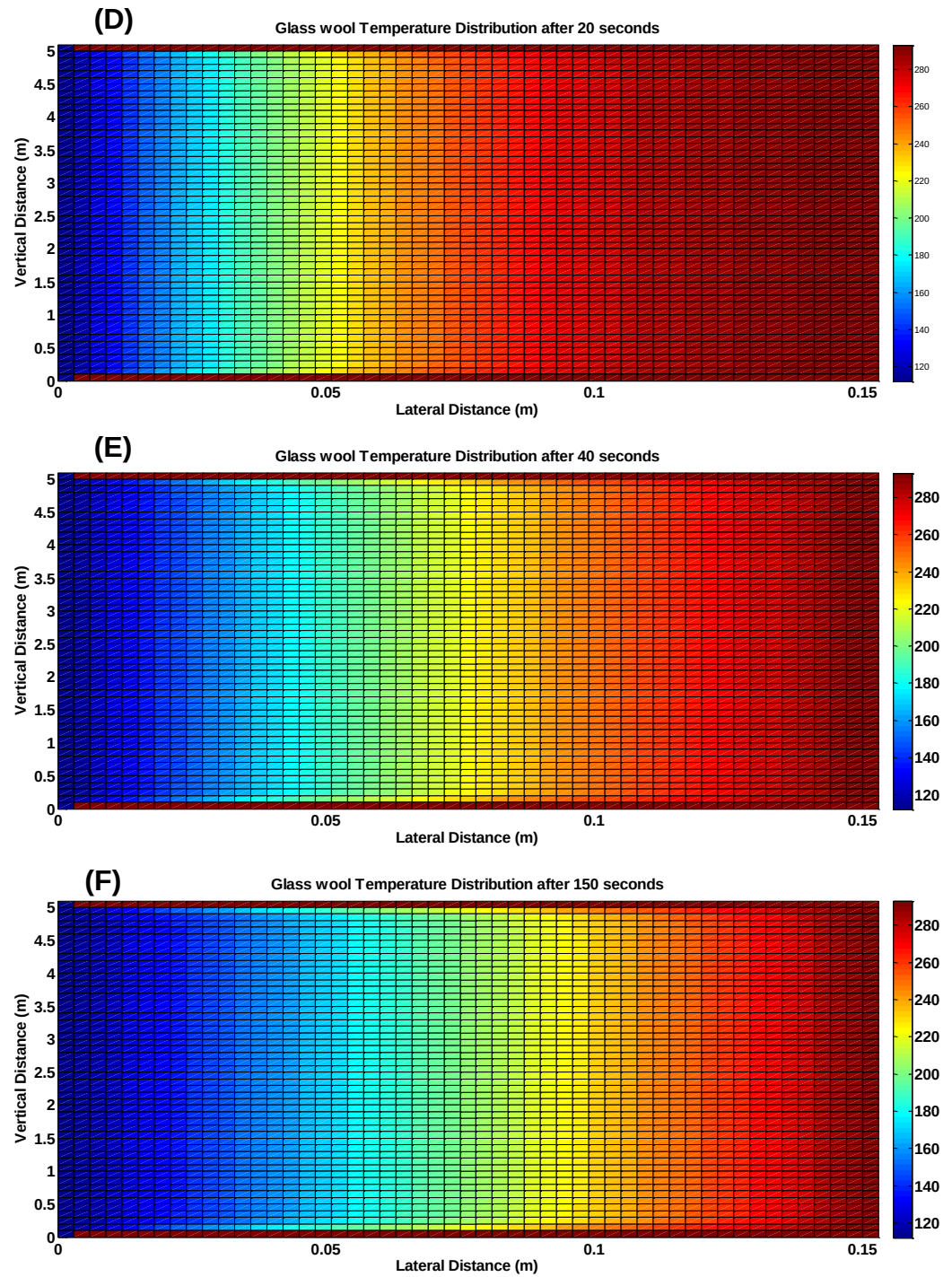


**Figure 5.9:** Temperature distribution in the glass wool at: (a) 5 sec, (b) 10 sec; (c) 14 sec; (d) 20 sec; (e) 40 sec; and (f) 150 sec.

A comparison of the 1-D and 2-D lateral temperature profiles at two different times are compared in terms of their percentage deviation. Figure 5.10 shows deviation plots based on the 1-D and 2-D solutions at location,  $z=100$  mm. The profiles are compared for cooling times of 10 and 100 sec, respectively. The black curve of Figure 5.10 indicates relatively smaller difference between the two

models after 10 seconds. At the longer timescale of 100 sec, we immediately notice an over 9 % deviation which is an indication that the difference between 1-D solution and the 2-D solution increases with time and the magnitude of this difference is controlled by the thermal properties of the cooled substrate. The vertical temperature profile results of the 1-D and 2-D solutions at lateral distances of 3 mm and 42 mm are compared in Figure 5.11, in terms of their respective percentage deviation. Compared to the 1-D solution, the percentage

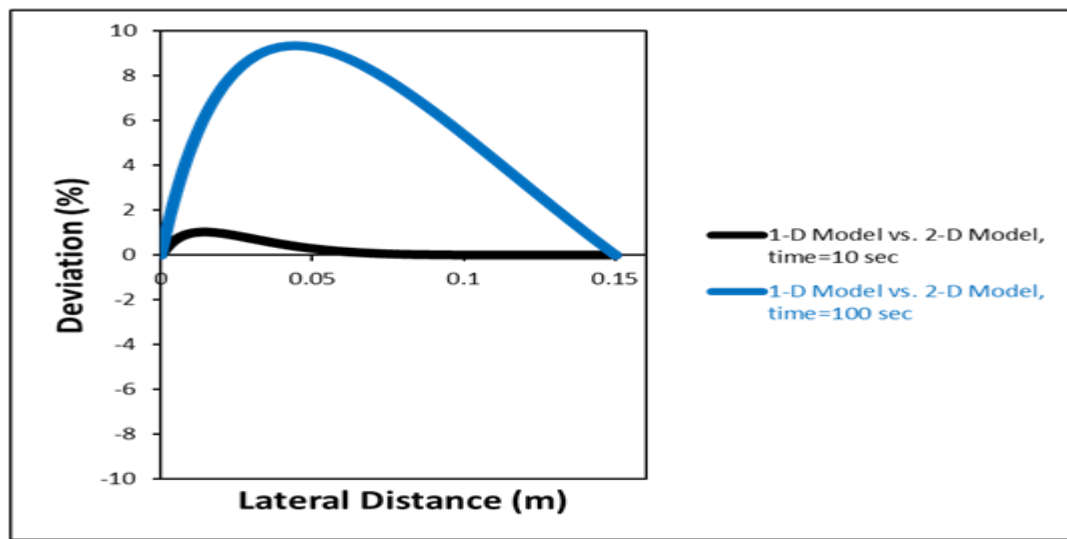


**Figure 5.10:** Comparison of the temperature distribution of the 1-D solution with those of the 2-D solution (in terms of percentage deviation) after 10 and 100 sec.

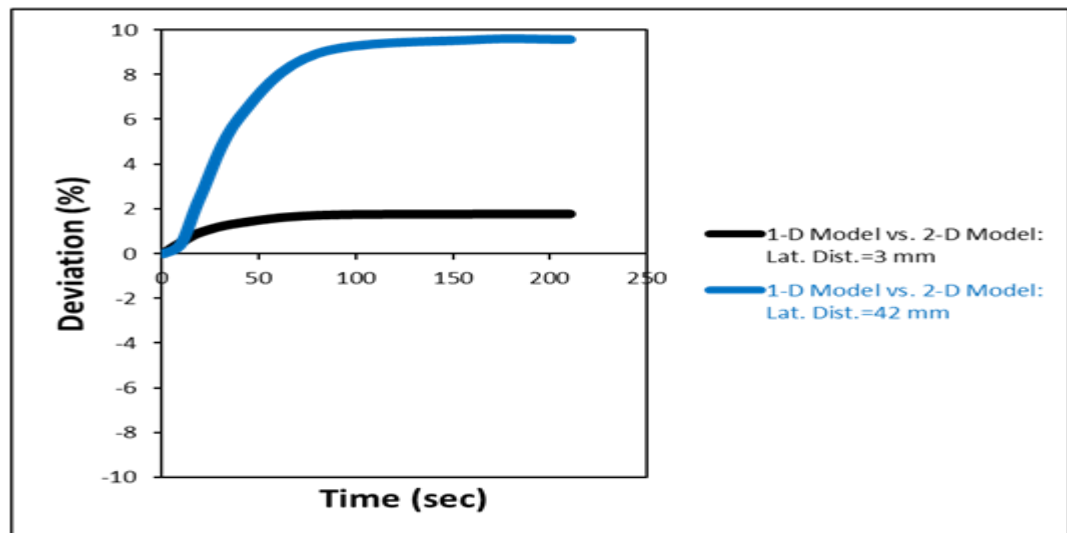


**Figure 5.11:** Comparison of the temperature profiles of the 1-D solution with the 2-D solution at a lateral distance of 3 and 42 mm, respectively in terms of percentage deviation.

deviation of the 2-D solutions was found to also increase with time. The increasing difference between the two models is indicative of the increasing influence of vertical heat conduction as time progresses. Again, a criterion for determining the validity of the 1-D solution can be based on the knowledge of a characteristic time above which the 1-D and 2-D solutions begin to significantly differ, thereby rendering the 1-D model increasingly inadequate for relatively longer times. In such situations, the use of the 2-D model should be recommended.

Figure 5.12 shows the front and/or interface movement (i.e. movement of the leading edge of the liquid) from the 1-D model compared with those obtained using the 2-D model. The reference depth is the maximum penetration depth of the liquid in the absence of heat and/or vaporization and this has also been plotted as the black curve of Figure 5.12. It is clearly evident from the graph that at approximately 5 sec following liquid entry into the porous layer, the motion of the liquid front based on the 1-D solution looks similar to the 2-D solution and

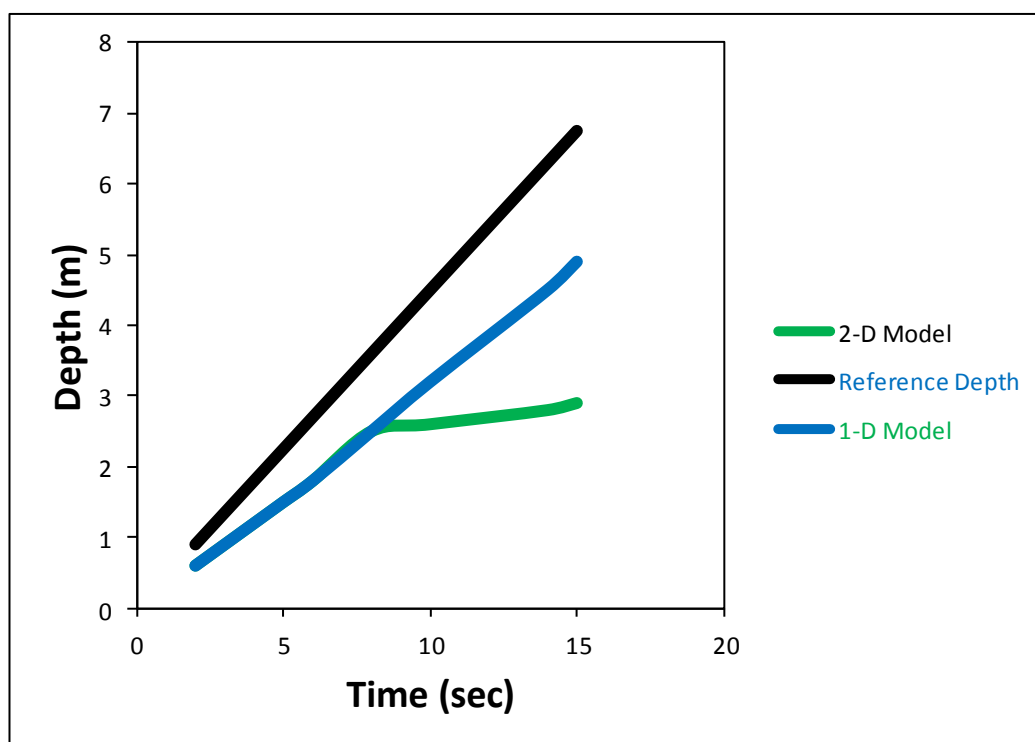


**Figure 5.12:** Interface movement for the 1-D and 2-D solutions during flow of LNG through glass wool.

this similarity ceases to exist after approximately 8 sec, which is when the 1-D and 2-D solutions began to noticeably differ, with this difference increasing with time. A closer examination of Figure 5.12 clearly shows that the liquid front has reached a depth of 4.9 m after 15 seconds using the 1-D model while the corresponding depth reached by the liquid front based on the 2-D model is 2.9 m. The retarding effect of vaporization on the motion of the liquid front is clearly evident for both solutions even at such an early time of 2 sec. Furthermore, heat conduction and vaporization effects resulted in the liquid front (for both solutions) only able to reach positions that are below the reference depth of 6.75 m. The higher prediction of the liquid penetration depth obtained for the 1-D model case is expected, since neglecting vertical heat conduction led to relatively smaller amount of heat supplied from the substrate, leading to relatively lower liquid vaporization rate which is then reflected in the relatively higher penetration rate of the liquid front (see blue curve of Figure 5.12) . Whereas for the 2-D simulation case, the vertical heat transfer effects, which was accounted for, increases the vaporization rate of the liquid, with the effects of vaporization then limiting the liquid penetration rate as the green curve of Figure 5.12 shows.

Figure 5.13 shows the calculated velocity of the liquid front/interface corresponding to the liquid front/interface curves presented in Figure 5.12. Again, the liquid front/interface velocity profiles based on the 1-D solution initially follows that of the 2-D solution until after approximately 8 seconds when the magnitude of the liquid front/interface velocity based on the 1-D and 2-D methods began to differ. This increasing difference in liquid front/interface velocity after 8 sec is in line with our expectation that the liquid front/interface will become less penetrative when a 2-D solution is applied. The application of the 2-D model results in a stronger effect of vaporization at any considered location down the porous layer, thus reducing liquid penetration rates. Ultimately, the computed liquid front/interface velocities for the 1-D solution is relatively higher than the results obtained based on the 2-D solution after 8 sec, which then explains the relatively faster motion of the liquid front/interface when the 1-D model was applied.

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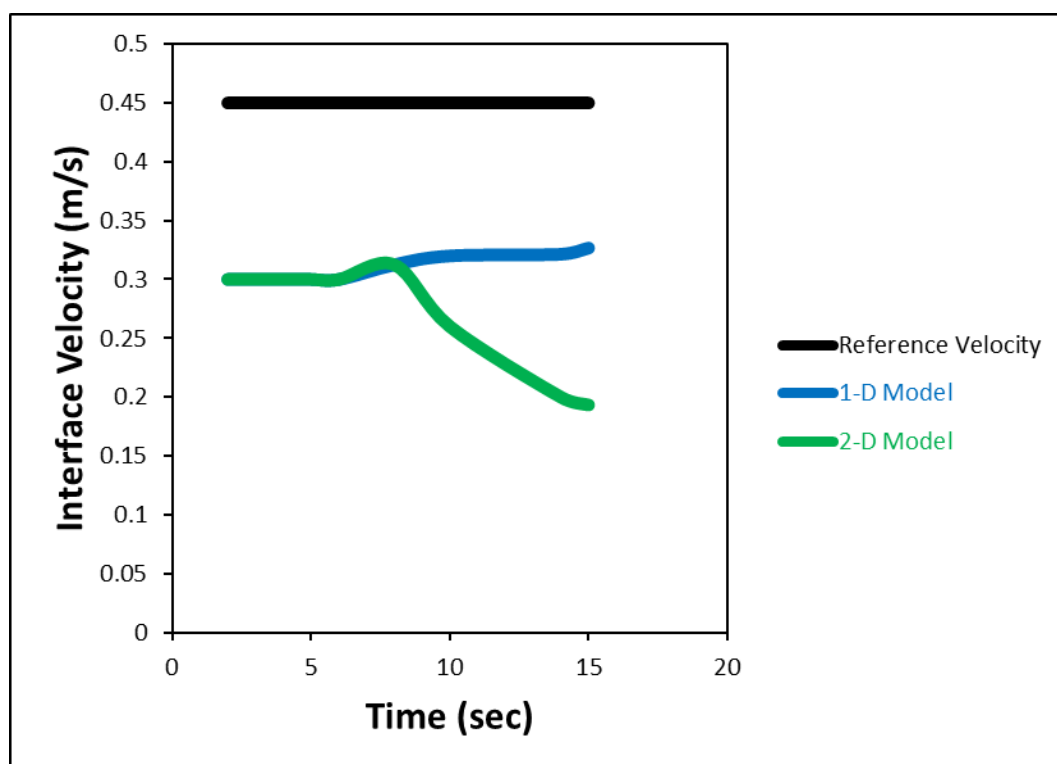


**Figure 5.13:** Interface velocity for the 1-D and 2-D solutions during flow of LNG through glass wool.

## 5.2 Discussion and Conclusions

The aim of this chapter was to examine the differences in the results of the 1-D model and the 2-D model solutions in order to provide some indication of the influence of vertical heat conduction effects. This is achieved by running sets of a 1-D and a 2-D simulation of a vaporizing flow of liquids (LNG) through soil and glass wool porous material and then comparing some of the results of the 1-D solution with those obtained using the 2-D solution. Some conclusions drawn from this chapter include:

- 1) A 2-D model for the prediction of the vaporizing flow of liquid through porous media was compared with the corresponding 1-D model. The model solves an advection equation which has been coupled to a transient, two-dimensional heat conduction equation in rectangular coordinates. Unlike the 1-D model which assumes purely horizontal (lateral) heat conduction effects, the 2-D model additionally incorporates the effects of vertical heat conduction within the solid matrix and it appears that the applicability of

the 1-D model can only be valid at a characteristic time that is mainly a function of the thermal properties of the substrate such as thermal diffusivity.

- 2) The vaporizing flow of LNG through dry soil will require the application of a time-varying heat transfer coefficient at the interface between the flowing liquid and the solid matrix. As the heat transfer coefficient is time-varying, the implementation of a full boiling heat transfer coefficient in the models allows for the prediction of the so-called three-regime boiling. Results from a comparative study of the 1-D and 2-D models reveal that the two models are different and that the 1-D model should be valid until a characteristic time, say  $t_{cr}$ , is exceeded. Above  $t_{cr}$ , the vertical heat conduction effects may be considered sufficiently important to warrant the use of the 2-D model.
- 3) In the vaporizing flow of LNG through glass wool, the heat transfer mechanism is essentially nucleate boiling regime. Results obtained from a comparison between the 1-D and 2-D solutions clearly differ and this difference increases with time. Thus, in order to predict the vaporizing flow of LNG through glass wool porous material, the 2-D model should be employed in order to account for the effects of heat conducted vertically through the substrate.

Vaporizing liquid flow through porous or penetrable media can sometimes be obstructed by a non-penetrable plain medium such as may occur following seepages of liquids through soil and onto groundwater or direct seepages of liquids onto non-penetrable steel or compacted soil. In such situations, vaporizing pools are formed and it might be useful to understand the behaviour of the pool forming on such surfaces. We address the problem of vaporizing pools on non-penetrable solid substrates in Chapter 6.

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# 6

## ***Modelling Vaporizing Liquid Pools on Non-penetrable, Solid Substrates***

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We presented, in Chapter 3 through 5, models describing the vaporizing flow of liquids through porous or penetrable media. We demonstrated with these models that heat conduction and vaporization effects affect liquid penetration rates. However, it is also expected that penetration of the liquid downward through the porous medium will, for the constant injection case considered in this work, continue until they encounter a non-penetrable solid substrate. Figure 6.1 below

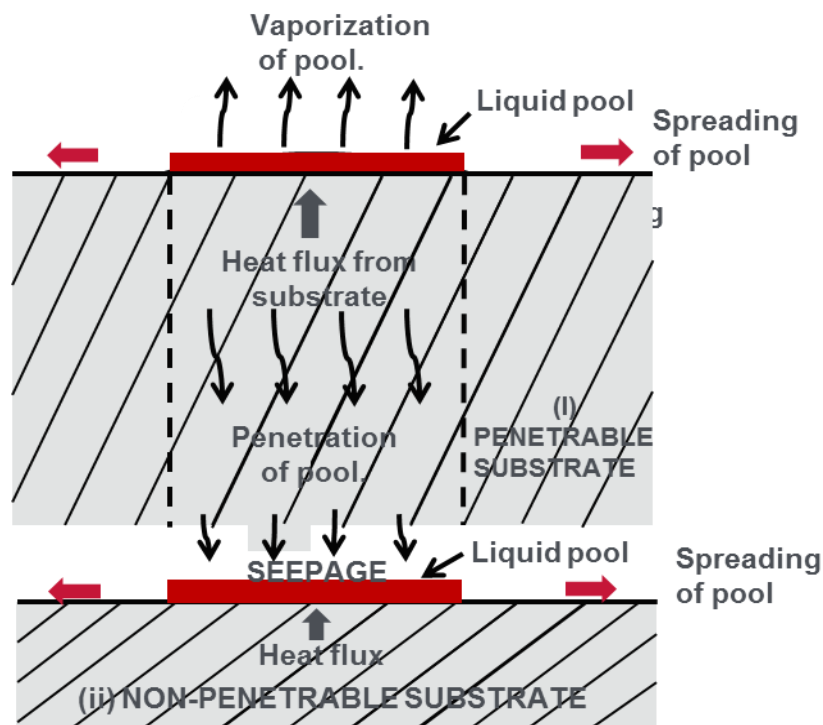


**Figure 6.1:** Schematic diagram of liquid spillage, pool formation and spreading, downward flow through the penetrable substrate and seepage onto non-penetrable substrate. We are interested in the fate of the liquid following seepage.

is a schematic diagram describing such a spill scenario. The aim of this chapter is, therefore, to present the implementation of a vaporizing pool model that describes the spreading and vaporization of a pool of liquid following seepage onto non-penetrable, solid substrates. The presentation in this chapter is organized into five sections. We presented, in Section 6.0, details of the liquid pool model implemented. Section 6.1 discussed the numerical solution, verification and validation of the implemented pool model. In Section 6.2, results of sample simulation cases are presented. In the last and final section (Section 6.3), the discussion and conclusions from this chapter are presented.

## 6.0 Liquid Pool Modelling

Solution predictions from pool models provide risk-based safety information on the total amount and duration of hazardous vapour that will evolve in the event of an accidental spill and/or leak (Woodward and Pitblado, 2010). These models are usually interfaced with both dispersion and pool fire predictive models. This section discusses the liquid pool modelling that describes liquid behaviour in the event of accidental seepages or spillages.

### 6.0.1 Typical Liquid Pool Behaviour following a Spill

An accidental spill of liquid onto non-penetrable, solid substrates could lead to the development of a pool that will simultaneously spread and vaporize. The cloud of vapour that is formed above the pool will disperse and the distance travelled by the dense vapour-cloud will depend on the atmospheric condition (e.g. wind speed) prevailing at the time of spill. Figure 6.2 illustrates a simple schematic of a liquid pool that is spreading and vaporizing on a non-penetrable solid surface. Here we see that the liquid pool of radius,  $R(t)$ , and height,  $H$ , spreading on the solid surface will cool the surface until a time when the solid surface temperature begins to decrease from its initial value. Depending on the thermophysical properties of the solid substrate, the decreasing temperature of the substrate surface will lead to a change in the boiling regime, first from the initial film boiling regime with a vapour film of thickness,  $\delta$ , to transition boiling regime and finally to nucleate boiling regime. The spreading pool vaporizes by absorbing the heat that is mainly supplied to it by the non-penetrable substrate.

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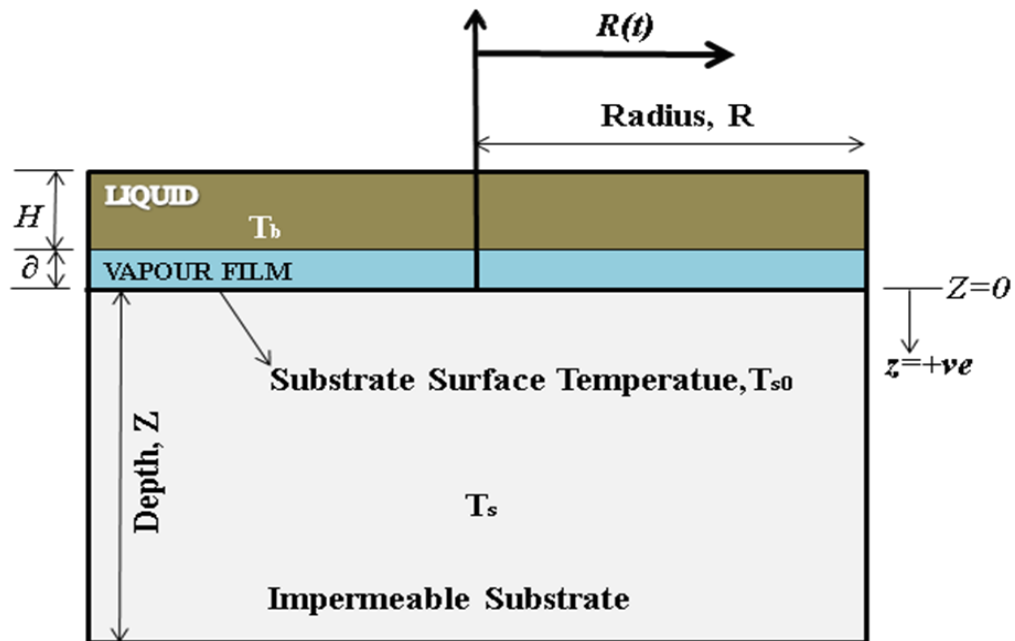


**Figure 6.2:** Schematic of a liquid pool spreading and vaporization model when the heat transfer at the heat exchange surface is solely controlled by film boiling.

### 6.0.2 Liquid Pool Model Assumptions

A realistic liquid model must be able to describe the most important physical effects of the simultaneous spreading and vaporization of the pool of liquid. For the purposes of this work, unless otherwise stated, the following simplifying assumptions apply:

1. The liquid source is just above the solid surface.
2. The solid surface is flat
3. The solid surface is frictionless.
4. The liquid spill is instantaneous.
5. The liquid spreads radially as a cylinder of radius,  $R$ , and height,  $H$ .
6. The pool spreading occurs solely in the gravity-inertia regime.
7. The effects of wind is negligible.
8. The change in substrate surface temperature is a function of time.
9. The boiling occurs in the film-boiling regime.

The “gravity-inertia” regime assumption is used to explain the fact that while gravitational forces induce the pool spreading, the forces of inertia oppose the spreading.

### 6.0.3 Heat Transfer to the Liquid Pool

Though the heat supply to the liquid pool has been proven to be dominated by the heat provided by the solid substrate (which is the only heat source assumed in the present work), other heat transfer contributions from solar radiation, atmospheric radiation and atmospheric convection (Cavanaugh et al., 1994; Dienhart, 1995) can also be included. For ignited pools, additional contribution from thermal radiation from pool fires can be significant (Hissong, 2008). In this work, we applied Equation 2 of Conrado and Vesovic (2000) for the modelling of the heat flux into the pool, repeated here for convenience,

$$q = h(T_s - T_b) \quad (6.1)$$

where  $T_s$  now refers to the time-varying surface temperature of the substrate surface (K) and  $T_b$  is the liquid boiling temperature (K). The film boiling heat transfer coefficient ( $\text{W/m}^2/\text{K}$ ),  $h$ , is estimated using the correlations of Klimenko (1981) presented in Section 2.0.3.1 of Chapter 2. The surface temperature of the substrate,  $T_n$  is determined via a numerical solution of the Fourier’s law (Ozisik, 1993) of heat conduction (see Equation 4.1 of Chapter 4) with a well-known analytical solution (for film boiling conditions) given by the following equation (Carslaw and Jaeger, 1959b),

$$T_n = T_b + (T_{n0} - T_b) \operatorname{erf} \frac{x}{(4\alpha_n t)^{1/2}} + \exp \left( \frac{hx}{k_n} + \frac{h^2 t}{k_n \rho_n C_n} \right) \operatorname{erfc} \left( \frac{x}{(4\alpha_n t)^{1/2}} + \frac{h(t)^{1/2}}{(k_n \rho_n C_n)^{1/2}} \right) \quad (6.2)$$

where  $T_{n0}$  refers to the initial temperature of the solid substrate (K),  $x$  is the substrate depth (m),  $\alpha_n$  is the substrate thermal diffusivity ( $\text{m}^2/\text{sec}$ ),  $t$  is the time (sec),  $\rho_n$  is the substrate density ( $\text{kg/m}^3$ ), and  $C_n$  is the substrate specific heat capacity ( $\text{J/kg/K}$ ).

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#### 6.0.4 Liquid Pool Spreading Rate

A pool spreading rate model as proposed by Vesovic (2007) was used to describe the spreading behaviour of the liquid pool. As already noted in Section 6.0.2 of this chapter, the model assumes that the spill is instantaneous with the flat circular liquid pool spreading on the solid surface with a spreading relationship given by (Vesovic, 2007),

$$\frac{dR}{dt} = \frac{(A_{par}M)^{1/2}}{R} \quad (6.3)$$

where  $M$  is the mass of the liquid pool and  $R$  is the radius of the liquid pool. The parameter,  $A_{par}$  is given by (Conrado and Vesovic, 2000; Woodward and Pitblado, 2010),

$$A_{par} = 2.6 \left[ \frac{g}{\pi \rho_{lp}} \right] \quad (6.4)$$

$g$  is the acceleration due to gravity and  $\rho_{lp}$  is the density of liquid pool.

#### 6.0.5 Liquid Pool Vaporization Rate

The liquid vaporization rate can be calculated using the following first order ordinary differential equation (ODE) (Conrado and Vesovic, 2000; Vesovic, 2007),

$$V \cong -\frac{dM_{vap}}{dt} = \frac{\pi R^2 q}{L_{vap}} \quad (6.5)$$

where  $V$  is the vaporisation rate and  $M_{vap}$  is the mass of vaporised liquid.

### 6.1 Numerical Solution of the Liquid Pool Model

The numerical solution of the liquid pool model discussed in Section 6.0 is presented in this section. The coupled differential equations describing the spreading (Equations 6.3 and 6.4) and vaporisation (Equation 6.5) of the liquid

pool are solved using the Fourth-Order-Runge-Kutta method (Conrado and Vesovic, 2000) subject to the following initial condition,

$$t = 0 \qquad M = M_0 \qquad R = 0 \qquad (6.6)$$

where  $M_0$  is the initial mass of the liquid spilled. The values of the remaining liquid mass and pool radius at new time step were calculated using the values from the previous time step. The solution of the system of differential and algebraic equations presented in this chapter were coded in FORTRAN. Figure C.1 of Appendix C is a flowchart showing the program structure for the liquid pool model. At the start of the simulation, the relevant initial condition and parameters are specified. Input parameters are initial substrate temperature, thermal properties of substrate, thermal properties of liquid, amount of liquid spilled, liquid pool radius, time and space discretization and run time. The transition and film boiling heat transfer coefficients between the pool and the substrate are calculated using Equations 2.10 and 2.11, respectively. The surface temperature of the non-penetrable solid substrate is calculated by using the Fourier's law of heat conduction (Ozisik, 1993; Carslaw and Jaeger, 1959b). The heat flux from the substrate and the vaporisation rate of the liquid were then calculated using Equations 6.1 and 6.5, respectively. Knowing the magnitude of the liquid pool radius and vaporisation rate, the change in liquid mass is then calculated by making use of the mentioned Fourth-Order-Runge-Kutta method (Conrado and Vesovic, 2000; Vesovic, 2007). The program exits when the change in mass becomes equal to zero. Full details of the source code and algorithm are contained in an Appendix CD.

### **6.1.1 Verification of Liquid Pool Model**

The aim of this verification study is to test and determine if the numerical solutions employed are accurately representing the mathematical models used for describing the liquid pool behaviour. The verification methodology is simple as we are comparing the numerical model with its analytical solution which, in the case of confined spill under film boiling, can be readily derived. Hence, we have derived the analytical solution of the liquid pool model by integrating and rearranging Equation 6.5. The new expression which describes the mass of liquid

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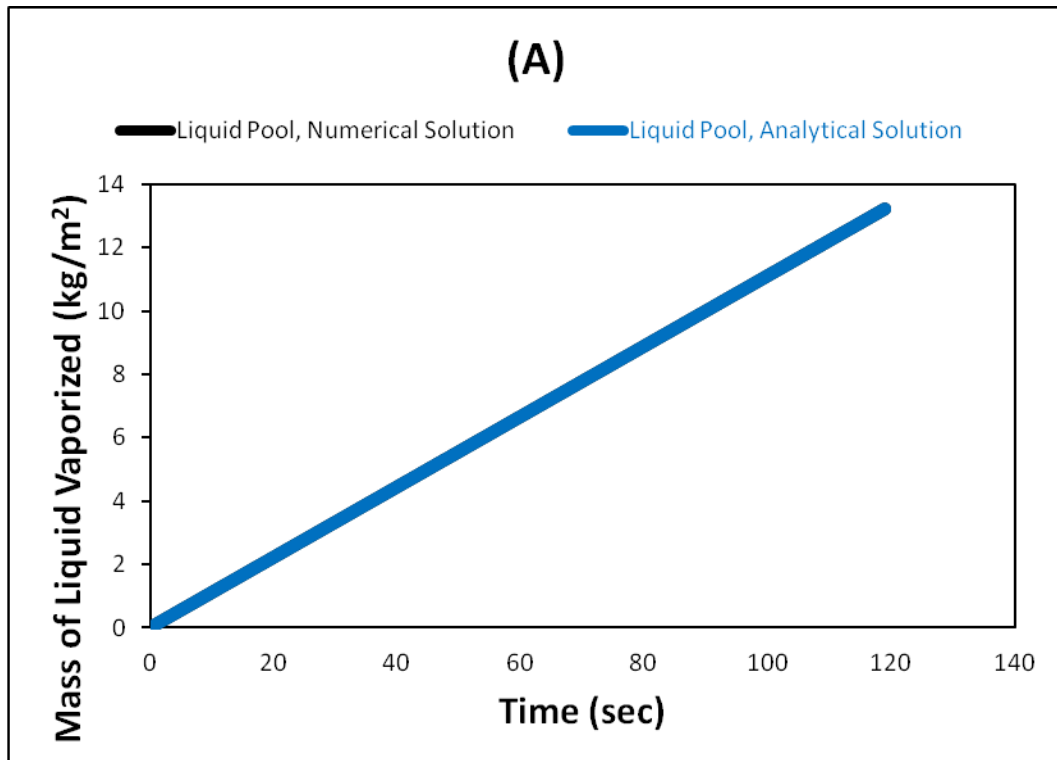
pool remaining,  $M_{REM}$ , as a function of initial mass spilled is given by Equation 6.7,

$$M_{REM} = M_0 - \left[ \frac{\pi R^2 q t}{L_{vap}} \right] \quad (6.7)$$

where all parameters are as defined previously. The relationship for the maximum vaporization time,  $t_{max}$ , can be obtained by rearranging Equation 6.7 given,

$$t_{max} = \frac{(M_0 - M_{REM})L_{vap}}{\pi R^2 q} \quad (6.8)$$

In Figure 6.3 (a), we have compared the results of the numerical solution with its corresponding analytical solution and the agreement is excellent as can be confirmed by the trend in their relative error estimation plot shown in Figure 6.3 (b).



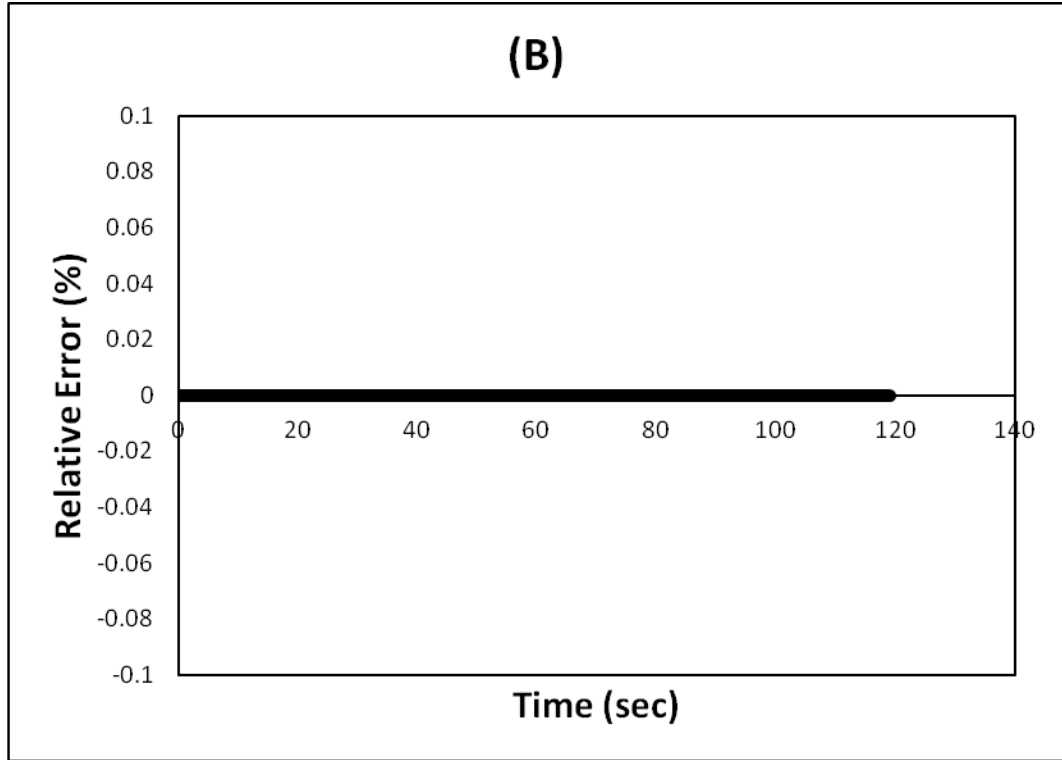


**Figure 6.3:** Liquid pool model verification – comparison of numerical solution with its corresponding analytical solution: (a) mass of liquid vaporised versus time, (b) relative error analysis.

### 6.1.2 Validation of the Liquid Pool Model

In validating the liquid pool model, we have selected the experimental data of Reid (1980). These problems focus on the measurement and interpretation of vaporization rates of LNG spilled on substrates that are or could be used as dike floor materials. As already discussed in Section 2.1.2 of Chapter 2, different types of substrates were tested in their work including soil, sand, insulating concrete, crushed rock, polyurethane and corrugated aluminium over soil. Reid (1980) reported that the soil, sand and concrete results could be correlated by the following simple one-dimensional heat transfer conduction model where the vaporization rate is given as the product of a constant multiplied by the inverse square root of time,

$$V_{reid} = \frac{(q)}{L_{vap}} = \left( \frac{B_{reid}}{L_{vap}} \right) t^{-1/2} \quad (6.9)$$

where the heat flux,  $q$ , is given by,

$$q = B_{reid} t^{-1/2} \quad (6.10)$$

and a boiling parameter,  $B_{reid}$ , given by,

$$B_{reid} = \sqrt{\frac{(k_s \rho_s C_{ps})}{\pi} (T_0 - T_b)} = \frac{S_{reid} L_{vap}}{2} \quad (6.11)$$

The parameters  $V_{reid}$ ,  $L_{vap}$ ,  $t$ ,  $k_s$ ,  $\rho_s$ , and  $C_{ps}$  are as defined previously. The parameter  $S_{reid}$  denotes the slope of the vaporized mass versus time curve as explained by Reid (1980) in his report.

We selected Tests 38 and 40 of Reid (1980) for validation purposes in this chapter. A dry sand test, Test 38, describes experiments with a sand substrate while a dry soil test, Test 40, describes Reid's experiments with a soil substrate. Three experiments were made with sand and seven experiments made with soil. Table 6.1 and 6.2 provided more details of these experiments. The substrate used

**Table 6.1** More details of Soil Tests in Reid's Experiments (Reid, 1980).

| <i>Test No.</i> | <i>Moisture<br/>(% by<br/>weight)</i> | <i>Initial Soil<br/>Temperat<br/>ure (K)</i> | <i>Soil<br/>Density<br/>(kg/m<sup>3</sup>)</i> | <i>Mass of<br/>Liquid<br/>Methane<br/>Poured<br/>(kg/m<sup>3</sup>)</i> | <i>Initial<br/>Liquid<br/>Depth<br/>(cm)</i> |
|-----------------|---------------------------------------|----------------------------------------------|------------------------------------------------|-------------------------------------------------------------------------|----------------------------------------------|
| 21              | 8.2                                   | 291                                          | 1300                                           | 3.85                                                                    | 0.92                                         |
| 40              | ~0                                    | 295                                          | 1300                                           | 6.63                                                                    | 1.58                                         |
| 41              | 8.2                                   | 293                                          | 1300                                           | 4.29                                                                    | 1.00                                         |
| 42              | 4.0                                   | 292                                          | 1400                                           | 6.69                                                                    | 1.60                                         |
| 44              | 8.2                                   | 293                                          | 1800                                           | 8.47                                                                    | 2.02                                         |
| 45              | 8.2                                   | 280                                          | 1800                                           | 12.3                                                                    | 2.94                                         |
| WR47            | 4.0                                   | 293                                          | 1800                                           | 6.45                                                                    | 1.55                                         |

in Test 40 was typical clay while a material similar to that used in normal concrete was used as substrate for Test 38. For both tests, the materials were first placed in a constant humidity, constant temperature jar for different samples. Before carrying out any of the tests, substrate samples were taken and

their water content measured by the weight loss after drying in an oven. The solid substrate which is most times 6 cm deep were prepared in a Styrofoam text box and thermocouples placed at different depths from the substrate surface.

**Table 6.2** More details of Sand Tests in Reid's Experiments (Reid, 1980).

| <i>Test No.</i> | <i>Moisture<br/>(% by<br/>weight)</i> | <i>Initial Soil<br/>Temperat<br/>ure (K)</i> | <i>Soil<br/>Density<br/>(kg/m<sup>3</sup>)</i> | <i>Mass of<br/>Liquid<br/>Methane<br/>Poured<br/>(kg/m<sup>3</sup>)</i> | <i>Initial<br/>Liquid<br/>Depth<br/>(cm)</i> |
|-----------------|---------------------------------------|----------------------------------------------|------------------------------------------------|-------------------------------------------------------------------------|----------------------------------------------|
| 38              | ~0                                    | 295                                          | 1600                                           | 68.8                                                                    | 2.08                                         |
| 39              | 1.3                                   | 292                                          | 1200                                           | 8.39                                                                    | 2.00                                         |
| 43              | 3.0                                   | 292                                          | 1500                                           | 6.97                                                                    | 1.66                                         |

The test conditions for the sand and soil tests (i.e. Tests 38 and 40) used in the present work are further shown in Table 6.3. Reid stated in his report that all simulations can be correlated in a linear region by a single straight line. It was clearly pointed out in Reid's report that there is significant "tailing", an occurrence attributable to the penetration of the LNG into substrate. Reid's heat flux calculations for the period of nucleate boiling can be written as,

$$\frac{M_{tot}}{A} = \int_0^t \frac{k_s(T_0 - T_b)}{\sqrt{\pi\alpha_s t}} dt = \frac{2k_s(T_0 - T_b)}{\sqrt{\pi\alpha_s}} t^{1/2} \quad (6.12)$$

and the corresponding heat flux calculation during film boiling is given as,

$$\frac{M_{tot}}{A} = \frac{2k_s^2}{\sqrt{\pi}} \frac{(T_0 - T_b)}{\alpha_s h} \int_0^Y E(Y) dY \quad (6.13)$$

where  $Y$ ,  $dY$  and  $E(Y)$  are defined, respectively, as,

$$Y = \frac{h\sqrt{\alpha_s t}}{k_s} \quad (6.14a)$$

$$dY = \left( \frac{h}{k_s} \right) \left( \frac{\alpha_s}{t} \right)^{1/2} \left( \frac{1}{2} \right) dt \quad (6.14b)$$

$$E(Y) = \sqrt{\pi}(Y)e^{(Y \times Y)} \operatorname{erfc}(Y) \quad (6.14c)$$

The variable  $M_{tot}$  in Equations 6.12 and 6.13 denote the total mass of LNG vaporized while all other variables are as defined previously. Reid (1980) based his work on the assumptions that:

- The substrate is homogenous and has constant properties.
- The LNG temperature is constant and equal to the boiling temperature
- The spill occurs instantaneously
- No side-wall effects are present, i.e., the substrate is infinite in both dimensions perpendicular to the heat flux.
- The substrate has infinite depth with a finite film boiling heat transfer coefficient on the substrate surface.

**Table 6.3** Details of Tests 38 and 40 of Reid's Experiments (Reid, 1980).

| Property                                                 | TEST 38  | TEST 40  |
|----------------------------------------------------------|----------|----------|
| Substrate Type                                           | DRY SAND | DRY SOIL |
| Density of Substrate (kg/m <sup>3</sup> )                | 1600     | 1300     |
| Thermal Diffusivity (10 <sup>-6</sup> m <sup>2</sup> /s) | 3.88     | 5.90     |
| Area (m <sup>2</sup> )                                   | 0.2      | 0.2      |
| Thermal Conductivity (W/m/K)                             | 5.18     | 6.4      |
| Substrate Thickness (m)                                  | 0.06     | 0.06     |
| Mass Spilled (kg)                                        | 1.75     | 1.33     |
| Mass Flux (kg/m <sup>2</sup> )                           | 8.75     | 6.63     |
| Initial Temperature of Substrate (K)                     | 295      | 295      |
| Initial Spill Radium (m)                                 | 0.25     | 0.25     |

Figures 6.4 (a) presents the performance of the present liquid pool model against Reid's data set for Test 38, in terms of the mass of LNG vaporized as a function of time. The corresponding relative error plot shown in Figure 6.4 (b) shows a two percent discrepancy between the implemented pool model and Reid's measurement. Despite this discrepancy, we believe there is quite good agreement

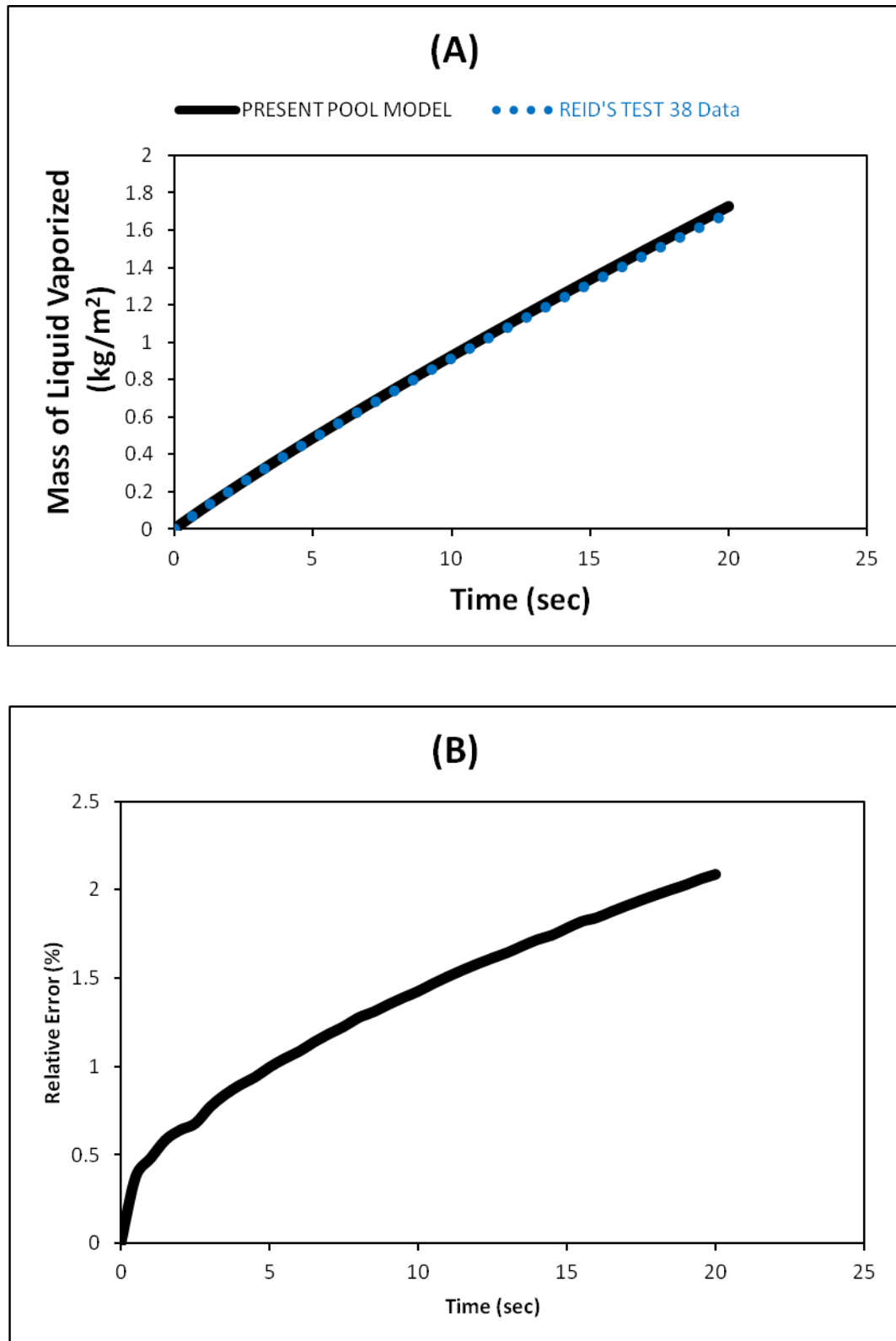


**Figure 6.4:** Liquid pool model validation – comparison of liquid pool model with the corresponding Reid's data set (Reid, 1980) for Test 38 showing the total mass vaporized as a function of time for a small scale spill of LNG on dry sand: (a) mass of liquid vaporised versus time, (b) percent error analysis.

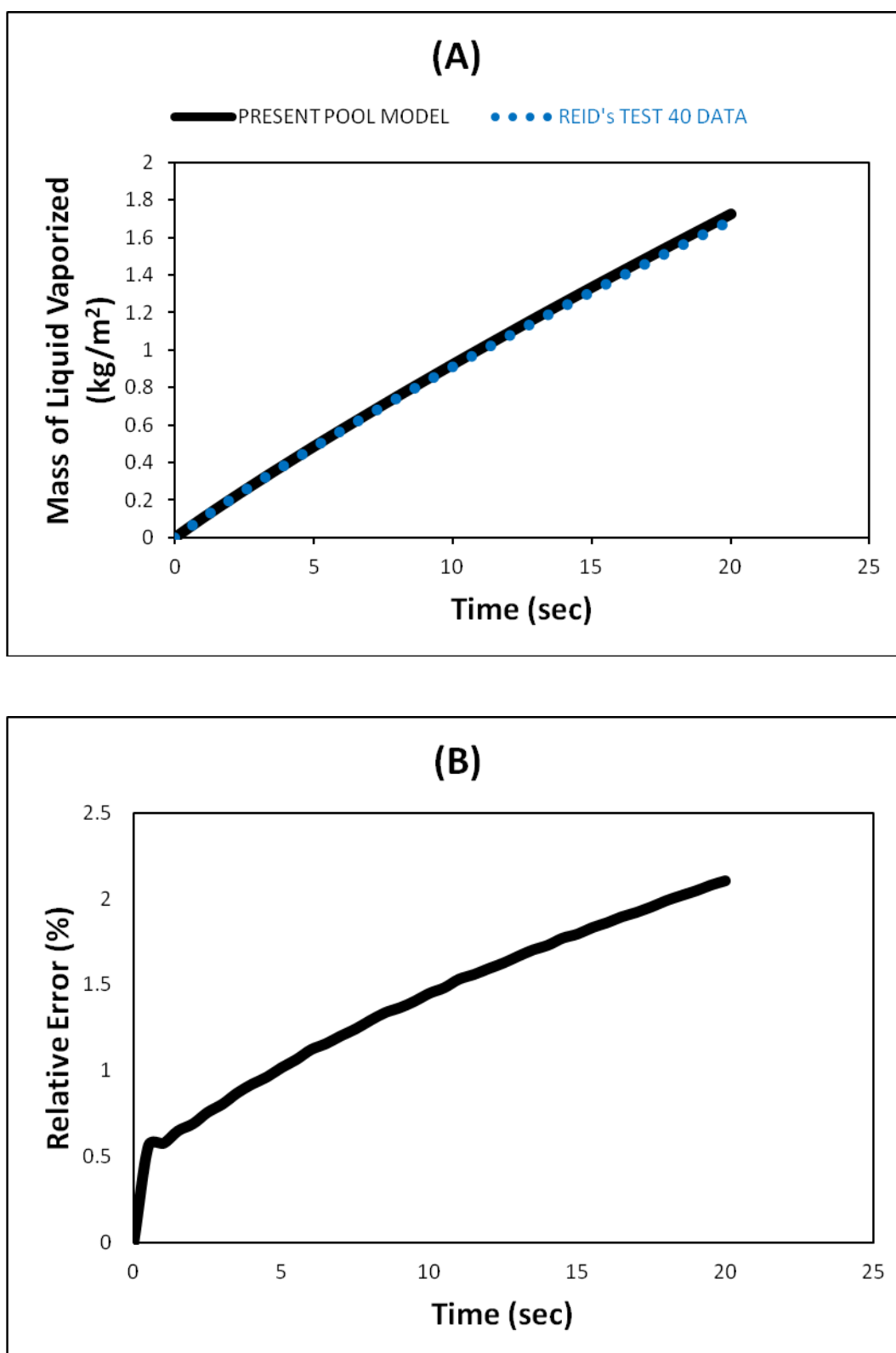


**Figure 6.5:** Liquid pool model validation – comparison of liquid pool model with the corresponding Reid's data set (Reid, 1980) for Test 40 showing the total mass vaporized as a function of time for a small scale spill of LNG on dry soil: (a) mass of liquid vaporised versus time, (b) percent error analysis

between the two results. The timescale for the variation in total mass vaporized was well captured by the liquid pool model. Figures 6.5 (a) and 6.5 (b) show the validation process for the dry soil test (Test 40) where good agreement between the liquid pool model and measurement is further demonstrated, with the discrepancy still falling within the two percent (2 %) range. The similarity of the curves in the sand and soil tests also supports Reid's observation that a spill on soil and sand do not significantly differ in their vaporization behaviour (Reid, 1980).

## 6.2 Results from Sample Simulation Cases

This Section presents results of sample simulation cases involving the use of LNG as the test fluid. For most of the analysis presented here, a relatively small volume of LNG is instantaneously spilled onto flat soil substrate with the pool spreading on the substrate surface limited by a confined enclosure of area, 0.2 m<sup>2</sup>. The substrate is of depth 0.06 m. This assumption ensures that the influence of spreading is so small that a pool diameter of 500 mm can be specified prior to the start of the simulation. Some of the simulation results presented are used to demonstrate LNG behaviour following a spill. A tabular listing of liquid and substrate thermal properties are shown in Table 6.4 and 6.5, respectively. Table

**Table 6.4** Properties of LNG (NIST WEBBOOK, 2011).

| <i>Liquid Property</i>              | <i>LNG</i>           |
|-------------------------------------|----------------------|
| Thermal Conductivity (W/m/K)        | 0.193                |
| Density (kg/m <sup>3</sup> )        | 422                  |
| Viscosity (Pas)                     | 113x10 <sup>-6</sup> |
| Latent Heat of Vaporisation (KJ/kg) | 512                  |
| Specific Heat Capacity (J/kg/K)     | 3350                 |
| Surface Tension (N/m)               | 0.014                |

**Table 6.5** Thermal properties of soil and steel (SST) (Shaw and Briscoe, 1978; Lee et al., 2011; NIST WEBBOOK, 2011).

| <i>Substrate Property</i>                                | <i>SOIL</i> | <i>STEEL</i> |
|----------------------------------------------------------|-------------|--------------|
| Thermal Conductivity (W/m/K)                             | 5.18        | 15.0         |
| Density of Substrate (kg/m <sup>3</sup> )                | 1600        | 7860         |
| Thermal Diffusivity (10 <sup>-6</sup> m <sup>2</sup> /s) | 3.88        | 3.82         |
| Specific Heat Capacity (J/kg/K)                          | 835         | 500          |

6.6 tabulates the conditions for the liquid pool simulations.

**Table 6.6** Values of parameters used for liquid pool simulations.

| <i>Parameter</i>                                                          | <i>Value</i> |
|---------------------------------------------------------------------------|--------------|
| Initial Substrate Temperature (K)                                         | 295          |
| Liquid Boiling Temperature (K)                                            | 112          |
| Gravitational Acceleration ( $\text{m}^2/\text{s}$ )                      | 9.81         |
| Film Boiling Heat Transfer Coefficient ( $\text{W}/\text{m}^2/\text{K}$ ) | 155          |
| Mass of Liquid Spilt (kg)                                                 | 0.2          |
| Pool Radius (m)                                                           | 0.252        |
| Depth of Substrate (m)                                                    | 0.06         |

Figure 6.6 shows the temperature-time history for a spill of LNG on soil. Because of the sufficiently large difference in temperature between LNG and the soil surface, a stable vapour film separating the soil and the cryogen limits the heat transfer rate as a result of its relatively low thermal conductivity. It can be seen from Figure 6.6 that it required 380 seconds for the surface temperature of soil to decrease towards the boiling temperature of the liquid. The temperature decrease is relatively rapid during the initial stages because the heat flux is

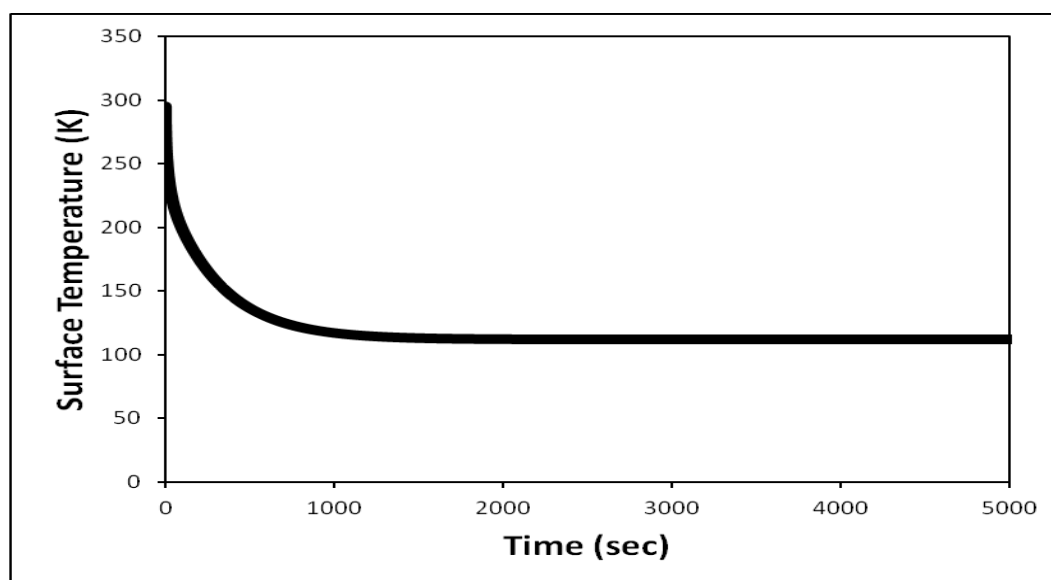


**Figure 6.6:** Temperature-time variations when LNG vaporizes above a soil substrate undergoing film boiling.

relatively high during the initial these stages. Figure 6.7 shows the corresponding predicted vaporization rate versus time for the film boiling case.

As expected, the vaporization rate decreases with time as the soil cools because the heat released by the soil decreases with time. Once the soil surface temperature approaches the boiling temperature of the LNG pool, the vaporization rate approaches zero. The black curve of Figure 6.8 shows the total

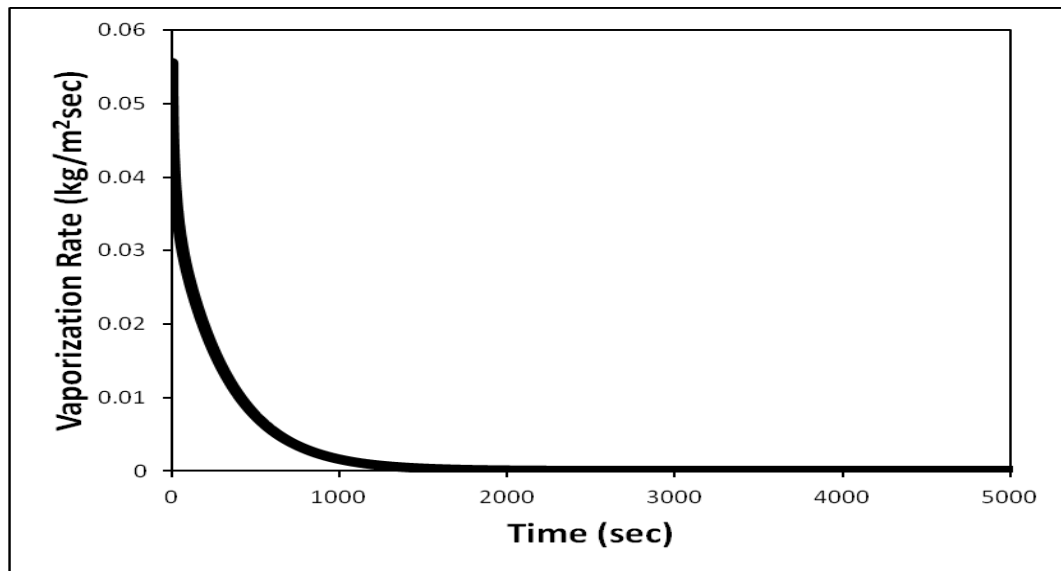


**Figure 6.7:** Vaporization rate versus time profile for an LNG pool undergoing film boiling.

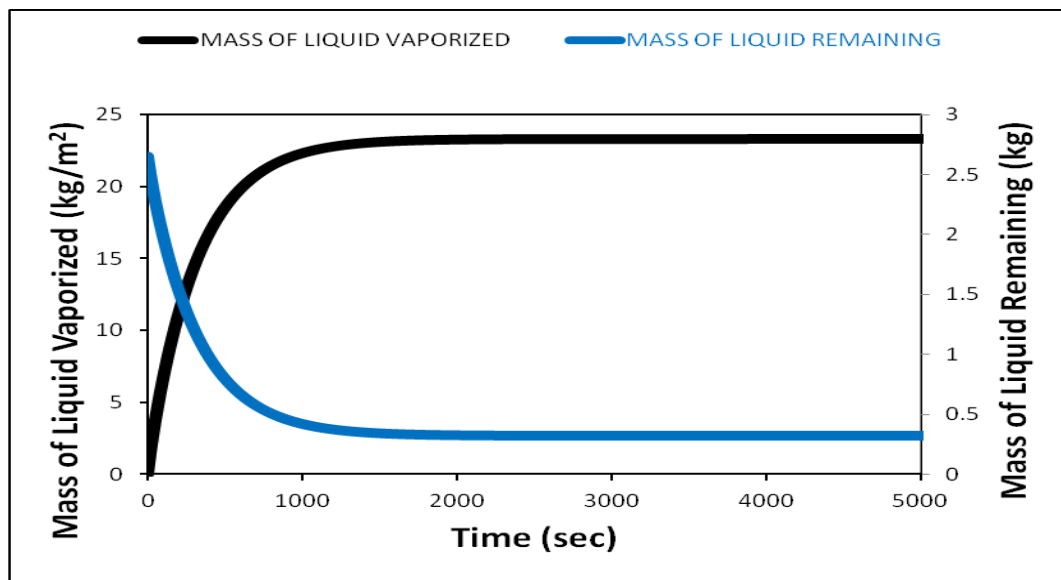


**Figure 6.8:** Profiles of the total mass of LNG vaporized per unit area versus time and the total mass of LNG remaining versus time. We add that the film boiling is an artificial constraint & that at around 1000 second, boiling stops as the temperature driving force is zero at this time.

mass of vaporized LNG as it boils on the soil surface while the blue curve of Figure 6.8 shows the corresponding total mass of the liquid remaining as a function of time. A relatively larger volume of vapour is generated at the initial stages of boiling as can be inferred from the steepness of the descending and ascending slopes of Figure 6.8. The nearly horizontal shape of the black and blue curves of Figure 6.8 signifies that the substrate has stopped cooling.

### 6.2.1 Effect of Three-Regime Boiling

Until now, we have originally assumed that the LNG pool boils only in the film boiling regime and wish to now investigate a more realistic case in which the heat transfer is allowed to transition into other regimes of boiling – the so-called three-regime boiling. For the confined spills under consideration in this chapter, a collapse of the vapour film is inevitable and will lead to a change in the boiling regime from film boiling to transition boiling, and ultimately to nucleate boiling. Figure 6.9 shows a comparison of film boiling and three-regime boiling in terms of cooling rate. Initially, the trends in temperature profiles are similar because they are both limited by a vapour film. Following the onset of transition boiling, the similarity breaks down as the blue curve deviates from the black one. A quick drop to the LNG temperature is further observed following a switch to nucleate boiling, where a perfect direct contact between the soil and LNG is immediately in place.

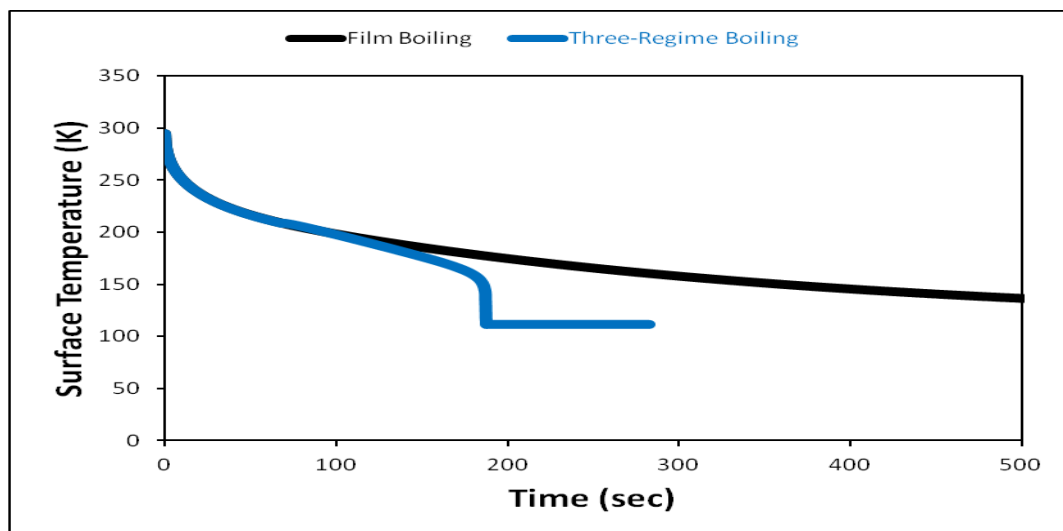


**Figure 6.9:** Comparison of cooling rate between a pool of LNG undergoing film boiling and three-regime boiling, respectively.

The difference between film and three-regime boiling can be illustrated in terms of their heat transfer coefficient profiles (see Figure 6.10). Following a shift to transition boiling, a rapid rise in heat transfer coefficient is observed with respect to the three-regime boiling case, implying that greater amount of heat is being released by the cooled soil during the transition boiling duration. The convective heat transfer coefficient is zero at the onset of nucleate boiling because of the absence of a vapour film between the liquid and solid. Following the onset of nucleate boiling regime, heat transfer is controlled by the thermal properties of the substrate.

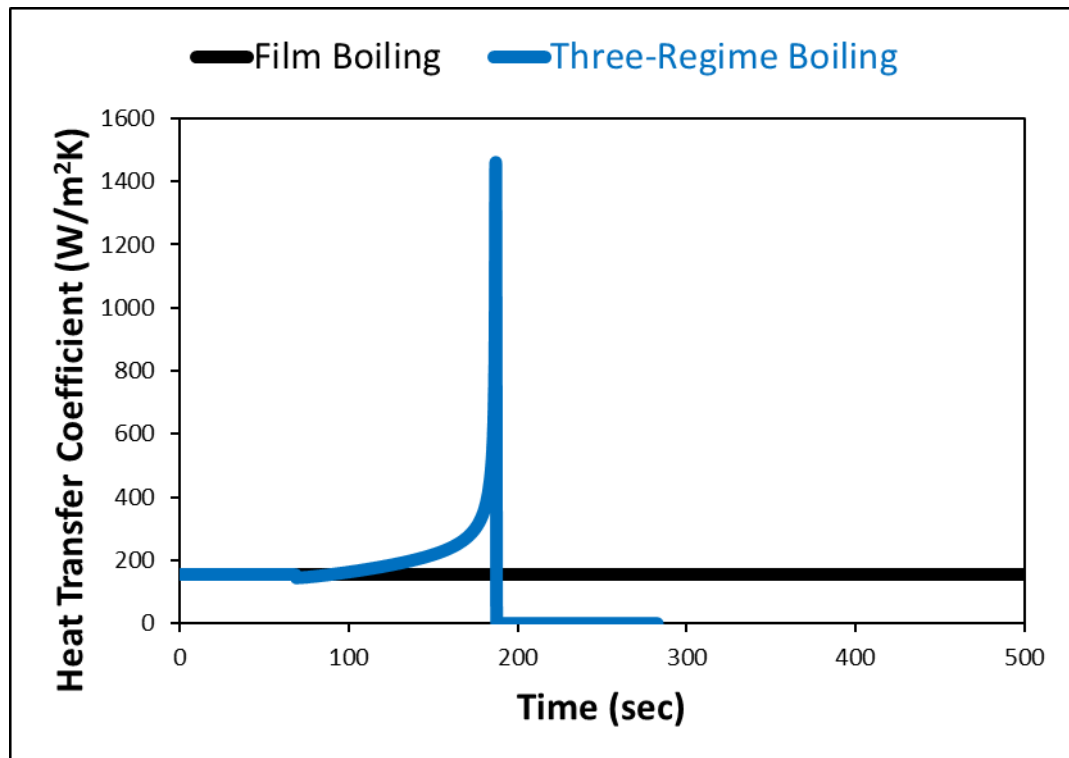


**Figure 6.10:** Comparison of convective heat transfer coefficient between a pool of LNG undergoing film and three-regime boiling, respectively.

A comparison of the vaporization rate profiles of film and three-regime boiling is presented in Figure 6.11. As already indicated above, the divergence of the blue and black curves of Figure 6.11 is clearly evident during transition boiling when the heat transfer coefficient between the pool and the soil is increasing due to the increasing LNG-soil contact duration. Vaporization rate increased from approximately 0.027 kg/m²/sec to a maximum transition boiling value of 0.082 kg/m²/sec during transition boiling. At the onset of nucleate boiling, the

vaporization rate first peaked to a value of  $1.09 \text{ kg/m}^2/\text{sec}$  before rapidly decreasing to the minimum value of  $0.042 \text{ kg/m}^2/\text{sec}$ , at which point vaporization of the pool for the three-regime boiling case has been completed. The high vaporization rate observed is mainly responsible for the much shorter vaporization times observed during three-regime boiling (see blue curve of Figure 6.11).

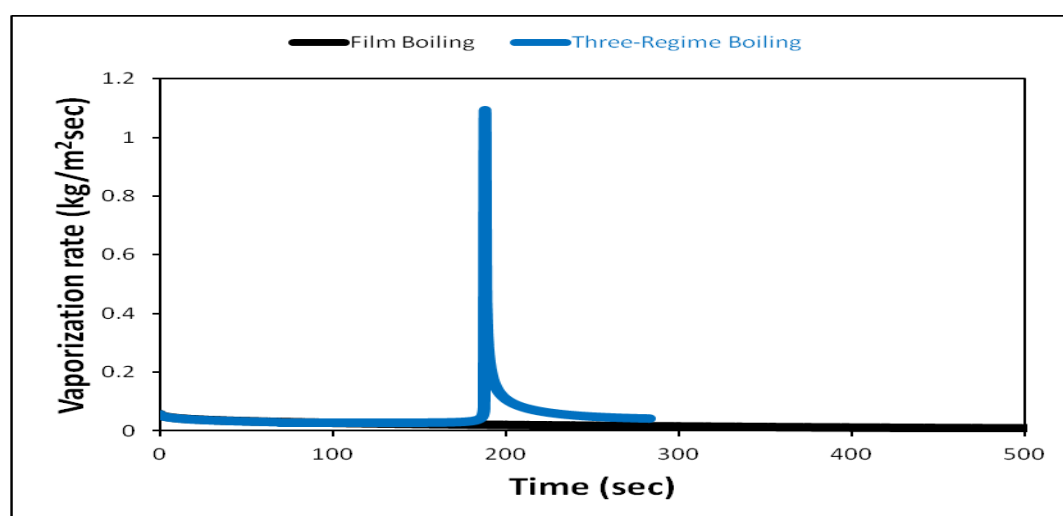


**Figure 6.11:** Comparison of vaporization rate profiles between a pool of LNG undergoing film and three-regime boiling, respectively.

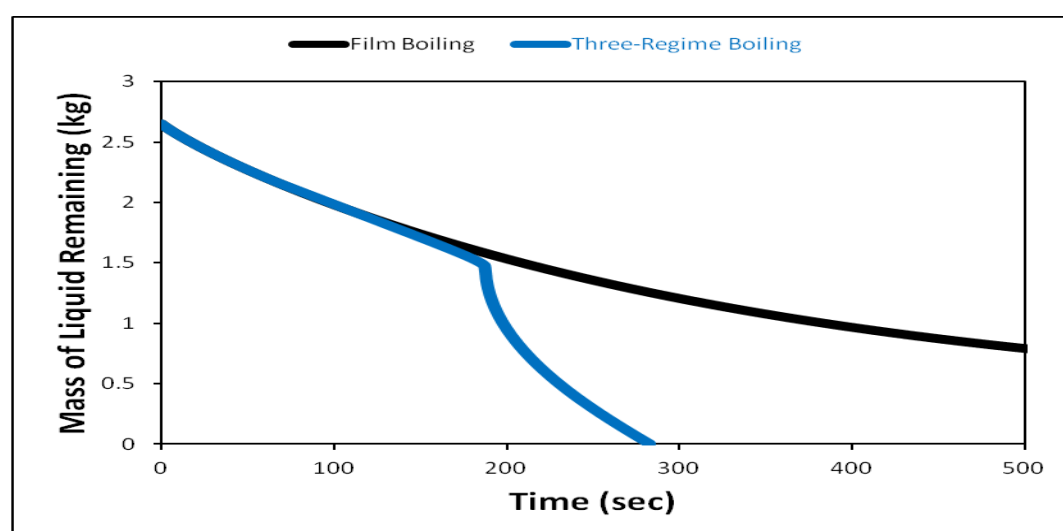


**Figure 6.12:** Comparison of mass of LNG remaining versus time between a pool of LNG undergoing film and three-regime boiling, respectively.

Figure 6.12 shows the corresponding profiles for mass of LNG remaining versus time for film and three-regime boiling. The figure reveals that the difference

between the rate of mass loss during film and three-regime boiling is significant after the onset of transition boiling and a more realistic prediction of vaporization rate and/or mass loss should require the incorporation of the effect of three-regime boiling in the model.

### 6.2.2 Effects of Initial Spilled Mass, Area of Confinement and Initial Substrate Temperature

The effects of initial spilled mass, area of confinement and initial substrate temperature on pool behaviour are presented in Table 6.7. It is seen that the results differ in how long the cooling and/or vaporization lasted. Table 6.7 shows that the maximum time for complete vaporization increases with the initial spilled mass. Furthermore, the cooling rate of the soil substrate is strongly influenced by the size of the enclosure holding the liquid pool (i.e. the area of confinement). A larger area of confinement implies that more surfaces for heat exchange is available. From Table 6.7, it is clear that increasing the area of confinement from 0.05 m<sup>2</sup> to 0.1 m<sup>2</sup> leads to a decrease in the maximum time required for the liquid to completely vaporize. It takes about 1320 seconds to completely vaporize a fixed amount of LNG spilled onto a confined area of 0.05 m<sup>2</sup> while a lesser time of 627 seconds would be required if the same amount is spilled onto a confined area of 0.1 m<sup>2</sup>. The time for complete vaporization further decreased to a value of 283 seconds when the area of confinement was increased

**Table 6.7** Values of some parameters varied during liquid pool simulations

| <i>Parameters</i>                          | <i>Time for Complete Vaporisation (<math>t_{max}</math>) [sec]</i> |
|--------------------------------------------|--------------------------------------------------------------------|
| <b>Initial Spilled Mass [kg]</b>           |                                                                    |
| Mass=1.33                                  | 188                                                                |
| Mass=2.65                                  | 283                                                                |
| Mass=5.31                                  | 627                                                                |
| <b>Area of Confinement [m<sup>2</sup>]</b> |                                                                    |
| Area=0.05                                  | 1322                                                               |
| Area=0.10                                  | 627                                                                |
| Area=0.20                                  | 283                                                                |
| <b>Initial Surface Temperature [K]</b>     |                                                                    |
| Temperature=255                            | 343                                                                |
| Temperature=275                            | 305                                                                |
| Temperature=295                            | 283                                                                |

to 0.2 m<sup>2</sup>. The influence of initial substrate temperature on substrate cooling rate is also illustrated in Table 6.7. It is seen that increasing the initial temperature of the soil decreases the time for complete vaporization of the LNG.

### 6.2.3 Effect of Convective Heat Transfer

We have stated earlier that the rate of vaporization of a liquid pool on a solid substrate is mainly driven by heat transfer from the substrate. Figure 6.13 examines the influence of convective heat transfer (from the surrounding air) on vaporization rate. The convective heat transfer coefficient was calculated using Equation A.2 of Hissong (2007). As can be seen from the curves in Figure 6.13, the influence of the convective heat transfer is relatively small suggesting that the most dominant heat to the pool is from the substrate. This has long been proven for a number of cryogenic liquids including LNG. By including the convective heat flux to the pool, the time for complete vaporization will decrease by just 18 percent which is relatively small in magnitude compared to the heat contribution from soil. For the purpose of this work, convective heat transfer from surrounding air and/or nitrogen is neglected.

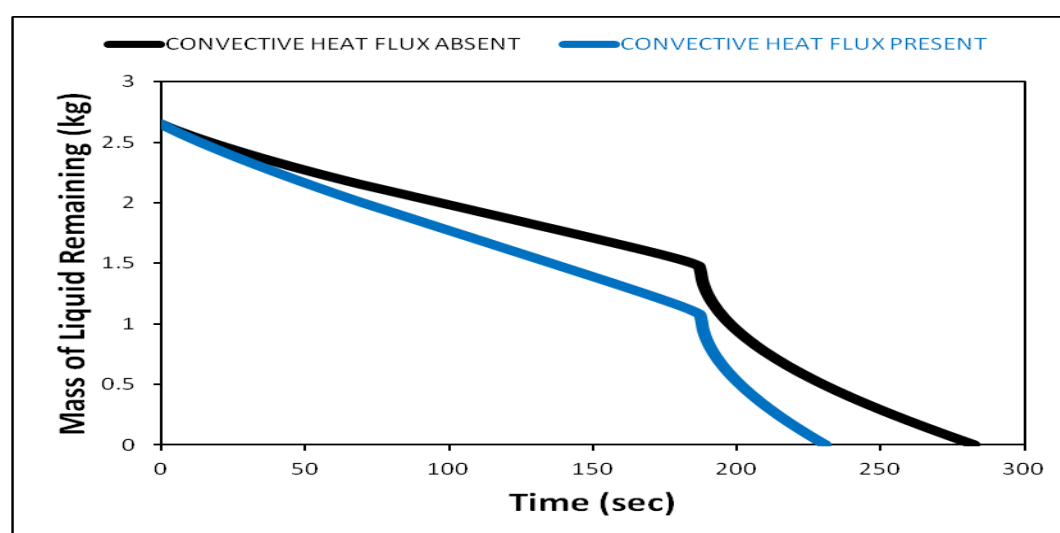


**Figure 6.13:** Profiles showing the mass of liquid remaining versus time with and without convective heat transfer contributions, respectively.

### 6.2.4 Effect of Substrate Type

In order to analyse how substrate type will affect the pool behaviour, we considered the effect of spilling LNG on a different type of substrate other than soil, in this case, stainless steel with thermal properties markedly different from

that of soil. Figures 6.14 and 6.15 show the effect of substrate type on cooling rate and mass loss, respectively. While it took about 190 seconds for the soil surface temperature to decrease close to LNG temperature (see Figure 6.14), a relatively longer time of 1200 seconds would be required for the stainless steel surface to cool close to LNG temperature. In comparison with the soil case, this is a factor of 7 increase. The corresponding mass loss profiles is shown in Figure 6.15, where it is indicated that the time for complete vaporization is approximately similar for the two substrate cases. The mass loss for the stainless steel substrate is less rapid when compared to the curves obtained for the soil

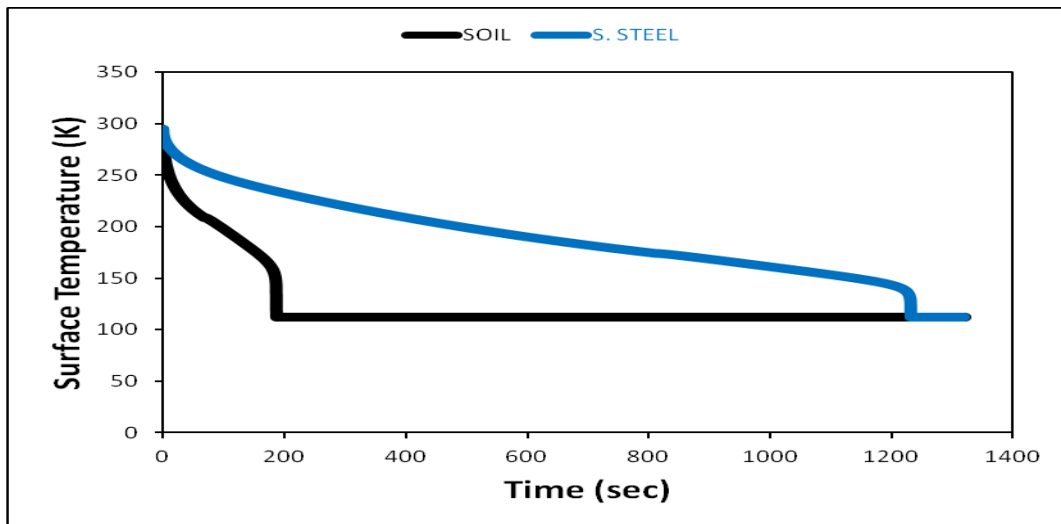


**Figure 6.14:** Comparison of predicted temperature-time history for LNG spill on two different types of substrates.

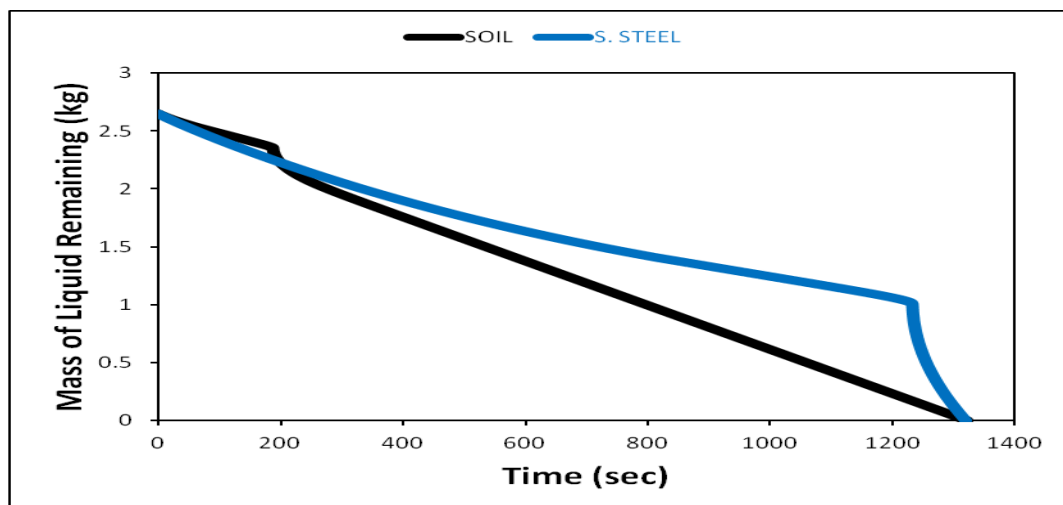


**Figure 6.15:** Comparison of predicted mass of liquid remaining versus time for LNG spill on two different types of substrates.

case because of the prolonged film boiling duration for the stainless steel case. Though the stainless steel substrate experienced both transition and nucleate boiling at the last stages of the boiling process, this did not drastically alter the relatively close timescale for complete vaporization observed for the two substrate cases. The results shown in Figures 6.14 and 6.15 suggest that the effect of substrate type and causes the major effect observed (e.g. rapid cooling rate and mass loss of the soil substrate, different duration of the respective boiling regimes) in these figures.

## 6.3 Discussion and Conclusions

A model to predict the vaporization of liquid spills was presented in this chapter. The model can predict a time dependent spreading rate, heat transfer rate and vaporization rate. The model allows for instantaneous spills from various containment systems as well as seepages from a porous medium. The model is solved numerically using a Fourth-Order Runge-Kutta method. The convergence of the numerical method was successfully tested and an optimum grid size was selected and used for the simulation of the sample problems presented in this chapter. Successful model verification was carried out by comparing results obtained from the liquid pool model with those from their derived analytical solution, with excellent agreement. The model was further validated using experimental data reported by Reid (1980) for instantaneous spills of LNG on a variety of substrates. Model predictions were in very good agreement with Reid's results.

To illustrate the model's predictive capability for example spill scenarios, simulations were performed for an instantaneous spill of LNG on a soil substrate. A film boiling limiting case was first studied assuming that the vaporization of the liquid is solely controlled by film boiling. The model was extended to account for the influence of three-regime boiling and a comparative study between these two boiling conditions conducted. Other relevant parametric trends such as initial spilled mass, initial surface temperature, area of confinement, substrate type and convective heat transfer effects were investigated in order to understand how their variations can influence the

spreading and vaporization of the liquid pool. Some of the specific conclusions from this chapter are:

- The spreading and vaporization of liquid pools on solid substrates can be predicted by an equation of the form suggested by Vesovic (2007) for spills on water, but with the requirement that suitable allowance must be made for incorporating the thermal properties of the solid substrates under consideration in this thesis. In addition, the relative density term in Vesovic's equation (Vesovic, 2007) must be replaced with an appropriate gravity term to reflect the change in surface properties.
  - Simulation results from the refined liquid pool model for predicting the vaporization of LNG on solid surfaces agree well with the available experimental data and a derived analytical solution.
  - Assuming the heat transfer mode between a confined pool of LNG and a soil substrate is film boiling, the vaporization rate of the liquid pool will be limited by the presence of a stable vapour film between the soil surface and the vaporizing LNG pool, leading to decreasing vaporization and heat transfer rates as a function of time. Relaxing the film boiling assumption to accommodate three-regime boiling, the decreasing surface temperature of soil leads to a change in the boiling regime, first to transition boiling, and ultimately to nucleate boiling. Under three-regime boiling, LNG vaporizes at a relatively higher rate which is several times the magnitude predicted by a boiling LNG pool undergoing film boiling. The vaporization times significantly decreases when the effect of three-regime boiling is incorporated in the model.
  - Results from the confined pool boiling simulations confirm that initial temperature, area of confinement and initial spilled mass can influence the vaporisation rate. The results suggest that these factors
-

affect the rate of vaporisation and the amount of time it will take a given volume of spilled liquid to completely vaporize.

- An important effect of substrate type on LNG pool behaviour has been confirmed for soil and stainless steel substrates. The stainless steel substrate cools less rapidly when compared to soil due to its prolonged film boiling duration. This prolonged film boiling duration occurred due to the relatively higher thermal conductivity of stainless steel, which causes more heat to be conducted to the surface, leading to a less rapid drop in surface temperature.
- The effect of including convective heat flux was investigated and it was concluded that, from heat transfer considerations, adding this heat source in the implemented pool model does not significantly alter the heat transfer and vaporization rates of the LNG pool.

# 7

## ***Applications***

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There are real systems where the use of the computer models presented in this thesis can be applied. Example applications include:

- a) LNG flow through flat joints of a GTT Mark III-Type LNG ship,
- b) LNG pool on inner hull of a GTT Mark III-Type LNG ship,
- c) Water flow through superheated geologic formations,
- d) LNG and other chemical penetration into soil,

The first three applications are studied in this chapter. The first two application cases are discussed next.

Figure 7.1 is a schematic diagram of a membrane tank insulation layer showing LNG flow through the flat joint of a GTT Mark III type LNG ship under primary and secondary barrier leakages. The flow is assumed to be pressure-driven with a mass flowrate determined from Darcy's law (Bear, 1988; Darcy, 1856). Seepage through the flat joint would result in LNG filling up the insulation space (IS) or void area (VA) as well as forming a pool on the inner hull substrate. Figure 7.2 depicts the heat transfer situations between the formed LNG pool and the inner hull of a GTT Mark III-Type LNG ship. Key heat transfer contributions are from inner hull, from nitrogen and from mastic. While nitrogen is used as inert gas, thermal insulator, methane detector and positive pressure regulator in insulation spaces of membrane LNG ships (SIGTTO, 2007), mastic provides support for the insulation layer. Figure 7.3 provides a slightly different description of Figure 7.1 with the location of mastic and other insulation layer features clearly shown.

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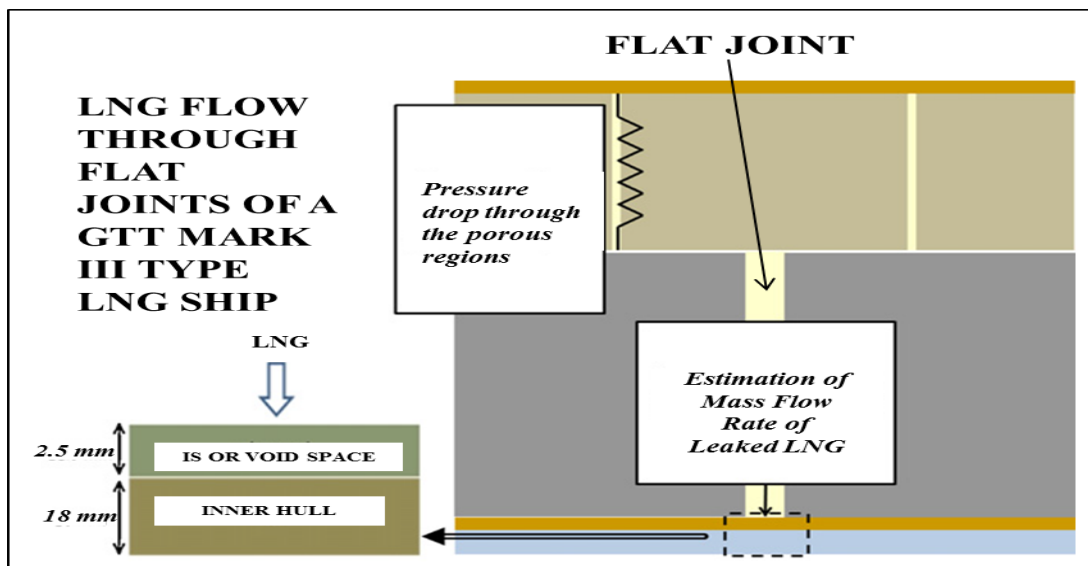


**Figure 7.1:** A schematic diagram of the Insulation layer of a membrane tank LNG ship showing a pressure driven LNG flow through a flat joint and onto an inner hull of the ship (Choi et al., 2012).

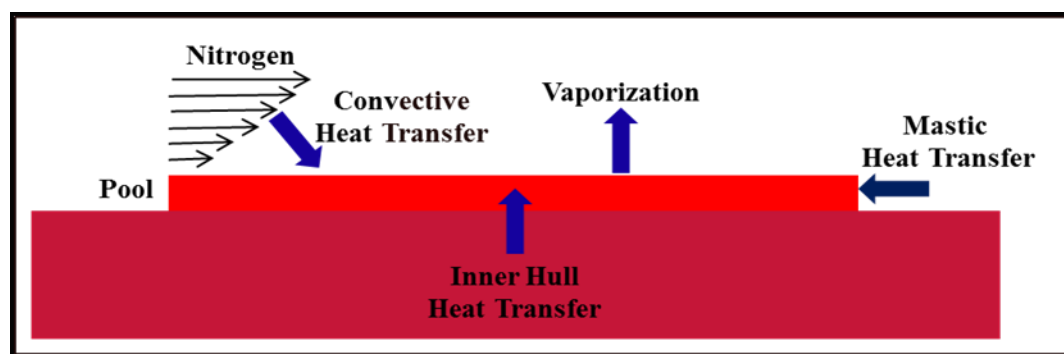


**Figure 7.2:** A schematic diagram showing LNG pool and mass transfer mechanisms in the event of seepage onto inner hull of a typical membrane tank LNG ship.

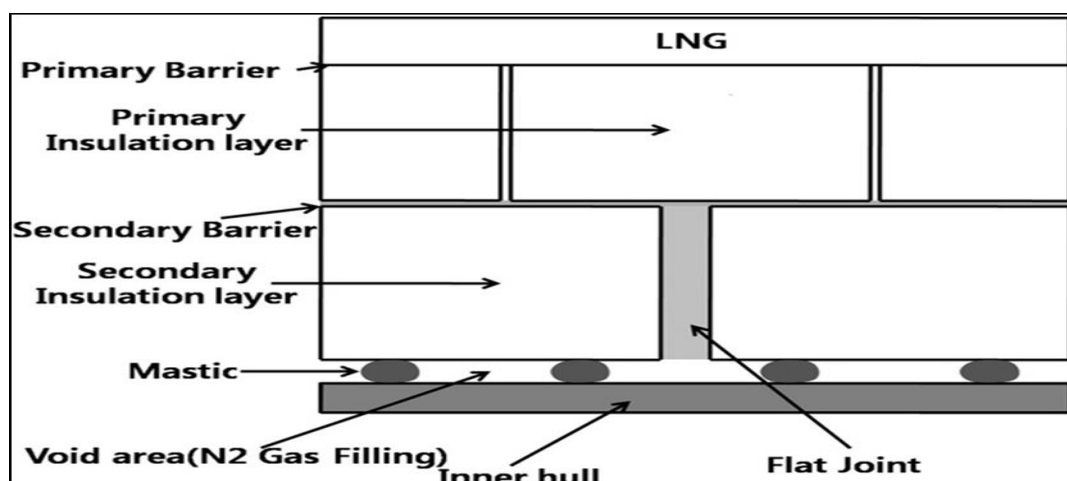


**Figure 7.3:** A schematic diagram of the insulation layer of a membrane tank LNG ship showing the key features of the insulation system (Lee et al., 2011).

A third application of the proposed model is in radioactive waste isolation where there is a probability of water seepage and subsequent corrosion of radioactive waste emplacement tunnels (Birkholzer, 2003). The bottom diagram of Figure 7.4 shows a mountain scale diagram of a radioactive waste emplacement site while the top diagram of Figure 7.4 shows a magnified view of the conditions near the radioactive waste packages. The high heat content of these waste packages naturally leads to the vaporization of the pore water within the surrounding geological formation, thus bringing them to a superheated state. Despite the capability of the superheated rock to vaporize any penetrating water, there is still concern that water could flow down the superheated rock, contact and corrode the radioactive waste emplacement tunnels, and seep into the waste packages before reaching accessible environment via groundwater (Birkholzer, 2003). It is therefore useful to predict how vaporization of the infiltrating water could affect water penetration rates.

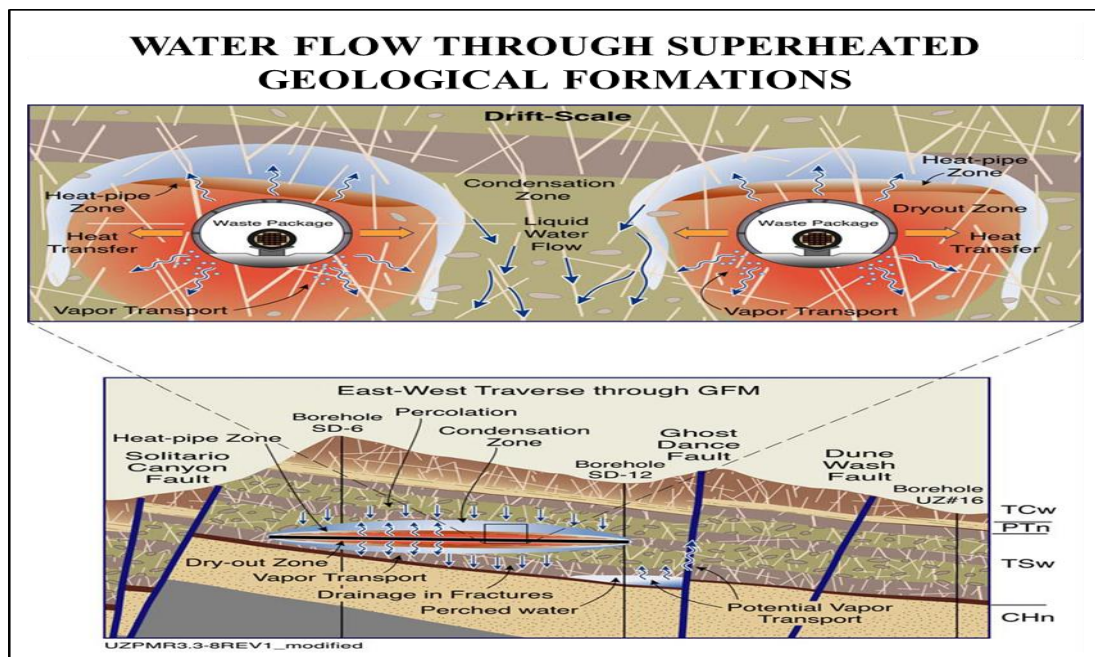


**Figure 7.4:** A diagram of processes around a radioactive waste emplacement tunnels

The fourth application is the prediction of LNG and other chemical penetration into soil. For conditions where the thermophysical properties of the soil allows very deep penetration of the liquid, there is a potential for contamination of groundwater, most especially, for reactive liquids such as ammonia or chlorine.

Figure 7.5 shows a spill scenario involving LNG and other chemical penetration into soil and onto water table. If the liquid penetrates deep into the substrate, the probability of reaching the water table is high. This fourth application case would not be specifically analysed in this work and was only mentioned to provide a sense of what has been accomplished (in terms of usefulness and versatility of the models presented in this thesis). The analyses of the three application cases are also aimed to illustrate the impact of heat transfer and vaporization rate on liquid penetration rates.

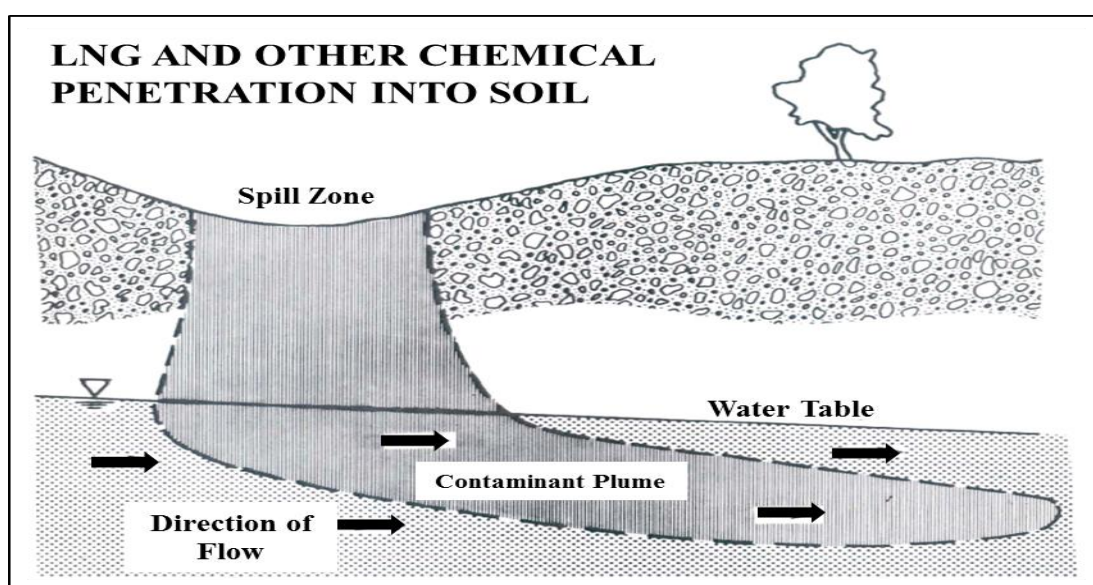


**Figure 7.5:** A diagram of LNG and other chemical penetration into soil and water table (Melhem, 2006).

This chapter is split into five main sections. In Section 7.0, we discussed some features of the type of ships used in LNG transportation. We presented in Section 7.1, an analysis of vaporizing LNG flow through flat joints of a GTT MARK III-Type LNG ships under secondary barrier leakages. In Section 7.2, a study of the fate of an LNG pool on inner hull of a GTT MARK III-Type LNG ship was evaluated. The third application case presented in Section 7.3 covered the study of an analogous process dealing with vaporizing hot water flow through superheated geologic formations above radioactive waste emplacement tunnels. The fifth and final section, Section 7.4, discussed and drew conclusions from chapter 7.

## 7.0 Features of LNG Ships

The ships used for the transportation of LNG can be differentiated by the type of cargo containment system (CCS) or cargo tanks used to store the cryogen. Two main types of cargo containment systems are currently available: the standard spherical (moss) tank designs and the membrane type designs (Harris, 1993; Woodward and Pitblado, 2010). A third design - the so-called prismatic type B (SPD) tank design (Woodward and Pitblado, 2010) are also in use. In Figure 7.6,



**Figure 7.6:** Features of a typical self-supporting spherical moss tanks (above) and the membrane type (below) (Vanem, 2008).

the outside features of the spherical and membrane type designs are depicted. The spherical moss tanks of the spherical design, shown as the top picture of Figure 7.6, are clearly visible to the eye. The membrane tanks of the membrane design (see bottom picture of Figure 7.6) are not visible because they are constructed to fit the inner hull structure of the LNG ship. Figures 7.7 and 7.8

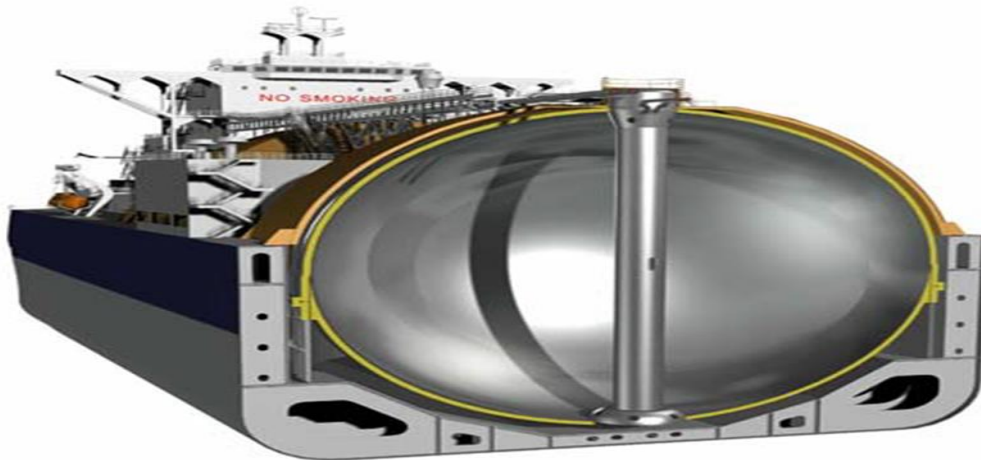


**Figure 7.7:** An illustration showing the inside features of a typical spherical (moss) tank LNG cargo containment system (GTT Website Illustrations).



**Figure 7.8:** An illustration showing the inside features of a typical membrane tank LNG cargo containment system (GTT Website Illustrations).

show the inside features of a typical spherical (moss) and membrane tank LNG ships, respectively. A common characteristics of both ship designs comes in form of their conformity to important structural requirements that is aimed to protect the ships from damage. Table 7.1 and 7.2 list typical dimensions of the spherical and membrane type LNG ships respectively. There are currently new large developments in LNG cargo capacity and dimensions – the so-called Q-Flex and Q-Max membrane systems with capacities of between 210,000-250,000 m<sup>3</sup> (Woodward and Pitblado, 2010).

**Table 7.1** Typical dimensions of spherical and membrane type LNG ships (Woodward and Pitblado, 2010).

| LNG Ship Type                    | Spherical Moss Type | Membrane Type |
|----------------------------------|---------------------|---------------|
| Cargo Capacity (m <sup>3</sup> ) | 138-173,000         | 125,000       |
| Cargo Tanks                      | 4                   | 5             |
| Overall Length (m)               | 277-290             | 293           |
| Breadth or Width (m)             | 43.3-45.8           | 41.6          |
| Draft (m)                        | 11.5                | 11.7          |

The spherical and membrane tank systems are constructed using steel and steel alloys and these steel materials are known to satisfy important structural requirements. However, the structural requirements do not necessarily prevent the brittle fracture of some of the ship's steel structures upon direct contact with the extremely cold LNG. Although the special steel materials used in

constructing the primary membrane (or primary barrier) harden upon contact with the cargo and have sufficient strength to resist embrittlement of the tank material (Woodward and Pitblado, 2010). However, it has been proven that LNG ship's susceptibility to brittle fracture is more pronounced with the membrane type design (Bae, et al., 2007; Lee, et al., 2011; Kalan and Petti, 2010; Choi, et al., 2012). Three types of membrane system are available which are the NO 96, the Mark III and the CS1 systems (see Figure 7.9). In all membrane type LNG ships, the cargo is isolated from the inner hull by layers of insulation (Wilson, 1974; Harris, 1993) usually consisting of two barriers or membranes that ensure cargo tightness. A secondary membrane prevents the cargo from reaching the inner hull of the LNG ship in the event of a primary barrier leak. Generally, the insulation layers maintain acceptable temperature for the inner hull as well as minimize the heat transferred to the cargo from the environment. A more detailed description of each of these three insulation features are provided below.

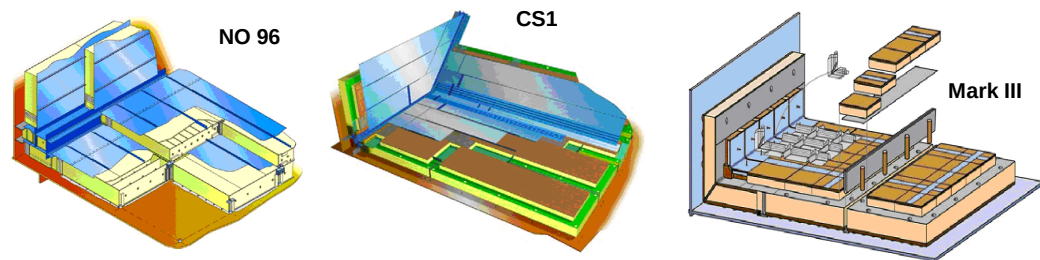


**Figure 7.9:** An illustration showing the features of the three major membrane systems: the NO 96; the CS1 and the Mark III membrane systems (GTT Website Illustrations).

#### The NO 96 Membrane LNG Cargo Containment System

The picture in the extreme left of Figure 7.9 above shows the features of the NO 96 membrane cargo containment system (Deybach, 2003) and Figure 7.10 shows a much larger view of the NO 96 Insulation System. This system is made of two identical metallic invar membranes or barriers (see the Invar Membrane Strake in Figure 7.10). The membrane in contact with the cargo is called the primary membrane or primary barrier. The secondary membrane or secondary barrier which separates the primary and secondary insulation boxes (or layers) ensures a 100 percent containment in case of LNG leakage through the primary membrane. Both the primary and secondary membranes are made of invar

which is a 36% nickel-steel alloy because such steel alloys decrease the tank's susceptibility to brittle fracture and leakage. The primary and secondary insulation layers are made of prefabricated plywood boxes filled with expanded perlite. The standard size of the boxes is 1 m by 1.2 m. The thickness of the insulation box is adjustable from 170 mm to 250 mm, to match any Boil-Off-Ratio (B.O.R) requirement. The typical thickness of the secondary insulation box is 300 mm. The primary insulation layer is secured by primary couplers, themselves fixed to the secondary coupler assembly. The secondary insulation box is laid and evenly supported by the double hull through load-bearing resin ropes, and fixed by means of the secondary couplers anchored to the inner hull.

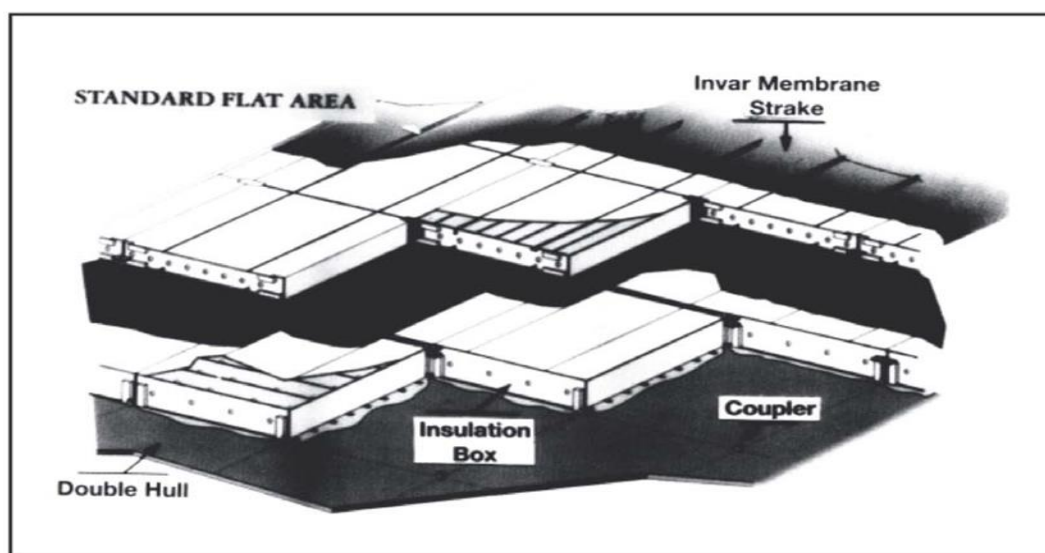


**Figure 7.10:** Insulation design features of NO 90 type LNG ships (International Safety Guide, 2010).

### The Mark III Membrane LNG Cargo Containment System

The Mark III system shown in the extreme right of Figure 7.9 and Figure 7.11 is composed of a primary stainless steel membrane or barrier. This primary membrane which is 1.2 mm thick and contains the LNG cargo, is positioned on top of a prefabricated insulation panel (Deybach, 2003). The primary membrane are of standard dimensions, 3 m by 1m. The secondary membrane is made of triplex. Triplex is a composite laminated material constructed of a thin sheet of aluminium between two layers of glass cloth. The secondary membrane (or secondary triplex membrane) is positioned inside the prefabricated insulation

panels between the primary and secondary insulation layers. Both the primary and secondary insulation layers consist of polyurethane (Demharter, 1998) foam panels (see Figure 7.11). There are spaces within each insulation layer filled with nitrogen gas that serves as a thermal insulator and helps in the detection of LNG leakages. Other materials of construction include plywood and top bridge pads. Between the secondary triplex membrane and the ship's inner hull, we can find the following materials: plywood, resin ropes, mastic, flat joint, plugs and reinforced polyurethane foam panels. The Mark III system are most susceptible to barrier damage, therefore, our study will focus on evaluating barrier leakages in Mark III type LNG ships after providing a brief description of the third membrane system – the so-called CS1 system.

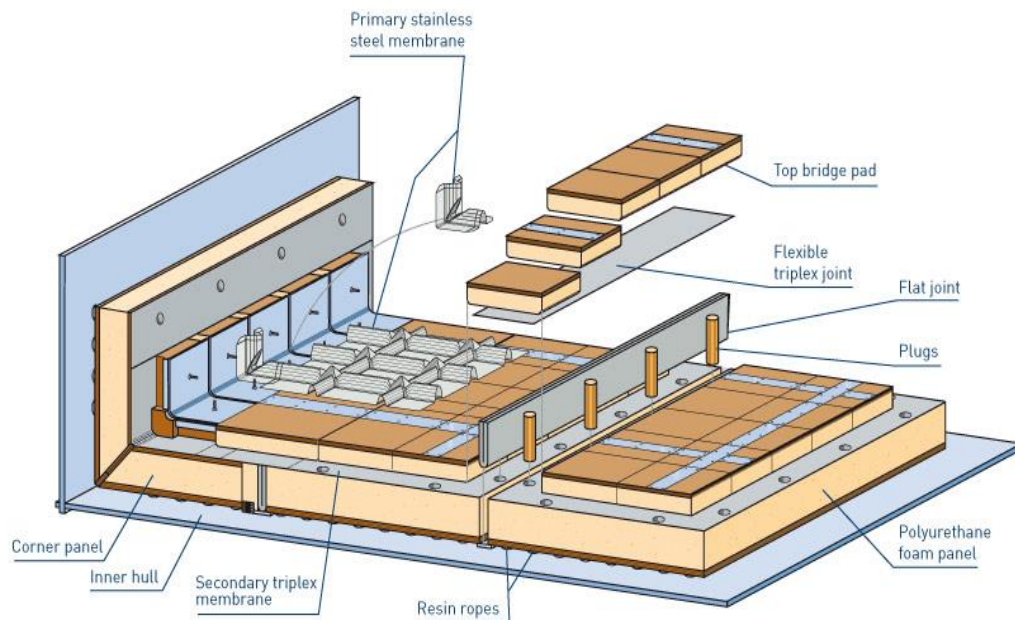


**Figure 7.11:** Insulation design features of Mark III type LNG ships (GTT Website Illustrations).

#### The CS1 membrane LNG Cargo Containment System

The CS1 system which is the middle picture of Figure 7.3 combines the features of both the NO 96 system and the Mark III system (Deybach, 2003). This design is composed of a primary membrane positioned on top of a prefabricated insulation panel including a complete secondary membrane. Like the NO 96 system, the primary membrane is made of invar, a 36 percent nickel-steel alloy, 0.7 mm thick, and contains the LNG cargo. Like the Mark III system, the

secondary membrane is made of triplex and is positioned inside the prefabricated insulation panels between the two insulation layers. The panels, which have the same standard dimensions as the Mark III system are bonded to the inner hull by means of resin ropes that anchor and spread evenly the loads.

## **7.1 Application Case No. 1: LNG flow through flat joints of a GTT Mark III-Type LNG ship**

The membrane-type LNG ships is now widely used in the LNG shipping industry (Harris, 1993; Deybach, 2003; Woodward and Pitblado, 2010). Increasing orders for this type of ship design can be linked to their robustness, flexibility, large capacity relative to other ship designs, and lower fuel consumption (Bae et al, 2007). However, we have already indicated that there are safety concerns that membrane-type LNG ships are most susceptible to barrier damage if incidents such as, for examples, grounding, unexpected collisions, sloshing impact damage and terrorist attack should occur (DOE, 2012). The hazards associated with such incidents include but not limited to: cryogenic damage to the steel structures of the ship (Kalan and Petti, 2010), cascading failure of the ship (Petti and Lopez, 2009), and fire and explosion (Woodward and Pitblado, 2010). By cascading failure, we mean the sequential failure of multiple tanks of an LNG ship (Kalan and Petti, 2010; Petti and Lopez, 2009; DOE, 2012).

A number of workers have assessed the potential failure of membrane type LNG insulation system under impact loads (Kim et al., 2011; Chun, et al., 2009; Shin, et al., 2003; Kim, et al., 2008; Noh, et al., 2009; Han, et al., 2009, Bergan, et al., 2009) but only a handful of workers assessed the failure of the membrane system under barrier leakages (Bae, et al., 2007; Lee, 2011, Choi, 2012). One way of addressing the problem of cargo tank failure is to model the evolution of LNG flowing within porous compartment areas of a ship following barrier leakages. Despite its importance, this problem has not been sufficiently addressed by researchers. Specifically, the problem of vaporizing LNG flow following secondary barrier leakages has not been adequately addressed. As a form of contribution, we have analysed, in this section, the safety of a GTT Mark III-Type LNG ship

under secondary barrier leakages by using the computer model developed in this work.

The problem geometry shown in Figure 7.12 and as suggested by Bae et al. (2007) was used for this analysis. We considered a worst case barrier leakage scenario in which the damage to the primary barrier occurred at the bottom of the cargo containment system. We assumed that the LNG tank is in fully-filled condition (approximately 98.5 %) with a  $12,500 \text{ m}^3$  (12,000,000 kg) configuration. We also assumed that the diameter of the hole formed at the rupture point is 4 mm. Figure 7.12 provides further details on the dimensions and geometry of

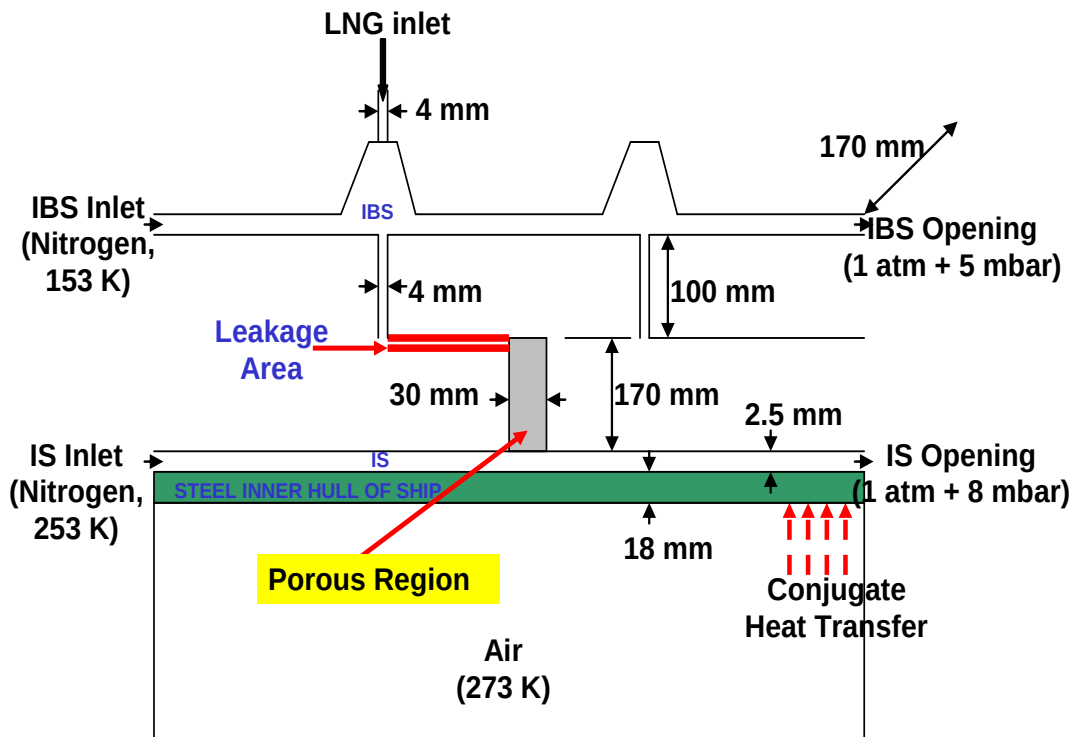


**Figure 7.12:** A schematic showing a worst case barrier leakage scenario and boundary conditions as used for the results analysis of application case no. 1 (Bae et al., 2007).

barrier leakage problem. The vertical length between the primary barrier rupture point and the surface of the secondary triplex membrane (or secondary triplex barrier) is 113.5 mm of which 100 mm is the vertical length of the primary insulation layer. The primary insulation layer has an inter-barrier space (IBS) through which nitrogen gas could flow. The vertical length of the secondary

insulation layer is 170 mm. This is essentially the maximum depth through which the leaking LNG could penetrate before reaching the inner hull. Within the secondary insulation layer, we can identify a porous region, of width 30 mm. This porous region is the flat joint made of glass wool porous material. Between the secondary insulation layer and the inner hull, a secondary insulation space (IS) or void area is observable from Figure 7.12. The secondary insulation space (IS) is a second space through which nitrogen gas could be injected to vaporize any leaked LNG. This secondary insulation space, of vertical length 2.5 mm, separates the secondary insulation layer from the steel inner hull of the ship.

In describing the barrier leakage process, we assume that a rupture of the primary membrane (or barrier) results in LNG leakage into the primary insulation layer. We also assume that both inter barrier space (IBS) and the primary insulation layer became saturated with LNG under very short duration and that the secondary membrane or barrier is also damaged or ruptured. Rupture of the secondary membrane results in LNG leakage into the flat joint (shown as the porous region in Figure 7.12). The LNG leaking into the flat joint will be subject to immediate vaporization. The rate of vaporization is dependent on flat joint properties as well as on the liquid-solid contact time and area. The problem now is to analyse how the vaporization effect could affect LNG penetration rates. Results from such analysis could provide understanding on the safety implications of LNG ship experiencing secondary barrier leakages.

#### **7.1.1 Simulation Case No. 1: Base case simulation of secondary barrier leakage with LNG flow through flat joint**

This simulation case is a secondary barrier rupture of a single GTT Mark III-Type cargo tank in fully-filled condition. A single tank in fully-filled condition containing approximately 12,500 m<sup>3</sup> of LNG has been proposed. The 2-D model discussed in Chapter 5 of this thesis is used to evaluate the simultaneous vaporization and penetration of the cryogen through the flat joint. The liquid flowrate (or leakage rate) based on a recent analysis by Choi et al. (2012) is assumed constant at 0.0002 kg/s. Table 7.2 lists the assumed conditions for both LNG flow and LNG pool (discussed in Section 7.2) simulations. The thermal properties of the flat joint are based on measured values from a GTT Mark III

membrane-type LNG ship (Bae et al., 2007; Lee et al., 2011). According to Table 7.2, the height and width of the flat joint are 0.17 m and 0.03 m, respectively, with the initial temperature of flat joint specified as 233 K. The diameter of the rupture (hole) in the secondary barrier is specified as 4 mm (Bae et al., 2007; Choi et al., 2012).

**Table 7.2** Assumed conditions for both LNG flow and LNG pool simulations.

| <i>LNG and Flat Joint Properties</i>    | <i>LNG</i>            |
|-----------------------------------------|-----------------------|
| LNG Temperature (K)                     | 112                   |
| Flat Joint Initial Temperature (K)      | 233                   |
| Inner Hull Initial Temperature (K)      | 273                   |
| Inner Hull Thermal Conductivity (W/m/K) | 15                    |
| LNG Thermal Conductivity (W/m/K)        | 0.193                 |
| LNG Density (kg/m <sup>3</sup> )        | 422                   |
| LNG Viscosity (Pas)                     | $1.18 \times 10^{-4}$ |
| Latent Heat of Vaporisation (KJ/kg)     | 512                   |
| LNG Specific Heat Capacity (J/kg/K)     | 3350                  |
| Surface Tension (N/m)                   | 0.014                 |
| Height of Flat Joint (m)                | 0.17                  |
| Width of Flat Joint (m)                 | 0.03                  |
| Secondary Barrier Hole Diameter (m)     | 0.004                 |
| Approximated Radius of Inner Hull (m)   | 16.4                  |

The blue curve of Figure 7.13 displays the motion of the liquid front versus time profile obtained when LNG flow is affected by heat. The black curve shows the corresponding profile when heat flux is equal to zero ( $q=0$ ). The horizontal line in Figure 7.6 indicates the threshold depth above which LNG seepage has occurred. It is clear from these results that liquid front in the absence of heat (i.e.  $u=0.0039$  m/s) reached the bottom of the flat joint faster than the liquid front affected by heat conduction. The blue curve clearly deviated from the black curve because of the vaporization effect that reduced liquid penetration rates. It is also clear from the results that the deeper the liquid front penetrates, the stronger this vaporization effect. According to Figure 7.13, it took approximately 65 seconds for the LNG to reach the bottom of the flat joint when the heat transfer and vaporization effects are applied (blue curve). A relatively smaller timescale of 45 seconds elapsed before the liquid front, in the absence of heat transfer and vaporization, reached the bottom of the flat joint.

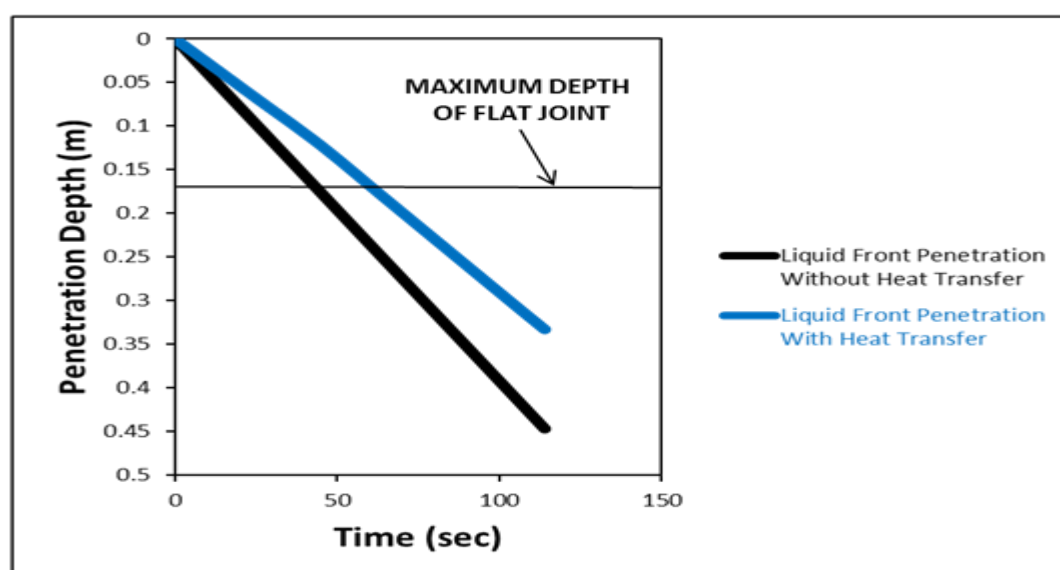


**Figure 7.13:** Penetration depth versus time curve for LNG flowing through a flat joint constructed of glass wool porous material (simulation case no. 1).

If we assume that the flat joint extends farther down the secondary insulation layer such that the bottom of the flat joint is at a depth of 0.5 m, then under the same thermal conditions as assumed above, both the vaporizing and non-vaporizing liquid fronts would reach penetration depths of 0.3 m and 0.45 m, respectively, under 114 seconds. The above results reveal that the depth reached by the liquid front following secondary barrier leakage would be relatively shallower when the flowing cryogen is affected by heat. Seepage duration for vaporizing LNG is about 31 percent longer than the seepage duration for the non-vaporizing case under similar flow conditions. This is expected because the liquid front velocity is much slower for the vaporizing flow than for the non-vaporizing case. Thus, even though total isolation of LNG from the inner hull may not be achievable at increasing times, the vaporization effect is, evidently, a factor that increases the LNG seepage duration. Seepage would certainly increase the potential for cryogenic damage of the inner hull (Bae, et al., 2007).

### 7.1.2 Simulation Case No. 2: Simulation of secondary barrier leakage considering two different values of leakage rates

The simulation conditions of Section 7.1.1 was repeated in this section. However, the secondary barrier leakage rate was decreased from its base case value of 0.0002 kg/s to 0.00011 kg/s (with the objective to assess its sensitivity to liquid

penetration rates). Figure 7.14 shows the liquid front penetration depth as a function of time at two different leakage rates. We also included the liquid front

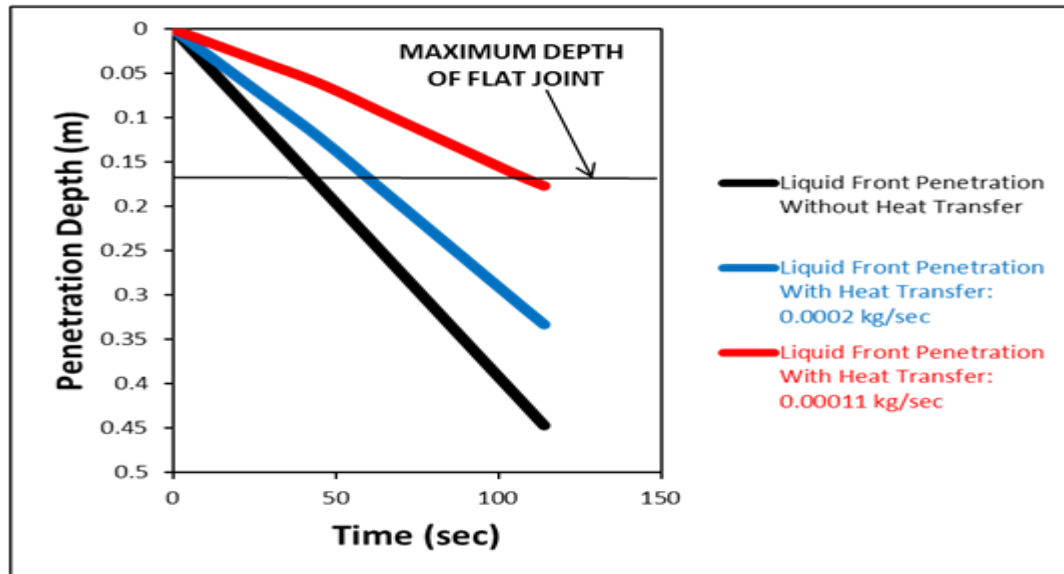


**Figure 7.14:** Penetration depth versus time curves for LNG flowing through a flat joint constructed of glass wool porous material with consideration of the effect of different leakage rates (application case no. 2).

penetration depth profile in the absence of heat for comparative purposes. The results indicate that decreasing the leakage rate decreases the depth reached by the liquid front. A relatively higher vaporization rate for the smaller leakage rate can be inferred because more time is available for heat transfer with the higher heat exchange rate implying a stronger vaporization effect that would subsequently lead to the liquid front reaching depths. The time taken to reach the bottom of the flat joint for the 0.00011 kg/s leakage case is 105 seconds compared to the 65 seconds obtained for the base case leakage rate (i.e. 0.0002 kg/s). Assuming the flat joint is extended to a depth of 0.5 m, the liquid front with respect to the lower leakage rate would only reach a depth of approximately 0.18 m, which is very close to the actual vertical depth of the flat joint. These results show that the maximum depth reached by the liquid front does seem to be significantly affected by varying the value of the leakage rate. Hence, leakage rate is an important parameter that should be considered when predicting the performance of a GTT Mark III-type LNG ship under barrier leakages.

### 7.1.3 Simulation Case No. 3: Simulation of secondary barrier leakage considering two different values of rupture (hole) diameters

We varied the rupture (hole) diameter to test its sensitivity to liquid penetration rates. As can be observed from comparing the red curve with the blue curve (base case profile) of Figure 7.15, the depth reached by the liquid front reduced for the case of 12 mm diameter hole because of an increase in the vaporization effect as a result of the increased heat transfer area between the LNG and glass wool. For the first 114 seconds of LNG flow down the flat joint, the vaporization effect ensured that the liquid front only reached a depth of 0.102 m for the 12 mm diameter case which is noticeably far away from the bottom of the flat joint. Considering the 12 mm case for the first 65 seconds, the liquid front could only reach a depth of 0.05 m. This observation is quite different from what was observed for the base case simulation of Section 7.1.1 where the liquid front reached the bottom of the flat joint just under 65 seconds. Thus, the difference in the depth versus time profiles between the 12 mm and 4 mm diameter cases differ by a factor of about 3, thus suggesting that the size of the ruptured hole is another important parameter in the determination of liquid penetration rates.

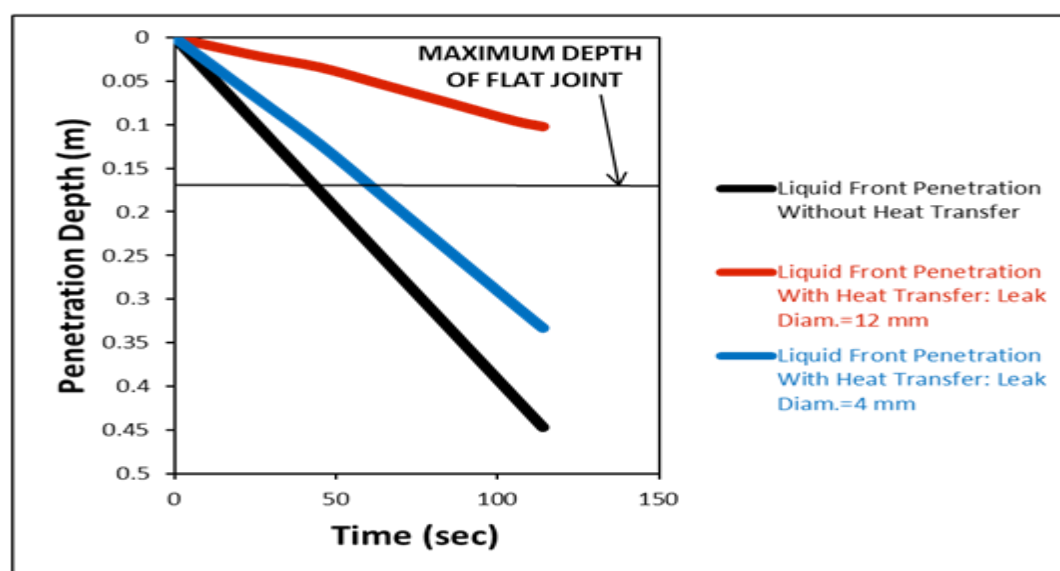


**Figure 7.15:** Penetration depth versus time curves for LNG flowing through a flat joint constructed of glass wool porous material with consideration of the effect of different rupture (hole) diameters (application case no. 3).

Sections 7.1.1 and 7.1.3 analysed LNG flow through a flat joint and established that the heat transfer and vaporization of the LNG has an effect on how deep the

liquid front would propagate down the flat joint. Results suggest that the effect of vaporization decreases the velocity of the liquid front relative to the initial injection velocity. For the simulation conditions examined in this section, the maximum time required for the liquid front to reach the bottom of the flat joint is in the range of a few minutes. Parametric trends further indicated that the vaporizing flow through the flat joint is significantly affected by both secondary barrier leakage rates and secondary barrier rupture (hole) diameters.

## **7.2 Application Case No. 2: LNG pool on inner hull of a MARK III-Type LNG ship**

It is expected that LNG Seepage through the secondary insulation layer and/or flat joint would result in LNG spillage onto inner hull of the ship, which can subsequently lead to a potential cargo tank failure resulting from the brittle fracture of the inner hull. Therefore, the objective of this second application case is to analyse the behaviour of an LNG pool on inner hull following seepage from the flat joint. But before proceeding, it may be worthwhile to state here that this pool-on-hull analysis is also useful in assessing the likelihood of cascading failure of the LNG ship. We may attempt a more specific definition of what is meant by “cascading failure” at this point - cascading failure is the damage that causes the spread of sufficient LNG cargo that then lead to additional damage or breaches to the vessel beyond the damage that produced the initial spill or leak (Petti and Lopez, 2009). We will analyse the results from predicted profiles of cooling rate (surface temperature), surface heat flux and vaporization rate. The ductile-to-brittle transition temperature criterion will be used in determining a threshold temperature of the inner hull surface below which the inner hull may be susceptible to brittle fracture. The ductile-to-brittle transition temperature assumed in this work is 213 K (-60°C) (Choi et al., 2012). Below this brittle transition regime, we expect the integrity of the LNG ship to be compromised and cargo tank failure is highly probable.

### **7.2.1 LNG Spill Conditions**

We are assuming, for the purpose of this work, that the heat source to the pool is solely from the inner hull. Heat contributions from mastic and nitrogen gas are considered negligible (Dienhart, 1995) and are hereby neglected. There are six

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spill conditions investigated in this section. The first spill condition (spill condition no. 1) is a base case simulation that provided useful information on the inner hull's response to instantaneous LNG spillages. In spill condition no. 2, the thermal conductivity of the inner hull was varied in order to assess the effect of this parameter on LNG pool behaviour. This spill is again assumed to be instantaneous. In spill condition no. 3, we investigated the influence of inner hull thickness on the vaporization rate of an instantaneous spill event. In spill condition no. 4, a continuous spill of LNG on inner hull is analysed in order to understand how pool behaviour for this case differs from the instantaneous spill case(s). The fifth spill condition (spill condition no. 5) studied the influence of continuous spill rates on pool spreading and vaporization. The last spill condition (spill condition no. 6) analysed how the size of the hole through which the LNG is continuously spilling, could influence the spreading and vaporization rate of the pool.

#### **7.2.1.1 Spill Condition No. 1: Instantaneous LNG spillage on inner hull with the developed liquid pool completely filling the void area and inner hull surface of the ship**

This base case LNG pool simulation is characterized by an LNG pool instantaneously covering the entire surface of the inner hull following spillage (or seepage as in this case). Figure 7.16 shows the different processes of inner hull heat transfer to the pool where it is indicated that LNG is initially (at  $t=0$ ) assumed to take the form of a flat cylinder above the inner hull, with the radius of the pool (of height,  $H$ ) equivalent to the radius of the inner hull and in direct contact with it (i.e. inner hull). The pool would vaporize upon contact with the inner hull with a film of vapour (of thickness,  $\delta$ ) immediately forming between the LNG and inner hull. This new film boiling heat transfer situation is shown in Figure 7.6 (b) for the specific case of stable film boiling. There is a second 'partial' film boiling – the so-called transition boiling (not shown here but can be visualized from Figure 2.3 of Chapter 2) characterized by a 'partial' collapse of the vapour film and intermittent LNG contact with the inner hull. A total collapse of the vapour film will result following a switch from transition boiling to nucleate boiling and Figure 7.16 (c) shows this situation. Clearly, the LNG pool

directly contacts the inner hull during nucleate boiling with the physical model resembling that of Figure 7.16 (a).

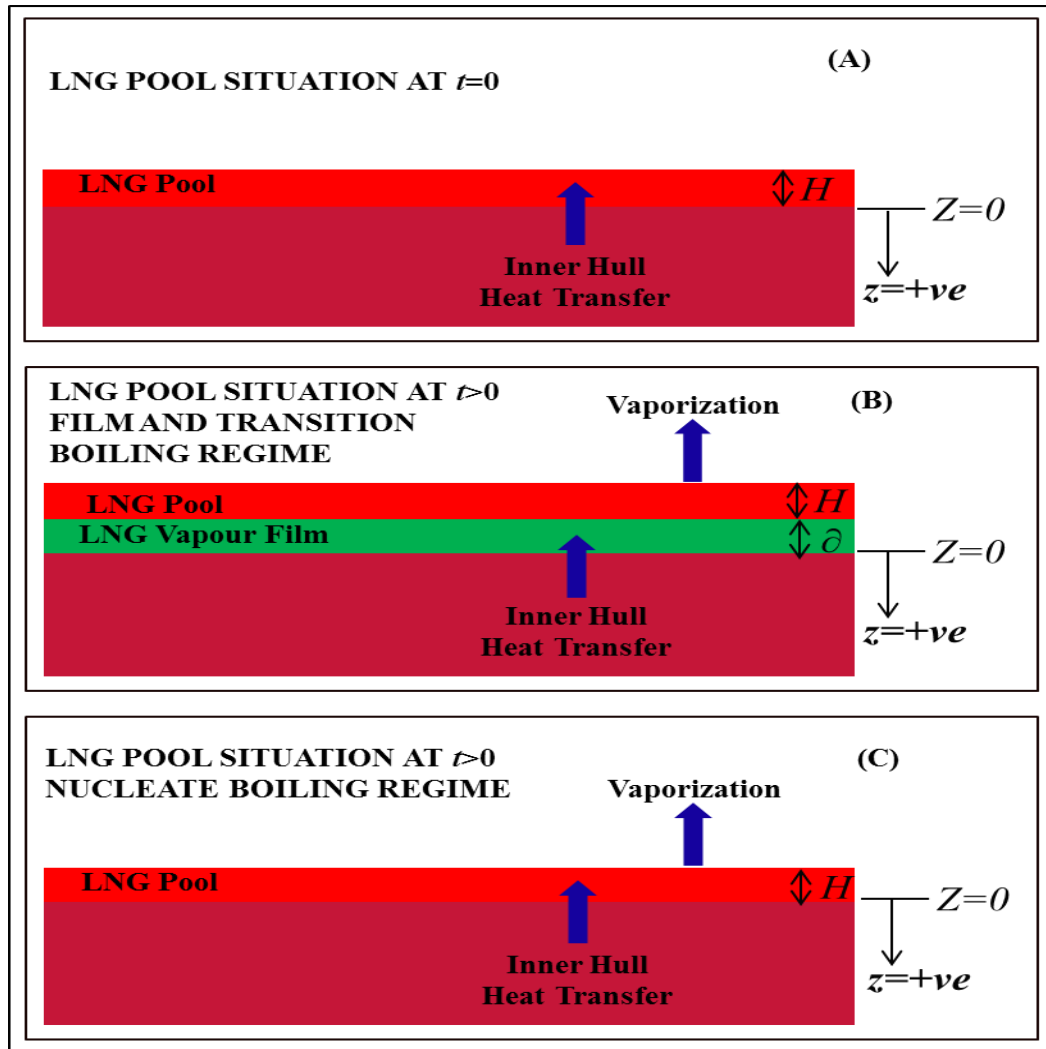


**Figure 7.16:** Schematic diagrams showing the different heat transfer situations in the event of an instantaneous LNG spill on inner hull of a Mark III membrane type LNG ship; (a) the heat transfer situation at  $t=0$ ; (b) the heat transfer situation during film and transition boiling on inner hull (the physical model of transition boiling indicating 'partial' LNG contact with inner hull is not shown); (c) heat transfer situation during nucleate boiling.

Figure 7.17 shows the cooling rate profiles for the inner hull and the heat transfer coefficient profiles between the LNG and inner hull. The inner hull is an 18 mm thick steel plate, with initial temperature specified as 273 K. The black curve is the cooling curve which shows the dependence of the inner hull surface temperature on time. The inner hull surface first cools in the film boiling regime. As time increase, a relatively faster cooling rate was noticed indicating that the

stable vapour film has partially collapsed and that the boiling regime has shifted to transition boiling. Ultimately, a shift to nucleate boiling is reached when the inner hull temperature decreased to 112 K. At this point, LNG is in direct contact with the inner hull surface. A very important observation from Figure 7.17 is the relatively short amount of time required for the inner hull surface to decrease to the ductile-to-brittle transition temperature of 213 K ( $-60^{\circ}\text{C}$ ). It took approximately 100 seconds for this to happen. After this time, the ship's inner hull becomes very susceptible to embrittlement or brittle fracture.

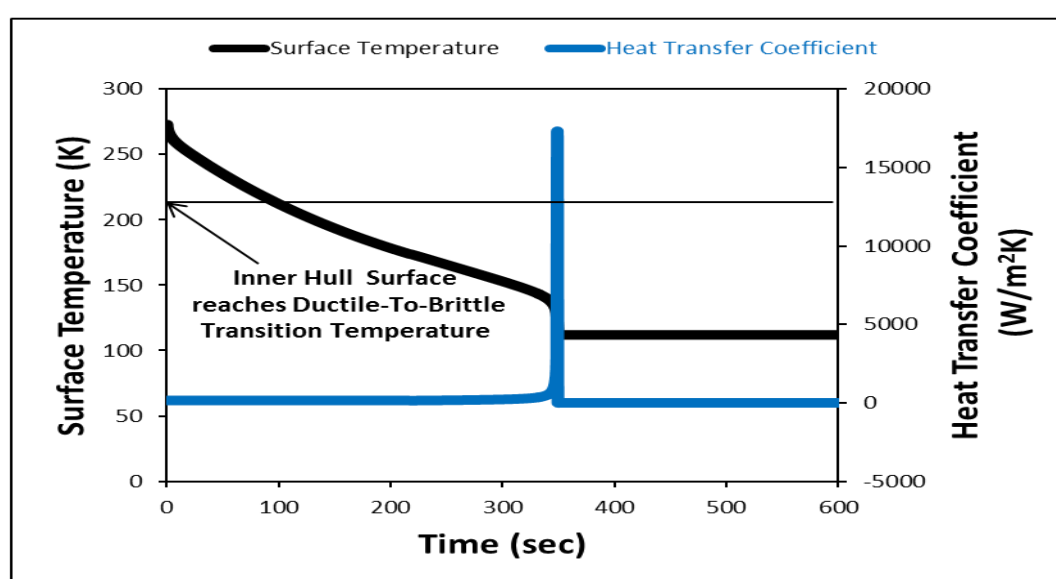


**Figure 7.17:** (a) Black curve: Cooling curve from instantaneous spillage of LNG on inner hull, showing the dependence of the inner hull surface temperature on time. (b) Blue curve: convective heat transfer coefficient as a function of time.

The blue curve of Figure 7.17 also shows the convective heat transfer coefficient versus time profile. As expected for film boiling, the heat transfer coefficient is relatively low during the first 4 minutes because the heat transfer is limited by the low-thermal conductivity vapour film. Thereafter, when the LNG begins to intermittently contact the inner hull, a rapid increase in heat transfer coefficient is noted, suggesting that the heat transfer is now within the transition boiling regime. At the onset of nucleate boiling, the convective heat transfer coefficient immediately drops to zero because the convective heat transfer coefficient is either negligible or non-existent under nucleate boiling. Figures 7.18 and 7.19 show the effect of changing heat transfer regimes in terms of surface heat flux

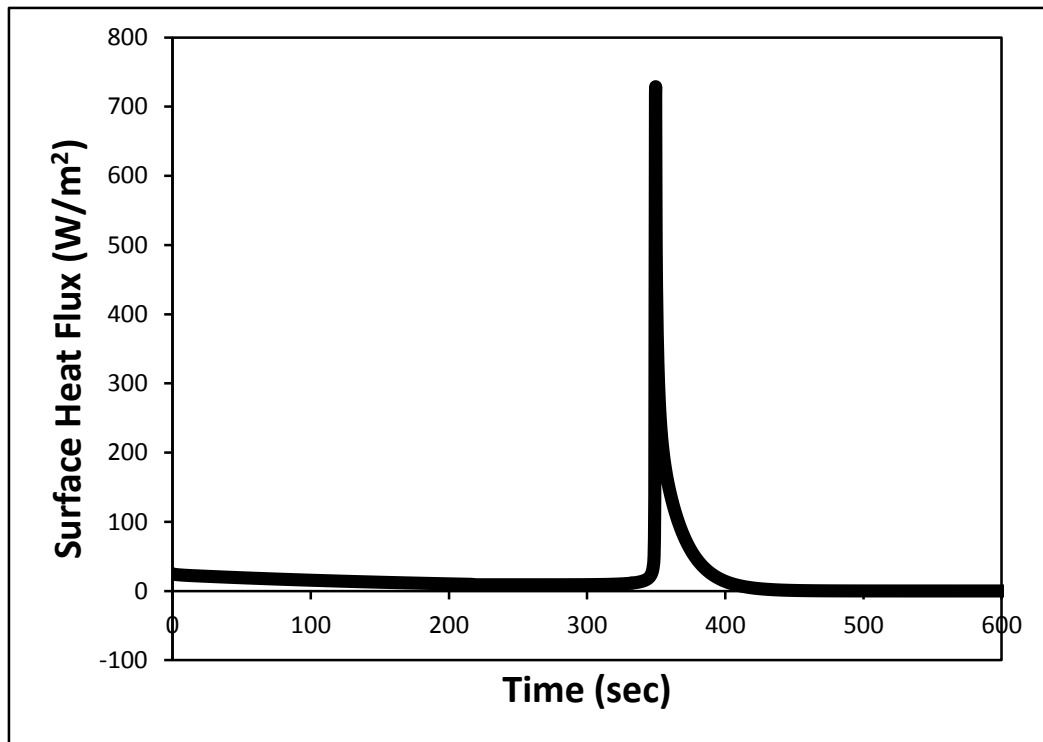


**Figure 7.18:** Surface heat flux versus time history for a spill of LNG on the inner hull of a GTT Mark III-type LNG ship.

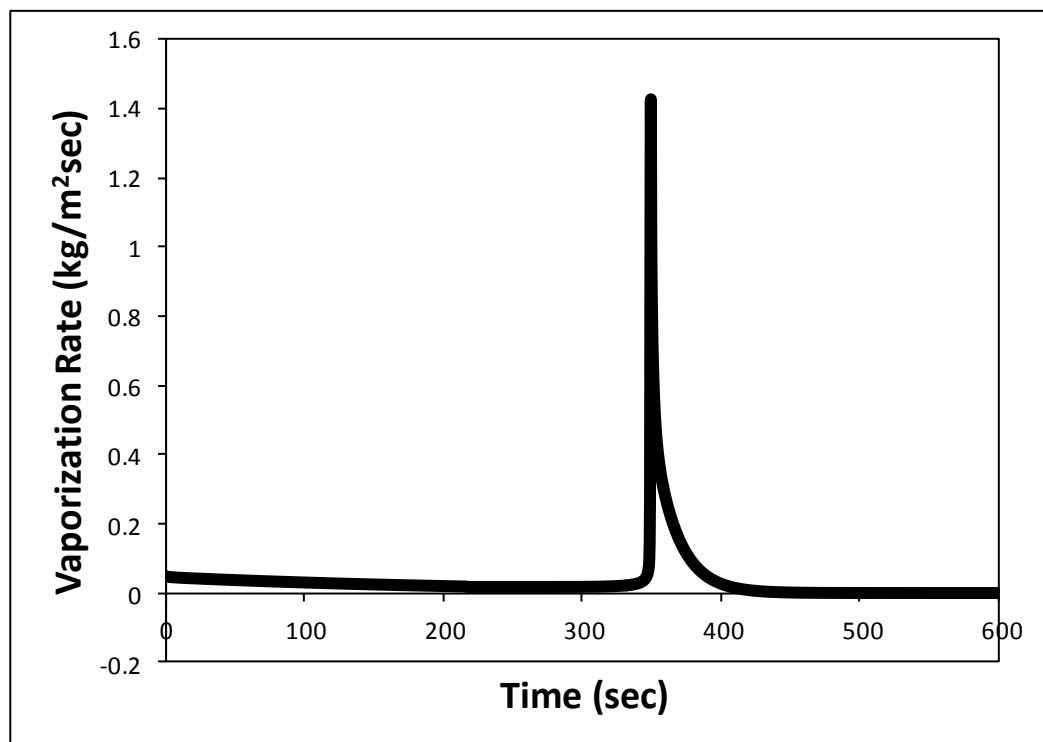


**Figure 7.19:** Vaporization rate versus time history for a spill of LNG on inner hull of a GTT Mark III-type LNG ship.

and vaporization rate. Again, the amount of heat being released by the inner hull during the first 4 minutes is small because the film boiling heat transfer coefficient controls the process within this duration. The vaporization rate is also relatively small as a result. The higher transition boiling heat transfer observed in Figure 7.18 is reflected in the rapid increase in surface heat flux which also explains the increase in LNG vaporization rate. This further explains the similar qualitative features between the surface heat flux and vaporization rate curves shown in Figures 7.18 and 7.19, respectively. The large magnitude of the surface heat flux and vaporization rate at the onset of nucleate boiling is caused by the rapid cooling of the inner hull to the temperature of the LNG, the relatively large magnitude of the inner hull thermal conductivity and the control of the heat transfer process by the thermal properties of the inner hull.

#### 7.2.1.2 Spill Condition No. 2: Instantaneous LNG spillage on inner hull considering three different specification of inner hull thermal conductivity

Figure 7.20 shows the results of a spill condition investigating the influence of inner hull thermal conductivity on the inner hull cooling rate and demonstrates how the change in thermal conductivity would influence how fast the ductile-to-brittle transition temperature would be reached. Three simulation cases were

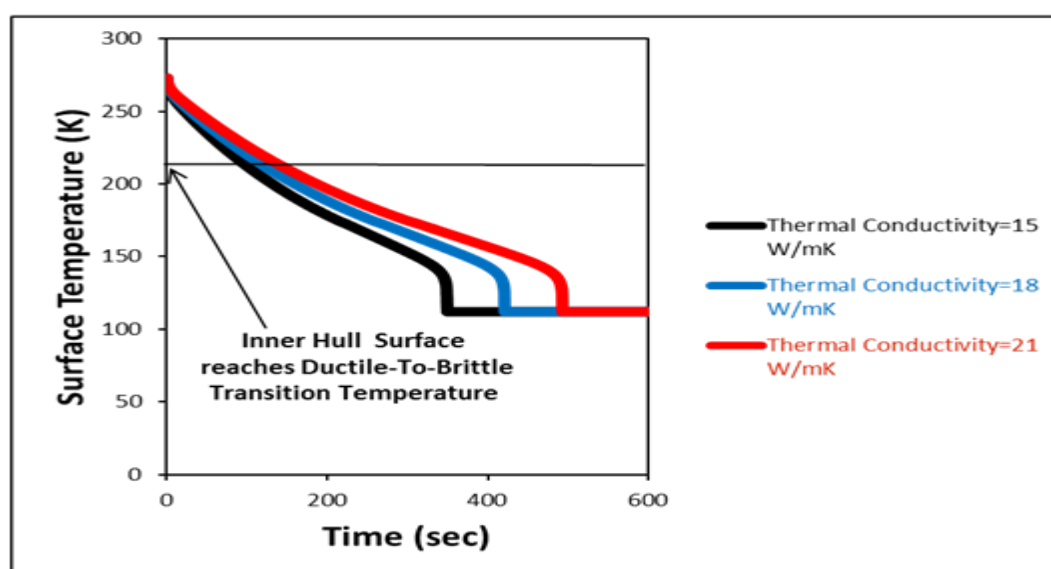


**Figure 7.20:** Predicted inner hull surface temperature profiles for different thermal conductivity values.

run, one with the thermal conductivity specified as 15 W/m/K (base case value), and the other two with the thermal conductivity increased to 18 W/m/K and 21 W/m/K, respectively. The relatively lower cooling trend noted for the higher thermal conductivity values can be attributed to the increased rate of heat flow to the surface of the inner hull, resulting in the cooling rate of the inner hull decreasing. We have attributed the small difference in the early cooling profiles to a ‘film-boiling effect’ where the vapour film thermal conductivity is limiting the heat transfer. If we assume that the ductile-to-brittle transition temperature is, say, 120 K, the differences in the timescale for entering the brittle fracture regime increases because the cooling of the inner hull surface would occur under a different boiling regime where the effect of thermal conductivity on the cooling rate would be stronger. The decrease in cooling rate with increasing thermal conductivity is small, especially, during film boiling. The difference in the time for the inner hull to reach the ductile-to-brittle transition temperature is also small which is clearly insignificant with respect to cargo tank safety. Although the difference increased during the later regimes of boiling, this would still not have positive safety implications since the threshold for the brittle fracture transition regime has long been exceeded during this period of boiling. Figure 7.21 shows the surface heat flux profiles obtained from the three thermal conductivity cases. The trends during film boiling are similar to the base case

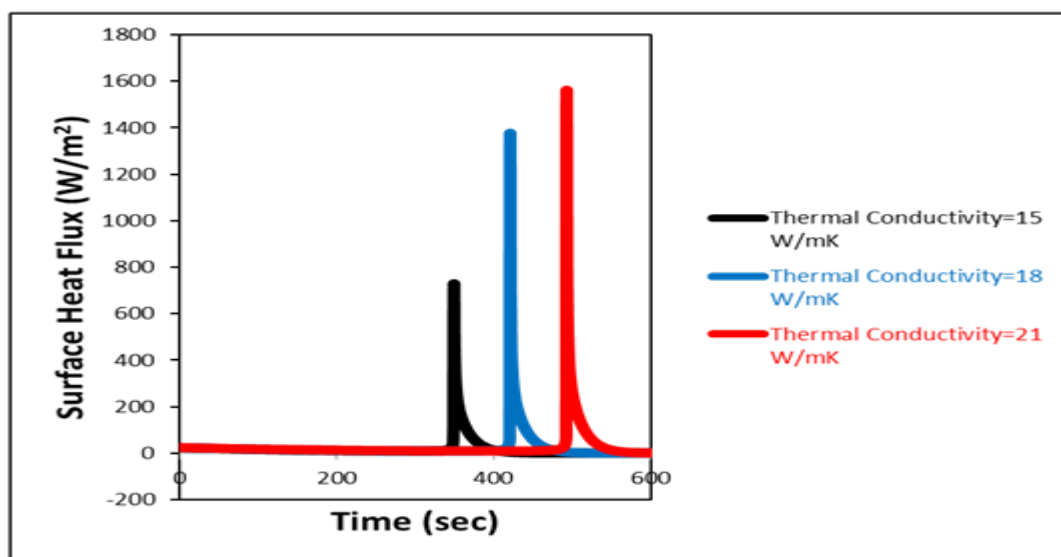


**Figure 7.21:** Predicted inner hull surface heat flux profiles for different thermal conductivities.

simulation. Also, increasing the inner hull thermal conductivity increases the rate and duration of inner hull heat flux and the time for the onset of the different regimes of boiling, with the observed spike signifying the time for maximum inner hull heat flux and LNG pool vaporization rate.

### 7.2.1.3 Spill Condition No. 3: Instantaneous LNG spillage on the inner hull considering two different specification of inner hull thickness

In the base case simulation presented in Section 7.2.1.1, the thickness of the inner hull was specified as 18 mm. Here we analyse the behaviour of an instantaneous LNG spill when a 36 mm thick and 54 mm thick inner hull was specified. The results of this study are shown in Figure 7.22. Figure 7.22 indicates a decreasing cooling rate as inner hull thickness increases. The temperature history for the 18 mm thick inner hull reveals a higher cooling rate than the 36 mm and 54 mm simulation cases. As the inner hull thickness increases, the time to reach the brittle transition temperature increases. These times are 94, 160 and 211 seconds for the 18, 36 and 54 mm thicknesses, respectively. However, the differences in the three profiles increases during the later regimes of boiling. These results suggest that inner hull thickness affects the cooling rate by increasing the amount of heat released by the inner hull for the vaporization of the LNG. A more prolonged film boiling is expected with larger substrate thickness. However, the relatively small difference in the times

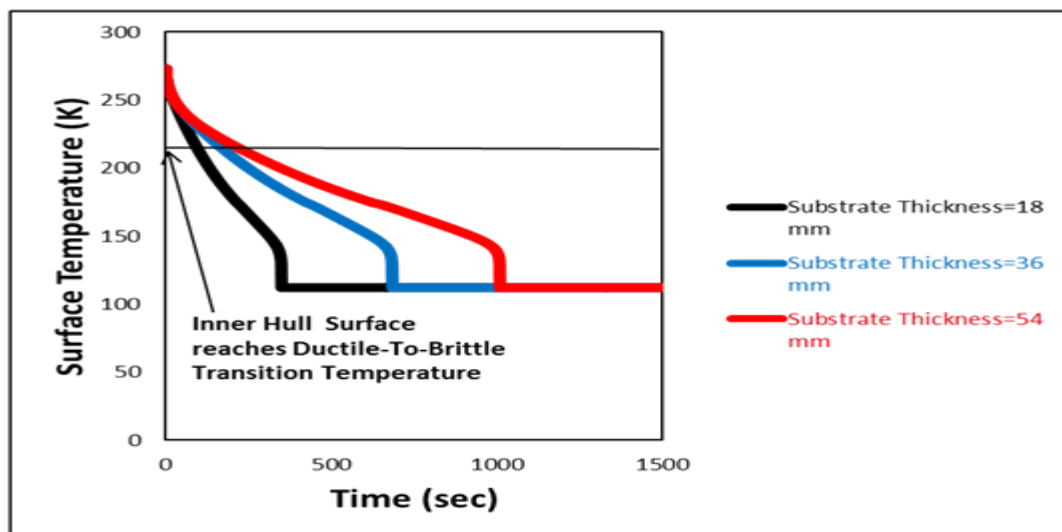


**Figure 7.22:** Predicted inner hull surface temperature profiles for different substrate thickness.

**Table 7.3** Results for inner hull thermal conductivity and thickness variation following instantaneous spill of LNG on inner hull.

| <i>Parametric Factor</i>            | <i>Time To Reach Ductile-To-Brittle Transition Temperature (sec)</i> | <i>Time For Onset Of Transition Boiling (sec)</i> | <i>Time For Onset Of Nucleate Boiling (sec)</i> | <i>Maximum Heat Flux At Onset Of Nucleate Boiling (KW/m<sup>2</sup>)</i> | <i>Maximum Vaporization Rate At Onset Of Nucleate Boiling (kg/m<sup>2</sup>sec)</i> |
|-------------------------------------|----------------------------------------------------------------------|---------------------------------------------------|-------------------------------------------------|--------------------------------------------------------------------------|-------------------------------------------------------------------------------------|
| <b>Thermal Conductivity (W/m-K)</b> |                                                                      |                                                   |                                                 |                                                                          |                                                                                     |
| $k_s=15$                            | 99                                                                   | 217                                               | 350                                             | 719                                                                      | 1.41                                                                                |
| $k_s=18$                            | 120                                                                  | 261                                               | 421                                             | 1370                                                                     | 2.67                                                                                |
| $k_s=21$                            | 140                                                                  | 304                                               | 493                                             | 1550                                                                     | 3.03                                                                                |
| <b>Inner Hull Thickness (W/m-K)</b> |                                                                      |                                                   |                                                 |                                                                          |                                                                                     |
| $Thickness_b=18$                    | 99                                                                   | 217                                               | 350                                             | 719                                                                      | 1.41                                                                                |
| $Thickness=36$                      | 175                                                                  | 429                                               | 684                                             | 721                                                                      | 1.41                                                                                |
| $Thickness=54$                      | 229                                                                  | 632                                               | 1010                                            | 1990                                                                     | 3.90                                                                                |

to reach the ductile-to-brittle transition temperature has no significant positive safety implications for the cargo containment system. Table 7.3 above lists the results of the sensitivity study on thermal conductivity and inner hull thickness, where it is shown that as both parameters are increased, the time to reach the ductile-to-brittle temperature, the time for the onset of transition and nucleate boiling, and the maximum heat flux and vaporization rate all increased.

#### **7.2.1.4 Spill Condition No. 4: Continuous LNG spillage on the inner hull considering the effect of spreading on inner hull surface**

In Sections 7.2.1.1 to 7.2.1.3, an assumption that the LNG spill is instantaneous was made. Further assumption that the instantaneous spill results in the complete filling of the void area was also considered. We believe these assumption to be plausible for sufficiently large breach (hole) sizes and/or appropriate for a large spillage of extremely short duration compared to spreading duration. For relatively small breach (hole) sizes, the spilled LNG may contact the inner hull surface in drops eventually forming a pool that would simultaneously spread and vaporize until the inner hull enclosure stops the spreading. In this section as well as in Sections 7.2.1.5 and 7.2.1.6, continuous spillages of LNG on inner hull are investigated. The same simulation conditions

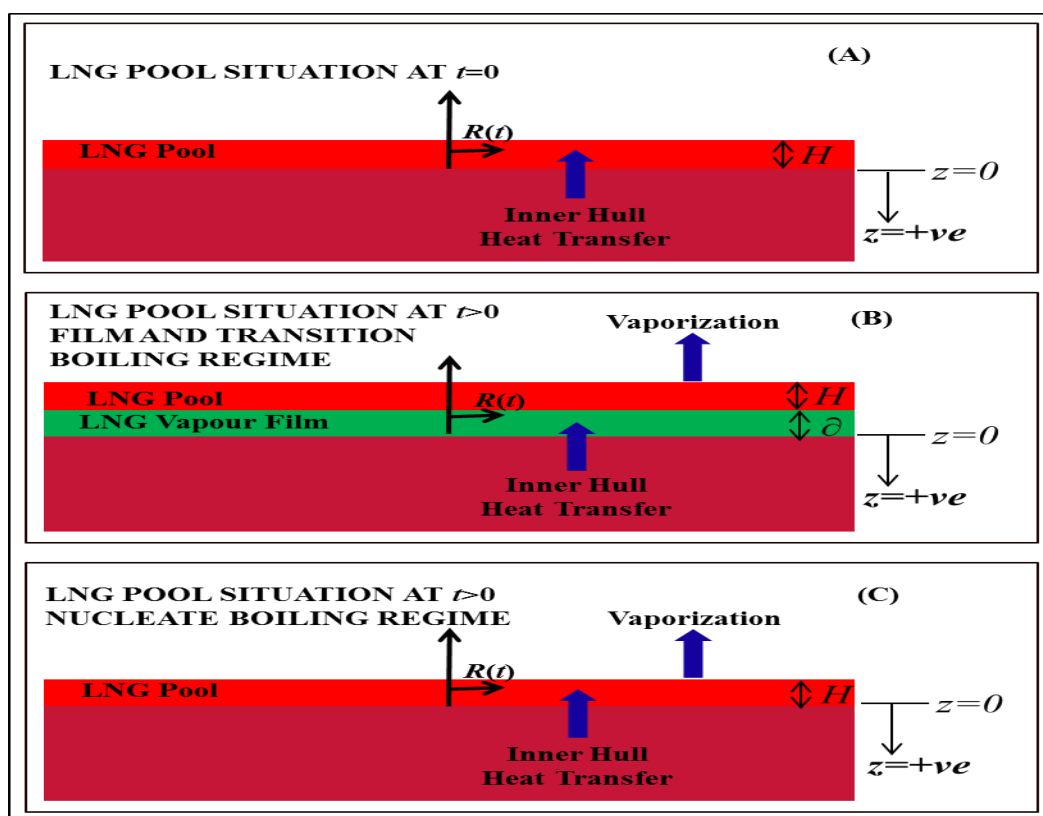


**Figure 7.23:** Schematic diagrams showing the different heat transfer situations in the event of a continuous LNG spill on inner hull of Mark III membrane type LNG ship with a spreading relationship coupled to the vaporization rate model.  $R(t)$  is the time dependent pool radius; (a) the heat transfer situation at  $t=0$ ; (b) the heat transfer situation during film and transition boiling on inner hull (the physical model of transition boiling indicating 'partial' LNG contact with inner hull is not shown); (c) heat transfer situation during nucleate boiling.

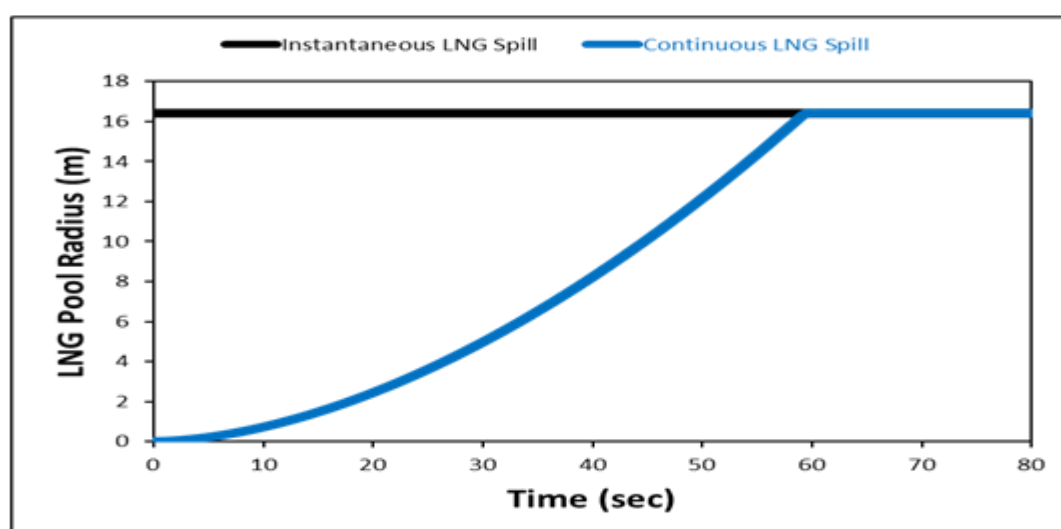


**Figure 7.24:** Comparison of the LNG pool radius as predicted by instantaneous LNG spillage on inner hull (black line) and by continuous LNG spillage on inner hull (blue line).

as discussed in Section 7.2.1.1 are applicable here only that this time, pool spreading is explicitly accounted for. Figure 7.23 indicates that LNG pool would spread on the inner hull surface until it is constrained by the inner hull enclosure. Figure 7.24 shows that the LNG pool radius progressively increases to the maximum radius of 16 m. Apparently, this time-dependent pool radius is relatively fast in that it took less than 1 minute for the spreading pool to cover the inner hull surface.

#### 7.2.1.5 Spill Condition No. 5: Continuous LNG spillage on inner hull considering five different specification of LNG spillage (leakage) rate

The continuous spill model predicts that as the spillage or leakage rate decreases, the LNG pool spreads relatively slower from the source leading to an increase in the pool spreading duration (see Figure 7.25). For example, a spillage rate of 0.3 kg/s results in the leading edge of the spreading pool reaching the inner hull enclosure under 17 seconds compared to 60 seconds with a spillage rate of 0.00001 kg/s. Thus, a high spillage rate would tend to cause a relatively faster spread of the pool until the inner hull enclosure limits the spreading. We may infer from these results that the continuous spreading pool that reaches the inner hull enclosure quicker would vaporize more since the contact time at the higher spillage rate would be longer.

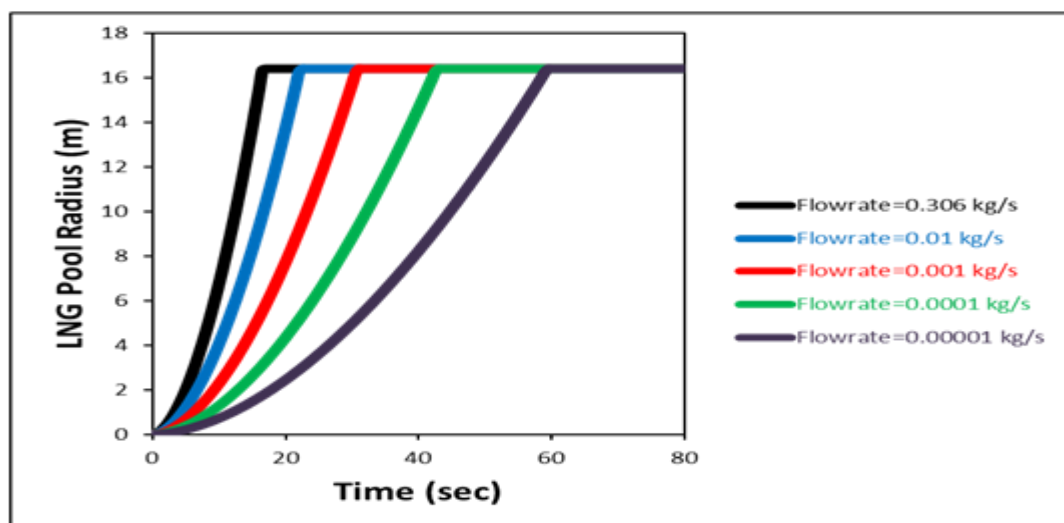


**Figure 7.25:** Predicted LNG pool radius for continuous spillage of LNG for different spillage rates (flowrates).

### 7.2.1.6 Spill Condition No. 6: Continuous LNG spillage on inner hull considering four different specification of LNG spill (hole) diameters/sizes

Here we will be investigating how a change in the size of the hole through which the LNG is spilling, would influence the pool spreading rate. Figure 7.26 shows the time-dependent pool radius for 2, 4, 6 and 10 mm diameter holes. It is clear from the results that as hole sizes increases, the spreading duration decreases. Table 7.4 summarizes the effect of changing the magnitude of the spillage rates and rupture (hole) diameters. The 'b' subscripts indicates base case properties.

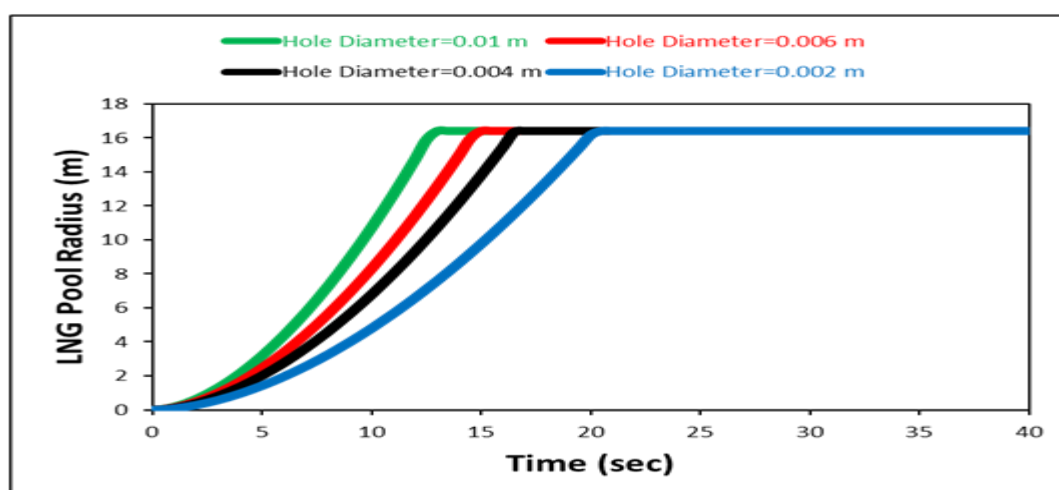


**Figure 7.26:** Predicted LNG pool radius for continuous LNG spillage for different hole diameters (hole sizes).

**Table 7.4** Results for spillage rates and hole diameter variation following continuous spill of LNG on inner hull.

| Parametric Factor           | Time for LNG to Cover the Entire Inner Hull Surface (sec) |
|-----------------------------|-----------------------------------------------------------|
| <b>Spillage Rate (kg/s)</b> |                                                           |
| $SR_b=0.31$                 | 17                                                        |
| $SR=0.01$                   | 23                                                        |
| $SR=0.001$                  | 31                                                        |
| $SR=0.0001$                 | 43                                                        |
| $SR=0.00001$                | 60                                                        |
| <b>Hole Diameter (m)</b>    |                                                           |
| $HD=0.002$                  | 21                                                        |
| $HD_b=0.004$                | 17                                                        |
| $HD=0.006$                  | 15                                                        |
| $HD=0.01$                   | 13                                                        |

### 7.3 Application Case No. 3: Simulation of vaporizing water flow through superheated geologic formations

This application case examines an analogous process in radioactive waste isolation technology where radioactive waste storage in geologic formations above the water table has been proposed as a veritable means of radioactive waste isolation. The superheated zones around the radioactive waste packages can reduce the downward flow of vaporizing water (Ramspott, 1991; Chestnut, 1992; Birkholzer, 2003; Nitao et al., 1992; Buscheck and Nitao, 1993; Buscheck et al., 1991). One aspect of the problem is to understand how the effect of water vaporization could hinder and/or prevent water penetration rates and/or seepage. There has been investigations on this subject by a number of authors (Pruess et al., 1984, 1985, 1988, 1990a, 1990b; Pruess and Tsang, 1993; Phillips, 1996) and these studies have demonstrated that a large, superheated, dry rock zone will form around the waste packages for several hundred years.

#### 7.3.1 Simulation Results based on Matrix-Dominated Flow

Water flow in superheated geologic formation can be either matrix-dominated or fracture dominated (Buscheck and Nitao, 1993; Buscheck et al., 1991). In this section, we present results of four simulation cases in which matrix-dominated flow of water through these superheated zones is analysed (see Figure 7.27). This

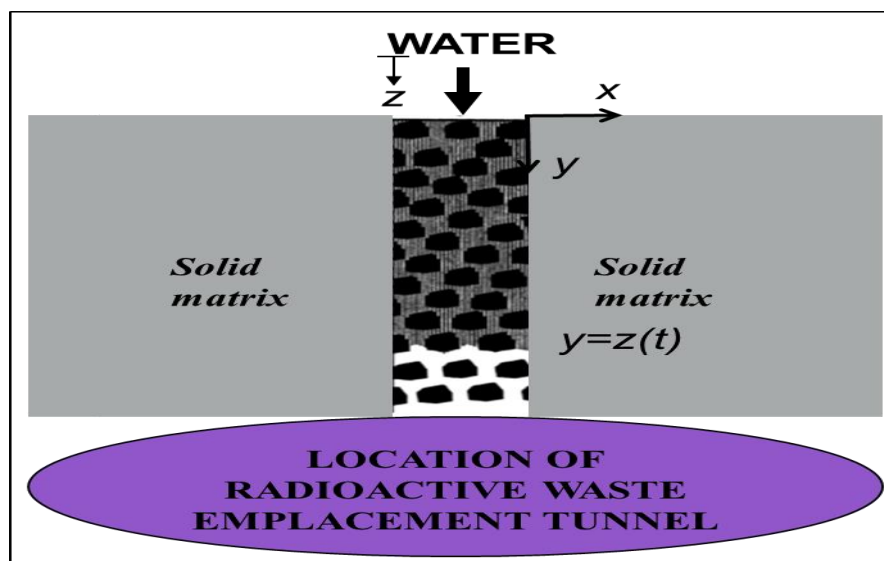


**Figure 7.27:** A schematic diagram showing the downward flow of water through the geological formation above radioactive waste emplacement tunnels.

analysis is particularly useful since existing studies have not adequately investigated this aspect of the problem until now. In Sections 7.3.1.1 through 7.3.1.3, we have presented some results of sample problems. We assume that the initial temperature of the superheated zone is approximately 377 K. We also assume that the superheated zone extends to a depth of 3 m above the crown of the radioactive waste emplacement drift or tunnel. The bounding solid matrices supplies the heat used for the vaporization of the flowing water. The location of the radioactive waste emplacement tunnels is just below the identified porous layer shown in Figure 7.27. Heat conduction within the matrix can occur in both the horizontal and vertical direction.

#### **7.3.1.1 Simulation Case No. 1: Base-Case analysis of water penetration into superheated rock overlying a radioactive waste emplacement drift**

This base case simulation study is to understand how far the liquid front could penetrate a superheated geologic formation (or porous layer) overlying a radioactive waste emplacement tunnel. Table 7.5 lists the parameters and properties used in all simulations in this section. Figure 7.28 shows the liquid front depth versus time for a period of 60 sec. For the first 10 seconds, the liquid front has already reached a depth of 400 mm. At times greater than 10 seconds, the liquid front could not move above this depth since a steady-state flow is already established at this time. Steady-state signifies the end of liquid front penetration even though constant water injection at the top is still ongoing. The vaporizing liquid front (blue curve) deviated significantly from the liquid front that predicts a non-vaporizing water flow (black curve). Consequently, the liquid

**Table 7.5** Assumed conditions for water flow simulations.

| <i>Water and Rock Properties</i>      |                       |
|---------------------------------------|-----------------------|
| Water Temperature (K)                 | 373                   |
| Rock Thermal Conductivity (W/m/K)     | 1.2                   |
| Water Thermal Conductivity (W/m/K)    | 0.58                  |
| Water Density (kg/m <sup>3</sup> )    | 961                   |
| Water Viscosity (Pas)                 | $2.91 \times 10^{-4}$ |
| Latent Heat of Vaporisation (KJ/kg)   | 401                   |
| Water Specific Heat Capacity (J/kg/K) | 4210                  |
| Surface Tension (N/m)                 | 0.059                 |
| Initial Rock temperature (K)          | 377                   |

front could only penetrate to the maximum depth of 400 m which is very far away from the bottom of the superheated porous layer (3 m). What these results suggest is that the probability of water seepage from the geologic formation and into the radioactive waste emplacement tunnels is affected by the vaporization effect which contributes to low water penetration rates.

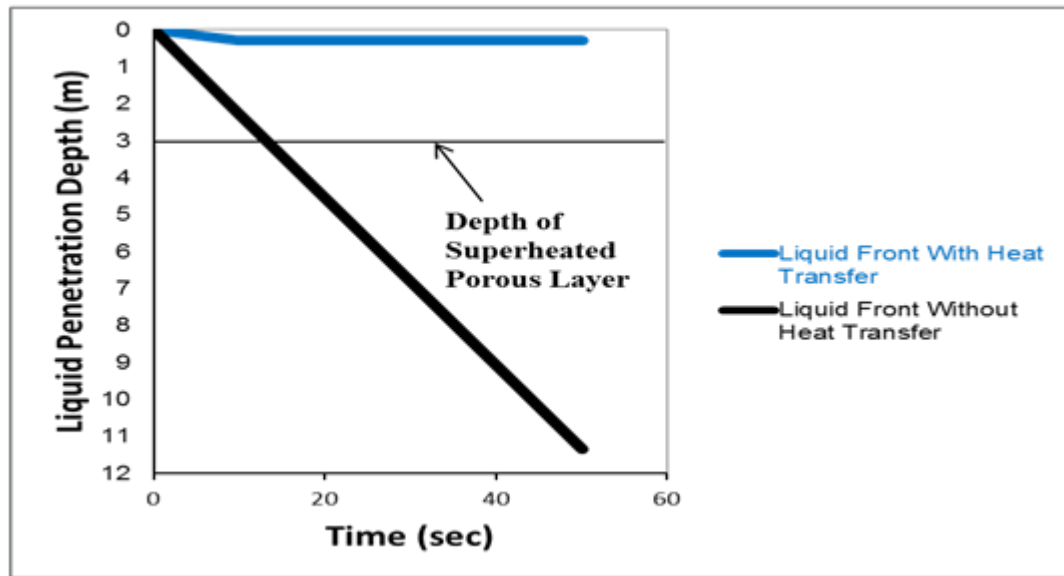


**Figure 7.28:** Predicted penetration depth versus time curve for liquid water flowing through superheated geologic formation above radioactive waste emplacement tunnels.

### 7.3.1.2 Simulation Case No. 2: Sensitivity analysis of water penetration into superheated rock considering two different mass flow rates (injection rates)

As Figure 7.29 shows, increasing the injection rate from its base case value of 0.0004 kg/s to 0.004 kg/s, the penetration depth reached by the liquid front increases rapidly. At 8 seconds, water seepage has already occurred under such short duration. Ignoring the presence of the drift crown at 3 m and assuming that the superheated rock has a new depth of 4 m, water seepage would not be possible this time even for the relatively higher injection rate of 0.004 kg/s. Thus, water injection rate is an important parameter determining liquid penetration rates.

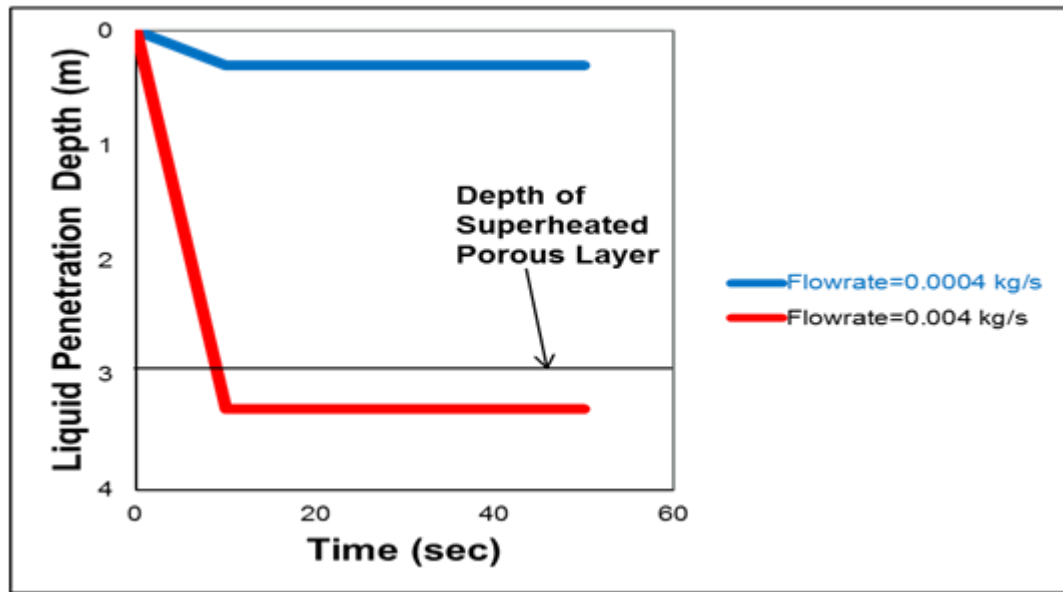


**Figure 7.29:** Predicted penetration depth versus time curve for liquid water flowing through superheated geologic formation for different injection (mass flow) rates.

### 7.3.1.3 Simulation Case No. 3: Sensitivity analysis of water penetration into superheated porous layer considering two different values of thermal conductivity

Figure 7.30 shows the results of a sensitivity study in which the thermal conductivity of the porous layer was increased to 2.4 W/m/K from the base case value of 1.2 W/m/K. The higher thermal conductivity porous layer leads to a

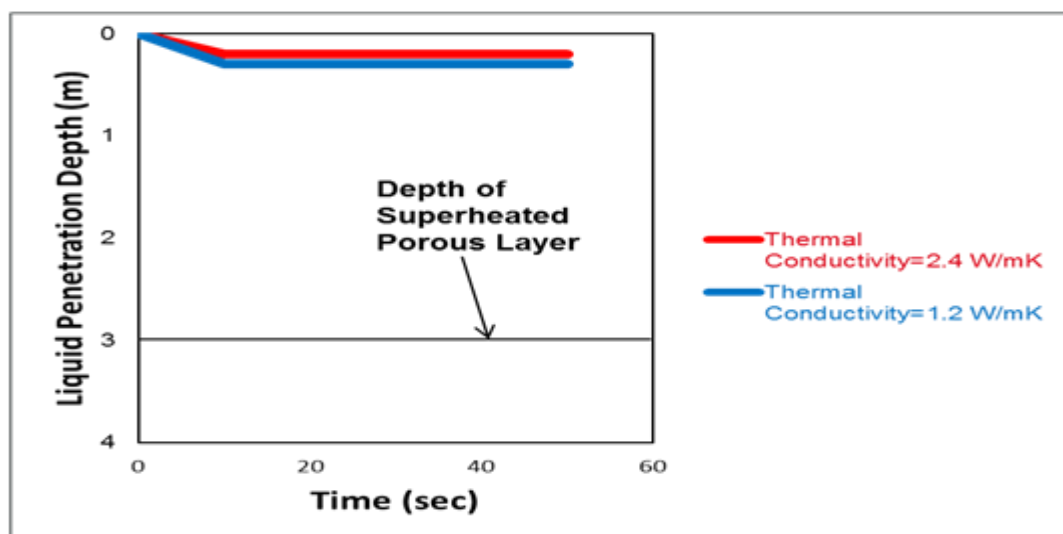


**Figure 7.30:** Predicted penetration depth versus time curve for liquid water flowing through superheated geologic formation considering two different thermal conductivity values.

small decrease in water penetration depth. Figure 7.30 also shows that the water was completely isolated from the waste canisters for both thermal conductivity cases, so we can judge the vaporization effect to be effective in both cases.

## 7.4 Discussion and Conclusions

In this chapter, some of the solution methods presented in this thesis have been applied to three real systems. The analyses demonstrated the use of the models for simulating: (1) vaporizing flow of LNG through flat joints of a GTT Mark III-Type LNG ship; (2) vaporizing LNG pool on inner hull of a GTT Mark III-Type LNG ship and; (3) vaporizing, matrix-dominated flow of water through superheated geologic formations above radioactive waste emplacement tunnels. The following conclusions can be drawn from these analyses:

- For the first application case involving LNG leakages into flat joints of a GTT Mark III-Type LNG ship, the results demonstrate that when the secondary barrier of a single LNG cargo tank in fully-filled condition is damaged, LNG penetration through the secondary insulation layer would occur under very short duration, approximately within the first 65 seconds of secondary barrier leakage. The penetrating LNG will vaporize but the vaporization effect is not sufficient to prevent LNG seepage unto the inner hull of the ship. One implication of seepage is that LNG reaching the inner hull can cause brittle fracture of the inner hull leading to potential failure of the cargo containment system. Thus, the vaporization effect is not an effective safety mechanism for the Mark III-Type LNG ships under secondary barrier leakages. Parametric analyses reveal that the depth of LNG penetration into the flat joint: (1) decreases with decreasing secondary barrier leakage rate and; (2) decreases with increasing secondary barrier rupture (hole) diameters.
  - For the second application case involving LNG spillage onto inner hull of a GTT Mark III-Type LNG ship, the results demonstrate that when LNG is instantaneously spilled on the inner hull surface of a membrane type LNG ship, a small fraction of a pool growing on the entire inner hull
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surface vaporizes as the inner hull surface cools. The ductile-to-brittle transition temperature is reached within the first 100 seconds following spillage. The switch to this brittle fracture regime will occur under film boiling. For continuous spills, the LNG first spreads while simultaneously vaporizing and it took approximately 60 seconds for the spreading LNG pool to reach the inner hull enclosure. A study to assess the parametric effects of inner hull thermal conductivity and inner hull thickness on LNG pool behaviour for the case of instantaneous spillage shows that increasing both parameters decreases the cooling rate of the inner hull and increases the vaporization of the liquid. In the parametric study for continuous spills, variation of spill rate and spill (hole) diameter reveals that increasing the spill rate and spill diameter reduces the time for the pool to reach the inner hull enclosure, with a marginal increase in the total amount of LNG vaporized.

- For the third application case involving a matrix-dominated water flow through superheated geologic formation above radioactive waste emplacement tunnels (or drifts), the results demonstrate that water flow through such superheated geologic formations results in the penetration of the liquid downwards through the superheated rock. Heat transfer has an important influence on water penetration rates exemplified by the effective isolation of the radioactive waste emplacement tunnels since water seepage did not occur for the base case simulation discussed in Section 7.3.1.1. Furthermore, water quickly reached a steady-state value because the specified temperature condition at the top of the drift crown located at the maximum finite depth of 3 m still persists at this depth. The overall implication of these results is that, if we assume the superheated rock to be unfractured, the flow of water through the superheated rock is effectively prevented from contacting the radioactive waste. However, we expect the situation to be different for a densely-fractured rock where fracture-dominated flow could lead to water contact with the waste canisters. Parametric studies investigated the sensitivity of injection (mass flow) rate, and thermal conductivity on the vaporizing, matrix-dominated water flow behaviour and it was clear that, with an

increase in injection rate, the maximum penetration depths reached by water increases. And with an increase in thermal conductivity of the superheated rock, a small decrease in the maximum penetration depth reached by water is observed.

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# 8

## ***Conclusions and Suggestions for Further Work***

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### **8.1 Conclusions**

A model for the description of a one-dimensional vaporizing flow of liquefied natural gas (LNG) through porous media was developed. The model includes a simplified momentum balance based on Darcy's law (Bear, 1988) and also includes the effects of heat transfer, boiling regime and vaporization on liquid penetration rates. The model solves a derived partial differential equation under consideration of specific boundary conditions and provides the profiles for the vapour mass fraction as a function of depth and time. The depth reached by the leading edge of the vaporizing liquid is known from the position of the front and/or interface inside the porous layer. The "front" and "interface" are distinguished by the magnitude of the vapour mass fraction at a particular time. The amount of vapour produced inside the porous medium increases with time until the vapour mass fraction equals unity and we have referred to the depth at which this occurs as corresponding to the position of the interface. The time-dependent "front" position exists prior to the formation of the interface. The model is compared with some analytical and numerical solutions including a commercial software (TOUGH2) and the comparison of the respective results shows very good agreement. The model has generic features and can handle a wide variety of fluids and substrates. The advantages of the model over TOUGH2 were identified in Chapter 4 of this thesis.

More importantly, many of the existing models simply ignored liquid penetration rates following a spill of LNG on porous or penetrable media. A few of the models that studied the effect of liquid penetration are lacking by not including important physical effects including: effects of heat conduction, vaporization and boiling regime on LNG penetration rates (Melhem, 2006). Therefore, by providing further understanding through the development of a model that accounts for these identified effects, some of the uncertainties surrounding the simultaneous vaporization and penetration of LNG through porous or penetrable media is significantly reduced. For the specific case of secondary barrier leakages of large membrane tank LNG ships, a useful model that can predict the simultaneous heat transfer and vaporization during LNG flow down a flat joint is, until now, lacking (Choi et al., 2012).

Simulation results based on the various model cases suggest that as liquid flows through the porous or penetrable media, different zones develop inside the medium. Initially, a two-phase zone which we have identified as Zone 1 develops close to the injection point and a single phase vapour zone – the so-called Zone 2 comes next. A third zone (Zone 3) which we have assumed comprise mainly of saturated air comes after the single phase vapour zone. If the solid matrix is sufficiently cooled such that the heat flux is equal to zero at a particular time, an additional single phase liquid zone called Zone 4 develops close to the entrance of the porous layer. We have earlier identified that the two-phase liquid-vapour zone (Zone 2) and the single-phase vapour zone (Zone 2) can be separated by a “front” or an “interface”, depending on the magnitude of the vapour mass fraction (see Figures 3.4 and 4.11 of Chapters 3 and 4, respectively) . Again, the position of the “front” or “interface” essentially tells us how far the liquid has penetrated the porous layer.

### **8.1.1 Comparison of the various models**

Our approach to the modelling involves a description of two vaporizing flow model cases which are: (i) 1-D model: and (ii) 2-D model. The 1-D model involved three different model cases, namely: BoilFlowModel I, BoilFlowModel II and BoilFlowModel III. We first considered the BoilFlowModel I case which is the case of a substrate providing an infinite heat source for liquid vaporization. The

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infinite heat source assumption ensured that the surface temperature of the solid matrix is constant. We then develop the BoilFlowModel II which describes the case in which the entire substrate surface temperature cools with time as the liquid flows through the porous layer and vaporizes. The BoilFlowModel III deals with the case in which only the surfaces that have been wetted by the liquid cools as a function of time.

The 1-D models are easily distinguishable with respect to: (a) the position of liquid front and/or liquid interface at a given time; (b) the time to generate an interface and; (c) the movement of the liquid front and/or interface. Figure 4.4 shows that interface position is shortest for BoilFlowModel I and longest for BoilFlowModel II. The BoilFlowModel III which closely follows the solution predictions of the BoilFlowModel II has its interface between the other two 1-D model cases. It is also obvious that the time required to generate this interface would be shortest for the BoilFlowModel I and longest for the BoilFlowModel II as already indicated above and from Figure 4.4.

These results show that the effect of vaporization on liquid penetration rates is strongest for the BoilFlowModel I and weakest for BoilFlowModel II with the BoilFlowModel III having an intermediate vaporization effect relative to the two cases. The 1-D models can further be distinguished in terms of movement of their respective liquid interfaces. The BoilFlowModel I has a stationary interface while the interface for the other 1-D model cases (e.g. BoilFlowModel III) moves suggesting that the BoilFlowModel I is a steady-state model whose interface signifies the maximum depth reached by the liquid front (see Figure 4.16 (a) comparing the interface predictions for BoilFlowModel I and BoilFlowModel III). We subsequently selected the BoilFlowModel III as the most realistic of the 1-D models and further extended it to a 2-D model case in order to test how the respective results could differ in handling specific vaporizing flow problems if vertical heat conduction within the solid matrix is accounted for. It was found that including vertical heat conduction effects further reduces liquid penetration rates (see Figure 5.12).

For the simulation runs based on the BoilFlowModel I, results indicate that the liquid front steadily moves as a function of time but the liquid interface does not move for this model case because of the equilibrium that has been established at a particular liquid front and/or interface location inside the porous layer (see Figure 3.7 of Chapter 3). For the simulation runs based on the BoilFlowModel II, BoilFlowModel III and the 2-D model cases, both the front and the interface moves as a function of time. The respective vapour mass fraction versus depth profiles clearly show the progress of the front and/or interface over time. Figures 4.13 (a) and 5.12 are example results showing the movement of the liquid front down the porous layer. Most of the vaporized liquid is moved downstream the front and/or interface into the single-phase vapour region (Zone 2) which as a consequence continuously grows in size. The velocity of the liquid (or front/interface) were tracked by the models and predictions of liquid velocity is dependent on the choice of vaporizing flow model. The simulation runs based on the various models have shown that the evolution of the different zones inside the porous layer including liquid vaporization and penetration rates strongly depends on the nature of the heat exchange assumptions at the interface between the flowing liquid and the solid matrix.

### **8.1.2 The Role of Heat Transfer**

In order to understand the role of heat transfer on liquid vaporization and penetration rates, we employed some of the models developed in this work in simulating some sample problems. We first focused on the steady-state model (i.e. BoilFlowModel I) and from simulation results, it was found that strong vaporization of the liquid caused by heat transfer has a retarding effect on the maximum extent of liquid front propagation. The reduced liquid front propagation is caused by the vaporization effect forcing the velocity of the front to propagate much slower than the initial injection velocity of the liquid, thus demonstrating how liquid penetration into the porous layer is significantly reduced throughout the duration of the flow. In contrast, when the time-dependent 1-D and 2-D model cases were studied, the front and/or interface move with time and the limiting vaporization effect was observed not to be as strong as the steady-state model case described earlier. These unsteady-state simulation cases reveal relatively low vaporization rates and relatively high penetration

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rates. Comparing the 1-D model (i.e. BoilFlowModel III) results with those of the 2-D model, for example, it was found that the liquid penetration rate for the 1-D case increases more than the rate observed for the 2-D model and the vaporization effect can be deduced as the reason for the difference in liquid front penetration rates. From these results, it is thus clear that the role of heat transfer and vaporization in the penetration of liquids flowing through porous media is to reduce the velocity of the liquid front and retard the propagation of the liquid down the media.

Examining one implication of the results obtained from the 1-D and 2-D vaporizing flow analyses, it can be demonstrated that a hazardous liquid injected into a porous or penetrable medium has, with or without vaporization, the potential to propagate down the porous layer and reach dangerous depth thresholds that could increase both safety and environmental risks. We can support this statement by referring to our findings from evaluating the safety of a GTT Mark III type LNG ship under secondary barrier leakages where it was established that seepage of LNG through the flat joint and onto the inner hull steel structure would occur, despite the vaporization of the cryogen taken place. LNG pool on inner hull has one clear hazard and/or implication in early or immediate failure of the LNG cargo containment system (or cargo tank) as a result of localized brittle fracture (embrittlement) of the steel structures of the inner hull.

Furthermore, hazards would increase if the failure of the cargo containment system is followed by cargo ignition when a source of ignition is present. Ignition of the LNG cargo would lead to catastrophic events such as large fires and explosions. In addition, if there were potential for further damage of the neighbouring tanks in the LNG ship, a cascading failure of all the cargo tanks may result. Therefore, seepage following secondary barrier damage of an LNG membrane tank has important performance implications for the design and constructions of current and future cargo containment systems of large membrane tank LNG ships since the heat transfer and vaporization rate effect capabilities of the flat joint and secondary insulation layers of these tanks would not provide an effective safety mechanism for a postulated secondary barrier

leakage of the containment system. These issues would need to be addressed by the LNG industry.

One of the ways through which some of these issues can be addressed is by improving the design of insulation systems, most especially, the structural materials (e.g. flat joint, re-inforced polyurethane foam (R-PUF)). Improvement of the insulation system and/or materials used in its construction should involve improving and/or establishing the real thermophysical and structural properties of the various materials, most especially, the properties of those materials through which the cryogen would likely propagate (e.g. flat joint). It would also be helpful to understand the real leakage rates and sizes for these large LNG ships under barrier leakages. Such a consideration would improve the heat transfer and vaporization capabilities of the insulation layers such that embrittlement of the inner hull can be constrained far above the ductile-to-brittle transition temperature of 213 K. Another way to address these issues is to develop methods for vaporizing LNG flow and pool predictions which would, upon extensive validation exercises, be useful in the hazard analysis of large LNG ships under barrier leakages. Overall, the basic objective in preventing cargo tank failure in the event of barrier leakage is to design and construct an LNG membrane insulation system that would possess the most desired resistance characteristics for preventing embrittlement for a given barrier leakage scenario.

### **8.1.3 The Role of Boiling Regime**

In order to understand the role of boiling regime on vaporizing flow behaviour, the relevant models were extended from the initially-assumed film-boiling flow to three-regime boiling flow by including full-boiling heat transfer coefficients. Findings from a comparative simulation of some sample problems reveal that the simultaneous vaporization and penetration rates of the flowing liquid is strongly affected by the type of boiling regime occurring at the interface between the flowing liquid and the solid matrix. Results from comparing vaporizing flow behaviour based on solely film boiling solution predictions with those obtained from three-regime boiling solutions have shown that the liquid front under film boiling is more penetrating than the solution predictions obtained under the three-regime boiling case (see Figure 4.30 (a) of Chapter 4). Furthermore, we

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have also shown that the penetration rate of the liquid front under a solely nucleate boiling process, when compared with the predictions of both film and three-regime boiling, closely follows the trends observed for three-regime boiling predictions (see Figure 4.35 of Chapter 4). However, it was also evident, from the thermophysical properties of the insulation material used in LNG membrane tanks, that the simultaneous vaporization and penetration of LNG down the flat may only be controlled by nucleate boiling and incorporating the influence of three-regime boiling appears not to be warranted for this particular application. However, for other applications, such as vaporizing flow of LNG through soil, incorporating the influence of three-regime boiling is clearly warranted, since for this type of system, the effect of boiling regime can be significant as soon as the first heat exchange surface is wetted by LNG.

#### **8.1.4 Parametric Trends**

Results obtained from a sensitivity study suggests that vaporizing fluid flow through porous or penetrable media is significantly affected by several parametric factors including but not limited to the magnitude of the injection velocity, substrate type, thermal conductivity, liquid type and the initial temperature of the substrate. There are differences in simultaneous liquid vaporization and penetration rate predictions when two different magnitudes of injection velocity are used. An increase in injection velocity results in very little heat being supplied for vaporization, and with the vaporization rate relatively reduced, liquid penetration rate relatively increases. Whereas when the injection velocity is relatively low, more heat is released for vaporization, and while the vaporization rate and its effect increases, the penetration rate decreases. We have considered the effect of initial temperature of the porous medium which indicated that, for smaller initial substrate temperature, vaporization rate is low with penetration rates increasing. For higher initial substrate temperature, vaporization rates increases with its impact on liquid penetration rates also increasing. Thus, the porous layer is penetrated less rapidly.

We investigated the effect of thermal conductivity and found that simultaneous liquid vaporization and penetration rates is affected by the magnitude of thermal conductivity specified during the sample simulation runs. For simulations with a

relatively low thermal conductivity, where the heat conduction effects were dominant, liquid vaporization rate increases and the liquid penetration rates decreases. The effect of thermal conductivity could partly explain the results used to demonstrate the influence of substrate type on simultaneous liquid vaporization and penetration rates. With a dry soil substrate as considered in this work, a relatively low penetration of LNG is observed as against the relatively high penetration rates observed when soil was substituted with a glass wool substrate.

The model was also used to demonstrate the influence of liquid type on vaporization and penetration rates. When five different types of liquids, namely: LNG, LPG,  $\text{LN}_2$ ,  $\text{LH}_2$  and  $\text{H}_2\text{O}$ , were injected into the porous layer under the same flow conditions, different penetration depths were reached by the different liquids. This difference can be attributed to the different viscosities and hydraulic conductivities of the respective liquids. Results show that that  $\text{H}_2\text{O}$  has the lowest vaporization rate and the highest penetration rate of all the five liquids studied. This is expected as the hydraulic conductivity of water is the highest of the five. LNG reached a shorter penetration depth than both water and LPG while liquid nitrogen and liquid hydrogen were the least penetrating. It is also obvious that the observed different penetration rates are direct consequences of the rate of vaporization of the respective liquids. Considering, for example, the case of  $\text{LN}_2$  flow, the  $\text{LN}_2$  vaporizes most rapidly, reaching a significantly lesser depth over time.

### **8.1.5 The Liquid Pool Model**

A liquid pool model was adapted from one of the best state-of-the-art models, specifically the one proposed by Conrado and Vesovic (2000) and Vesovic (2007). Because the pool model by Vesovic (2007) only predicts the spreading and vaporization of LNG on water, this model had to be modified to be able to predict pool behaviour on non-penetrable solid substrates such as steel. The pool model implementation led to the prediction of the cooling rate of a non-penetrable solid substrate and the vaporization rate of the liquid. Both confined and unconfined LNG spills were modelled. The modified pool model were verified and validated by comparison with derived analytical solution and experimental data from

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literature. All verification and validation results showed excellent agreement. Results from the confined pool boiling simulations confirm that initial temperature, area of confinement and initial spilled mass can influence the vaporization rate of the liquid pool.

The modified pool was also used to study and advance the state-of-the-art on LNG vaporization behaviour following spillages on inner hull of a large membrane tank LNG ship. This aspect of the problem has not, until now, been investigated and/or elucidated. This is in addition to the vaporizing LNG flow study involving large LNG ships undergoing secondary barrier leakages. With this second study, the state-of-knowledge on the vaporization behaviour of LNG flowing through porous regions (e.g. flat joint) of membrane tank LNG ships under barrier leakages has been advanced. These particular research needs have been highlighted by Choi et al. (2012) in their recent paper dealing with barrier leakages of large membrane tank LNG ships. We believe the barrier leakage study presented in Chapter 7 provided sufficient understanding on the fate of LNG penetrating the secondary insulation layers of these type of ships.

In respect of LNG pool behaviour on inner hull of a large GTT Mark III-type LNG ship, it was found that the inner hull steel substrate reached the ductile-to-brittle transition temperature of  $-60^{\circ}\text{C}$  within 1 minute of secondary barrier leakage. This result suggests that as LNG spills on inner hull, a pool of the cryogen forms on the inner hull and fills the insulation space. For an instantaneous spillage of LNG on the inner hull, the inner hull surface in contact with the pool will cool below the brittle fracture transition temperature under very short duration and the heat transfer between the LNG and inner hull will initially be limited by a stable vapour film (film boiling). A relatively long cool-down period under this film boiling regime is observed implying that the ship's hull structure is susceptible to brittle fracture prior to the switch to the later regimes of boiling (namely, transition and nucleate boiling regimes). The observed situation is quite similar to what was observed during continuous spillages of LNG on inner hull, where the pool initially spread and vaporized until the pool area becomes equivalent with the surface area of the inner hull. This situation reduces the inner hull cooling rate and liquid vaporization rate

since no new surface area is available for heat exchange between the liquid and the substrate. The pool modelling work identified the effect of spreading, leakage rate and rupture (hole) diameter on LNG vaporization. It was found that spreading effect is not significant while simulations with increasingly low leakage rates, the rate at which the LNG pool covers the inner hull decreases. A similar observation is made for increasingly low rupture (hole) diameter.

These results provide new understanding for constraining the cooling rate of the inner hull and the vaporization rate of the LNG which are required in order to maintain a safe inner hull temperature following barrier leakages of LNG ships. Furthermore, these results demonstrate that LNG pool vaporization cannot prevent the inner hull from reaching the ductile-to-brittle transition temperature during the initial cooling and vaporization process when the heat transfer between the pool and inner hull is controlled by film boiling. The film boiling regime is the regime with the slowest cooldown period and which has the conditions in which the longest time for reaching the brittle fracture transition temperature can be observed. A sensitivity study to assess the effects of inner hull thermal conductivity and inner hull thickness on LNG pool behaviour for the case of instantaneous spillages shows that increasing both parameters decreases the cooling rate of the inner hull and increases the vaporization rate of the liquid. In the parametric study for continuous spills, variation of spill rate and spill (hole) diameter reveals that increasing the spill rate and spill diameter reduces the time for the pool to reach the inner hull enclosure, with a marginal increase in the total amount of LNG vaporized.

Overall, the LNG pool analyses have demonstrated the use of a computer model for simulating the spreading, vaporization and cooling rates observed when LNG seepages from a flat joint occurs. The less rapid cooling rate created by increasing the inner hull thickness had been determined to be a factor that impacts on the cooling rate of the inner hull, most especially, during the later stages of the spill. Less impact is observed by increasing the thermal conductivity. However, the cooling of the inner hull during instantaneous and continuous spillages has little to no positive effect on safety. It is also shown that the timescale required for the inner hull to reach the ductile-to-brittle transition temperature is not

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significantly affected when varying the magnitudes of either thermal conductivity or substrate thickness. Even though these results demonstrate that vaporization rate of the pool may not be critical in preventing the inner hull from reaching the brittle fracture transition temperature, nonetheless, the present work demonstrated the need to account for spreading and vaporization phenomena in addition to inner hull cooling process, specifically in the case of both instantaneous and continuous spillages. However, the recommendation for safety professionals interested in evaluating the safety implications of LNG seepages following secondary barrier leakages is to carefully examine the characteristics of the LNG spill scenario and to perform a sensitivity study over a broad range of thermal conductivity and thickness values. The results from the simulations can be used in design and construction studies for LNG cargo containment systems. They can also be used to satisfy the prescriptive requirements of LNG codes such as, for example, prescribing that the inner hull should be constructed of special steel materials (of specific thermal conductivity or special coating) for optimal resistance to cryogenic temperature.

The simulation of an analogous process of matrix-dominated water flow down superheated geological formation that was performed in chapter 7 demonstrates that the rate of penetration of water down the superheated porous layer depends significantly on the mass flow rate, and to a lesser extent on the thermal conductivity of the solid matrix. For most situations of matrix-dominated flow, water is isolated from the radioactive waste emplacement tunnels thereby avoiding the potential corrosion and contamination of ground water

## **8.2 Suggestions for Further Work**

Although the vaporizing fluid flow models presented in this thesis are useful for understanding relevant phenomena and has elucidated the simultaneous vaporization and penetration processes that are important in understanding vaporizing fluid flow through porous or penetrable media, the models could still be extended in several ways including but not limited to the following:

- Incorporate the effects of LNG composition on vaporizing flow behaviour. In the current work, LNG was modelled using the properties of pure methane. It would be insightful to understand how, say the liquid front, would propagate when LNG compositional effects are accounted for;
  - Include the effects of temperature-dependent properties of the substrates;
  - Include the effects of phase velocities. Current work used Darcy's law so it would be helpful to use a more rigorous momentum balance to describe liquid motion and/or movement down the porous medium thus allowing for the incorporation of other relevant physical effects into the model.
  - Perform experiments to better understand how the vaporization effect limits liquid penetration rates and to validate developed models;
  - Apply an implicit finite difference or finite element scheme. The current numerical scheme is the explicit finite difference method. It should be explored how the application of an implicit scheme or any other relevant numerical scheme could compare with the present scheme in terms of improvements on the convergence of the numerical procedure.
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# 9

## ***Appendices***

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## ***A. Description of the Iterative Procedure for the Vaporizing Flow Model***

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The flowchart of Figure A.1 in this first appendix provides further details of the FORTRAN implementation of the computer program presented in this thesis.

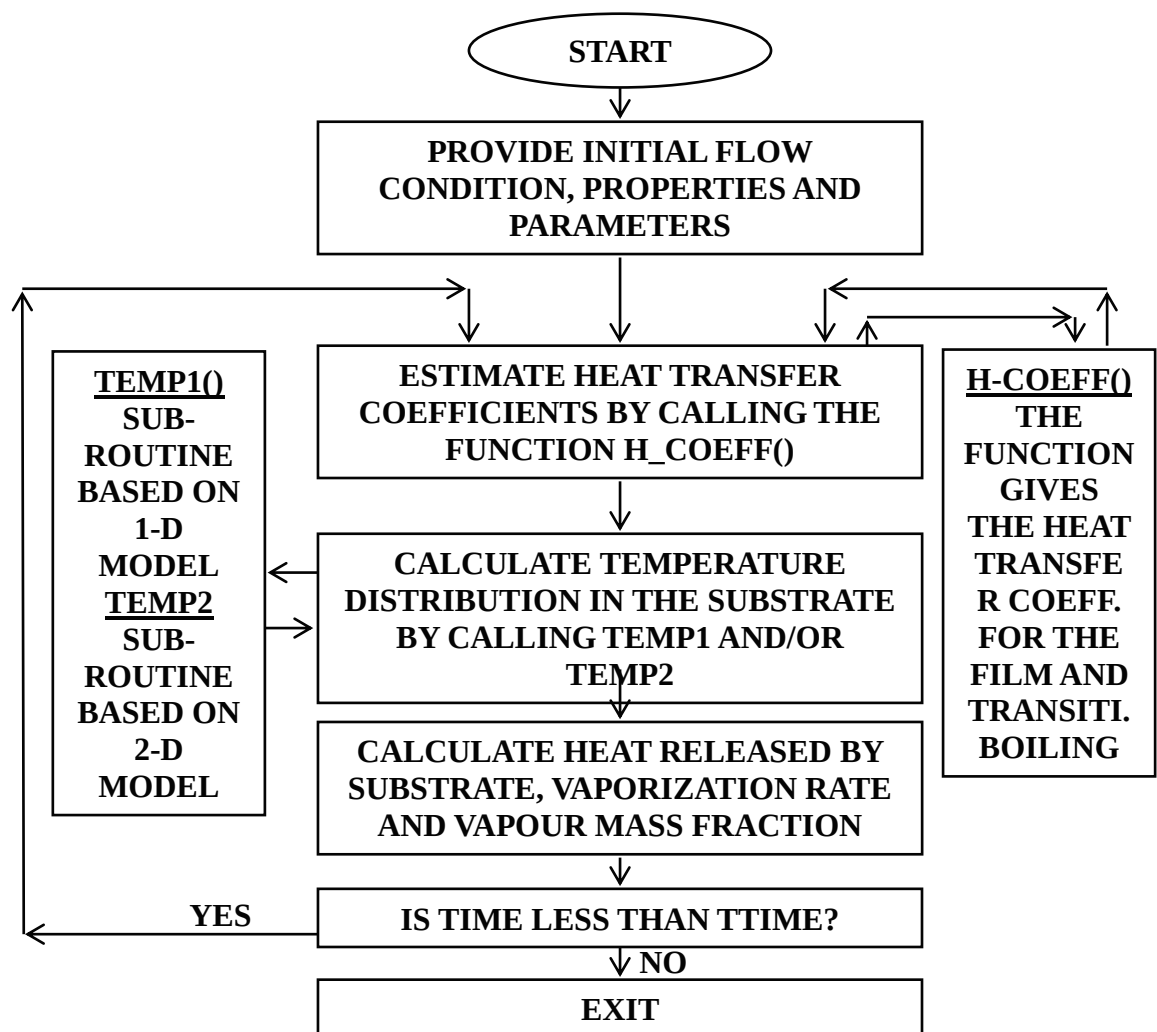


**Figure A.1:** A flow chart showing the program structure used in running the 1-D (including the BoilFlowModel I, BoilFlowModel II and BoilFlowModel III) and the 2-D model.

All the three versions of the 1-D model, namely; BoilFlowModel I, BoilFlowModel II and BoilFlowModel III as well as the 2-D model all follow the same structure as shown in Figure A.1. The computation follows a number of steps including:

- At the start of the simulation, relevant initial conditions, properties and parameters are specified. Example initial input data set may include liquid injection rate into the porous medium, injection velocity, injected liquid width or cross-sectional area of flow, depth and width of porous layer, initial substrate temperature distribution, thermal properties of porous layer such as thermal conductivity and thermal diffusivity, size of discretization and control parameters such as total simulation time (ttime).
- Following the specification of all initial conditions, the H-COEFF() function is called in order to estimate the heat transfer coefficients. Both the film and transition boiling heat transfer coefficients are determined by H-COEFF(). If the substrate surface temperature is below the leidenfrost temperature, film boiling heat transfer coefficient is calculated using the correlations of Klimenko (1981), the correlations of Kalinin et al. (1975) is used to estimate the transition boiling heat transfer coefficient.
- The calculation of the heat transfer coefficient is followed by the calculation of the temperature distribution in the substrate. Subroutine TEMP1 is called if one of the three available 1-D models is desired. The first 1-D model based on the infinite heat source assumption (i.e. BoilFlowModel I) simply returns a constant surface temperature with a constant film boiling heat transfer coefficient. The calculation of the temperature distribution in the porous layer are performed according to the respective choice of time-dependent models including the BoilFlowModel II and BoilFlowModel III (both within subroutine TEMP1()) and the 2-D model (subroutine TEMP2).
- Upon calculation and return of the respective temperature gradients at the interface between the liquid and the solid matrix, the heat flux,

vaporization rate and vapour mass fraction profiles are calculated in the next step. For such time-dependent heat conduction cases undergoing either film or transition boiling, the surface temperature of the substrate are calculated and returned while the surface heat flux is returned if the vaporization process has shifted to nucleate boiling regime.

- With the knowledge of surface temperature and surface heat, the program calculates other properties in the next step including the vaporization rate and the vapour mass fraction. The program exits if the specified total simulation time or run time is exceeded.

An CD containing full description of the source codes and algorithms is included as appendix.

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## ***B. Description of Runge-Kutta Numerical Solution of the Vaporizing Pool Model***

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The Runge-Kutta Method is used to solve the vaporizing pool model implemented in Chapter 6 of this thesis (Conrado and Vesovic, 2000; Zubairu, 2011). The numerical procedure involves a determination of the pool radius and mass loss at end and beginning of current time steps. Taking  $p+1$  as the current time step and  $p$  as a previous time step (Conrado and Vesovic, 2000; Zubairu, 2011), the time-dependent radius and mass of the vaporizing pool can be determined. Defining values of current time step gives the following (Zubairu, 2011),

$$R_{p+1} = R_p + \Delta R_p \quad (\text{B.1a})$$

$$M_{p+1} = M_p + \Delta M_p \quad (\text{B.1b})$$

where the increment  $\Delta R_p$  and  $\Delta M_p$  are obtained from using the following steps (Conrado and Vesovic, 2000; Zubairu, 2011),

$$\Delta R_p = \frac{1}{6}(\Delta R_{p,1} + 2\Delta R_{p,2} + 2\Delta R_{p,3} + \Delta R_{p,4}) \quad (\text{B.2a})$$

$$M_p = \frac{1}{6}(\Delta M_{p,1} + 2\Delta M_{p,2} + 2\Delta M_{p,3} + \Delta M_{p,4}) \quad (\text{B.2b})$$

and the quantities  $R_{p,1} - R_{p,4}$  and  $M_{p,1} - M_{p,4}$  are obtained as follows;

$$\Delta R_{p,1} = \left( 2.69 \frac{M_p g}{\pi R_p^2 \rho_l} \right)^{0.5} \Delta t \equiv R'(M_p, R_p) \Delta t \quad (\text{B.3a})$$

$$\Delta M_{p,1} = - \left( \frac{\pi R_p^2 h (T_s - T)}{L_{vap}} \right) \Delta t \equiv M'(R_p) \Delta t \quad (\text{B.3b})$$

$$\Delta R_{p,2} = R' (M_p + 0.5\Delta M_{p,1}, R_p + 0.5\Delta R_{p,1}) \Delta t \quad (\text{B.4a})$$

$$\Delta M_{p,2} = M' (R_p + 0.5\Delta R_{p,1}) \Delta t \quad (\text{B.4b})$$

$$\Delta R_{p,3} = R' (M_p + 0.5\Delta M_{p,2}, R_p + 0.5\Delta R_{p,2}) \Delta t \quad (\text{B.5a})$$

$$\Delta M_{p,3} = M' (R_p + 0.5\Delta R_{p,2}) \Delta t \quad (\text{B.5b})$$

$$\Delta R_{p,4} = R' (M_p + \Delta M_{p,3}, R_p + \Delta R_{p,3}) \Delta t \quad (\text{B.6a})$$

$$\Delta M_{p,4} = M' (R_p + \Delta R_{p,3}) \Delta t \quad (\text{B.6b})$$

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## ***C. Description of the Iterative Procedure for the Vaporizing Pool Model***

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The program structure used for the vaporizing pool model is depicted in Figure C.1. At the start of the simulation run, all initial conditions and parameter values are provided including initial spilled mass, initial radius, step size and run time. This input specification should also include either confined or unconfined spills. The H-COEFF() is used to calculate the film and transition

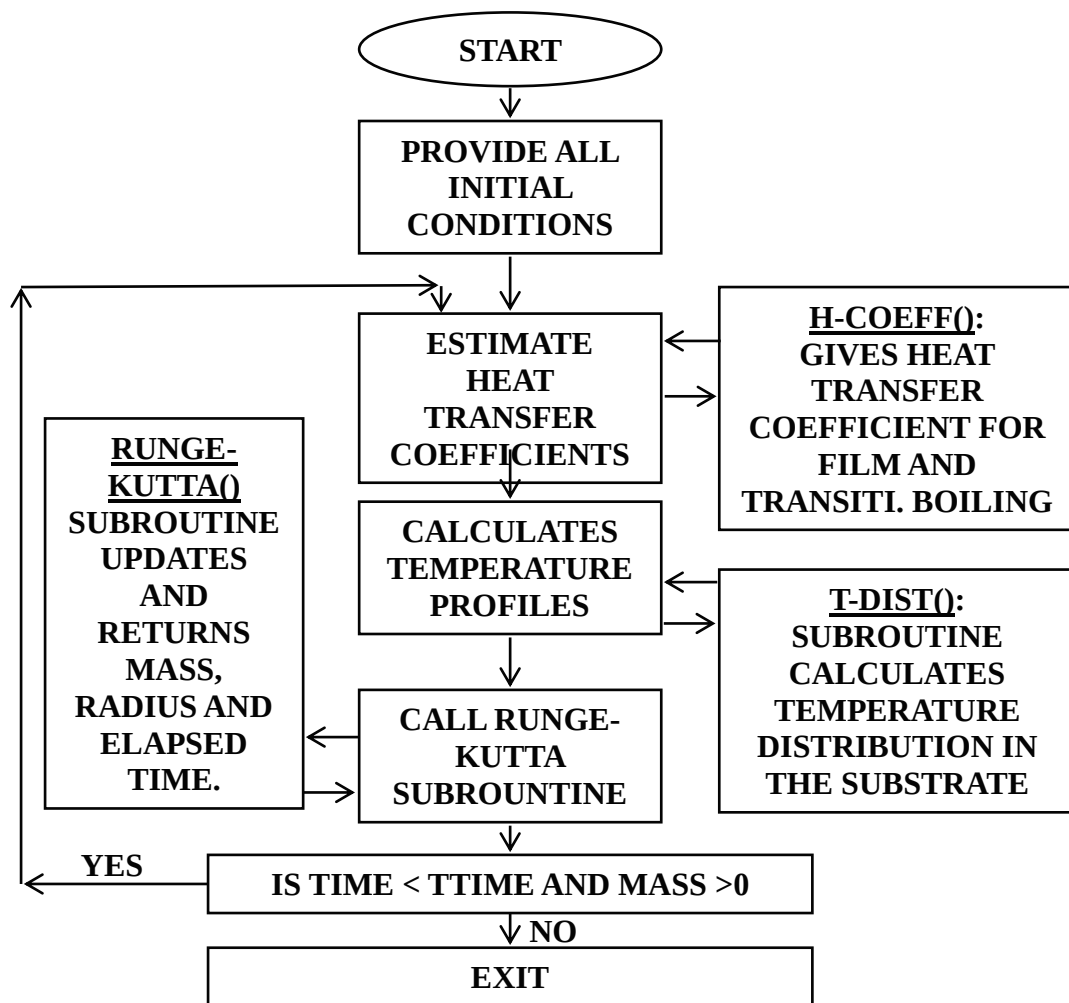


**Figure C.1:** A flow chart showing the program structure used in running the vaporizing pool model which predicts the spreading and vaporization rate of cryogenic liquids following spills on non-penetrable solid surfaces.

boiling heat transfer coefficient. The leidenfrost temperature or the superheat of the substrate is used to determine the point of switch from film boiling to transition boiling. This is followed by the calculation of the temperature distribution in the substrate including an estimation of the substrate surface temperature. This is achieved using the subroutine T-DIST(). Thereafter, the Runge-Kutta subroutine is called in order to evaluate the time-dependent changes in mass and radius. The Runge-Kutta method uses two functions to evaluate the coupled vaporization and spreading equations. Details of the Runge-Kutta method have been provided in Appendix B. After updating the mass, radius and the time elapsed, the iterative procedure continues until either the initial mass has been completely vaporized or the time elapsed has exceeded the total simulation time which has been specified at the start of the simulation. A CD containing full description of the source codes and algorithms is included as an appendix CD.

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## ***D. Chronological List of Minor and Major LNG Accidents***

Although the LNG industry has maintained an enviable safety record, there are still cases of minor and major LNG accidents and we have made a chronological list of some of these accidents in Table D.1.

**Table D.1:** A chronological list of some major and minor LNG accidents between 1944 and 2007 (Cabrillo, 2006; Choi et al., 2012).

| <i>Accident Date</i> | <i>Comment</i>                                                                                                                                                                                                                                                                                                                                                                                                                                     |
|----------------------|----------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|
| 1944                 | The East Ohio Gas LNG Tank located in Ohio, Cleveland is a small LNG facility where a tank failed and no earthen berm was available. Vapour cloud formed and filled the surrounding streets and storm sewer system. Natural gas in the vaporizing LNG pool ignited. This led to 128 deaths.                                                                                                                                                        |
| 1965                 | This accident occurred in Canvey Island, United Kingdom when LNG spilled during a transfer operation resulting in one person getting seriously burnt                                                                                                                                                                                                                                                                                               |
| 1965                 | In the same 1965, LNG spillage occurred at a Jules Verne facility as a result of overfilling that led to deck fractures.                                                                                                                                                                                                                                                                                                                           |
| 1965                 | LNG ship Methane Princess disconnected after discharge causing valve leakage and deck fractures.                                                                                                                                                                                                                                                                                                                                                   |
| 1971                 | LNG ship Esso Brega, located in La Spezia LNG Import Terminal in Italy. Tank developed a sudden increase in pressure during loading. LNG vapour discharged from the tank safety valves and vents. Tank roof slightly damaged. No ignition. This is the first documented LNG rollover incident.                                                                                                                                                     |
| 1973                 | In this Texas Eastern Transmission LNG facility, Staten Island, NY, US. During the repairs, vapours associated with the cleaning process apparently ignited the mylar liner. Fire caused temperature in the tank to rise, generating enough pressure to dislodge a 6-inch thick concrete roof, which then fell on the workers in the tank. Forty people were killed. Industrial incident unrelated to the presence of LNG (construction incident). |
| 1973                 | At Canvey Island, UK, a small amount of LNG spilled upon a puddle of rainwater, and the resulting flameless vapour explosion, called a rapid phase transition (RPT) led to Glass breakage.                                                                                                                                                                                                                                                         |
| 1974                 | At this Massachusetts facility, valve leakage and deck fractures occurred during a loading operation.                                                                                                                                                                                                                                                                                                                                              |
| 1974                 | LNG ship Methane Princess in port touched bottom in Arzew.                                                                                                                                                                                                                                                                                                                                                                                         |
| 1975                 | At this Philadelphia Gas Works LNG facility, an iso-pentane intermediate heat transfer fluid leak caught fire and burned the entire vaporizer area. However this accident is not caused by LNG.                                                                                                                                                                                                                                                    |
| 1977                 | LNG ship Aquarius overfilled while Loading.                                                                                                                                                                                                                                                                                                                                                                                                        |

|      |                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                    |
|------|------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|
| 1979 | Columbia Gas LNG Regasification Terminal located at Cove Point, Maryland, US. Killed one person and another person got seriously injured because of an explosion which occurred within an electrical substation. The problem started when LNG leaked through LNG pump electrical penetration seal, vaporized, passed through 200 feet of underground electrical conduit, and entered the substation. Since natural gas was never expected in this building, there were no gas detectors installed in the building. The normal arcing contacts of a circuit breaker ignited the natural gas-air mixture, resulting in an explosion. |
| 1979 | Mostefa Ben-Boulaid Ship experienced valve leakage and deck fractures during unloading.                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                            |
| 1979 | Pollenger Ship experienced valve leakage and deck fractures during unloading.                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                      |
| 1979 | El Paso Paul Kayser Ship while at sea got stranded and experienced severe damage to bottom, ballast tanks, motors water damaged, bottom of containment system set up.                                                                                                                                                                                                                                                                                                                                                                                                                                                              |
| 1979 | LNG ship grounded in Japan due to bad weather                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                      |
| 1979 | LNG ship water ballast tank leakage into insulation space of cargo tank                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                            |
| 1980 | LNG ship grounding in Gilbralter; 2500 tonnes of steel were damaged including deformation of cargo tank.                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                           |
| 1980 | LNG ship Libra while at sea fractured its tail shaft resulting in the shaft moving against the rudder.                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                             |
| 1980 | LNG Taurus in port got stranded and ballast tanks were all flooded causing extensive bottom damage.                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                |
| 1984 | The engine room of LNG ship Melrose caught fire and the fire was limited to engine room. No structural damage was sustained.                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                       |
| 1985 | LNG ship Gradinia while at port experienced steering gear failure but no details of structural damage were reported.                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                               |
| 1985 | LNG ship Isabella while unloading experienced cargo valve failure, cargo overflow and deck fractures.                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                              |
| 1989 | LNG ship Teller while loading broke mooring leading to hull and deck failures.                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                     |
| 1990 | LNG ship Bachir Chihani while at sea sustained structural cracks allegedly caused by stressing and fatigue in inner hull.                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                          |
| 1993 | At an Indonesian liquefaction facility, Indonesia, LNG leakage from open run-down line during a pipe modification project entered into an underground concrete storm sewer system and underwent a rapid vapour expansion that overpressured and ruptured the sewer pipes resulting in storm sewer system experiencing substantial damage.                                                                                                                                                                                                                                                                                          |
| 2002 | LNG ship Norman Lady located at East of the Strait of Gibraltar, while at sea collided with a U.S. Navy nuclear-powered attack submarine, the U.S.S Oklahoma City. In ballast condition. Ship suffered a leakage of seawater into the double bottom dry tank area.                                                                                                                                                                                                                                                                                                                                                                 |
| 2004 | LNG grounding in Korea due to navigation error.                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                    |
| 2004 | LNG ship failure due to cargo containment failure in secondary barrier.                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                            |
| 2004 | Skikda I, Algeria. Fire completely destroyed the train 40, 30, and 20, although it did not damage the loading facilities or                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                        |

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|      |                                                                                                                                                                                              |
|------|----------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|
|      | three large LNG storage tanks also located at the terminal. Complete details are pending until completion of ongoing accident investigation. Twenty-seven people were killed and 56 injured. |
| 2005 | LNG ship sloshing damage in an LNG heavy weather.                                                                                                                                            |
| 2006 | LNG sloshing damage during ballast voyage with minimum LNG.                                                                                                                                  |
| 2006 | LNG ship due to cargo containment failure in secondary barrier.                                                                                                                              |
| 2007 | LNG ship sloshing damage in an LNG heavy weather.                                                                                                                                            |

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