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Computational Modelling of Thermofluid Flashing in MSF Desalination

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A thesis submitted to the National University of Ireland as fulfilment of the requirements for the Degree of Doctor of Philosophy

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ABSTRACT

Phase change inside a desalination flashing chamber arises from the surface evaporation with self-boiling of liquid brine flow due to a reduction of pressure. Flashing flow is the key to desalination of the feed water in every flashing chamber in multi-stage flash (MSF) desalination systems. The flashing process in the evaporation zone of a flashing chamber is investigated. A new classification of the flashing process as ideal, infinite and finite is proposed.

A computational field model for the flashing process in the evaporation zone inside a flashing chamber is developed around a two-phase volume of fluid (VOF) formulation. Two different phase-change mechanisms are allowed for, based on the saturation temperature and on the vapour pressure, respectively, that enable the model to compute the phase change regions, also the level and the shape of the free surface. The commercial computational fluid dynamics (CFD) code Fluent is used. The model is validated using data for an existing flashing chamber by solving for steady, two-dimensional, multiphase flow inside a flashing chamber without a baffle.

The model is then applied to investigate the effects of variations in the inlet brine flow rate and inlet brine temperature, for both finite and infinite flashing process classifications. Thermo-fluid behaviour and flow details are predicted, and performance results are presented in terms of flow patterns and distributions of gauge pressure, temperature, vapour volume fraction and mass transfer inside the flashing chamber. The predictions are analysed to estimate MSF design factors such as the non-equilibrium temperature difference and flashing efficiency. It is found that the chamber evaporation zone behaves differently for finite and infinite flashing processes. As has been observed experimentally, the flashing performance and vapour production are improved when the brine flow rate and its temperature are increased.
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<tr>
<td>$A$</td>
<td>area $[m^2]$</td>
</tr>
<tr>
<td>$BPE$</td>
<td>boiling point elevation $[K]$</td>
</tr>
<tr>
<td>coeff</td>
<td>rate factor of vapourisation or condensation (inverse of a relaxation time) in Lee Wen Ho model (Eq. 4) $[s^{-1}]$</td>
</tr>
<tr>
<td>$c_p$</td>
<td>specific heat at constant pressure $[J/kg, K]$</td>
</tr>
<tr>
<td>$c_v$</td>
<td>specific heat at constant volume $[J/kg, K]$</td>
</tr>
<tr>
<td>$C$</td>
<td>concentration factor $[kg/kg_{sol}]$</td>
</tr>
<tr>
<td>$d$</td>
<td>vapour bubble diameter $[m]$</td>
</tr>
<tr>
<td>$E$</td>
<td>total energy $[J]$</td>
</tr>
<tr>
<td>$E_q$</td>
<td>represents the total energy for each phase (Eq.4.7) $[J]$</td>
</tr>
<tr>
<td>$f_{max}$</td>
<td>maximum bubble nucleation frequency (Eq. 3.5) $[s^{-1}]$</td>
</tr>
<tr>
<td>$\vec{F}$</td>
<td>external body force $[N]$</td>
</tr>
<tr>
<td>$F_{col}$</td>
<td>collapse coefficient [-]</td>
</tr>
<tr>
<td>$F_{vap}$</td>
<td>vapourisation coefficient [-]</td>
</tr>
<tr>
<td>$g$</td>
<td>gravitational acceleration $[m/s^2]$</td>
</tr>
<tr>
<td>$h$</td>
<td>height of inlet orifice to flashing chamber $[m]$</td>
</tr>
<tr>
<td>$h$</td>
<td>heat transfer coefficient $[kW/m^2.K]$</td>
</tr>
<tr>
<td>$h$</td>
<td>enthalpy $[kJ/kg]$</td>
</tr>
<tr>
<td>$H$</td>
<td>height of flashing chamber $[m]$</td>
</tr>
<tr>
<td>$J_{w,max}$</td>
<td>maximum nucleation rate [-]</td>
</tr>
<tr>
<td>$k^*$</td>
<td>thermal conductivity (Eq. 4.6) $[W/m , K]$</td>
</tr>
<tr>
<td>$k_{eff}$</td>
<td>effective conductivity (Eq. 4.5) $[W/m , K]$</td>
</tr>
<tr>
<td>$k_t$</td>
<td>turbulent thermal conductivity (Eq. 4.6) $[W/m , K]$</td>
</tr>
<tr>
<td>$L$</td>
<td>length of flashing chamber $[m]$</td>
</tr>
<tr>
<td>$LH$</td>
<td>latent heat $[J/kg]$</td>
</tr>
<tr>
<td>LMTD</td>
<td>logarithmic mean temperature difference $[K]$</td>
</tr>
<tr>
<td>$m$</td>
<td>mass flow rate $[kg/s]$</td>
</tr>
<tr>
<td>$m_{l\to v}$</td>
<td>mass transfer rate from liquid to vapour phase $[kg , m^3/s]$</td>
</tr>
<tr>
<td>$m_{v\to l}$</td>
<td>mass transfer rate from vapour phase to liquid $[kg , m^3/s]$</td>
</tr>
<tr>
<td>$m$</td>
<td>mass $[kg]$</td>
</tr>
<tr>
<td>$M$</td>
<td>molecular weight $[kg/kmol]$</td>
</tr>
<tr>
<td>$n$</td>
<td>bubble density number [-]</td>
</tr>
<tr>
<td>$\vec{n}$</td>
<td>unit vector, outward-normal to $S$ [-]</td>
</tr>
<tr>
<td>NETD</td>
<td>non-equilibrium temperature difference $[K]$</td>
</tr>
<tr>
<td>$Pr$</td>
<td>Prandtl number [-]</td>
</tr>
<tr>
<td>PR</td>
<td>performance ratio [-]</td>
</tr>
<tr>
<td>$p_s$</td>
<td>gauge pressure $[Pa]$</td>
</tr>
<tr>
<td>$p$</td>
<td>absolute pressure $[Pa]$</td>
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\( p_{op} \) flashing chamber operating pressure
\( p_v \) vapour pressure [Pa] (indication of a liquid’s evaporation rate)
\( P \) total pressure [Pa]
\( q \) heat flux [W/m²]
\( Q^\prime \) flashing heat flux (Eq. 3.7) [kW/m²]
\( \dot{Q} \) heat transfer term (Eq. 2.3) [W]
\( r_c \) critical radius of the bubble [m]
\( R \) universal gas constant [ J/kmol K]
\( R' \) rate of mass change of a single bubble [kg/s]
\( R_b \) bubble radius [m]
\( R_c \) mass transfer source term, bubble collapse (Eq. 7) [ kg m³/s ]
\( R_e \) mass transfer source term, bubble growth (Eq. 7) [ kg m³/s ]
\( S \) control surface [ m²], entropy [kJ/kg °C]
\( S_{sh} \) source term contains contributions from radiation and any other volumetric heat sources (Eq. 4.5) [-]
\( t \) time [s]
\( T \) temperature [K]
\( T_{brine} \) brine temperature [K]
\( T_{in} \) inlet temperature [K]
\( T_{liq} \) temperature of the liquid phase [K]
\( T_{out} \) outlet temperature [K]
\( T_{sat} \) saturation temperature [K]
\( T_{v} \) temperature of the vapour phase [K]
\( TBT \) top brine temperature [K]
\( TTD \) total terminal temperature difference [K]
\( ^{\hat{}} u \) internal energy (Eq. 2.3) [J]
\( U \) Overall heat transfer coefficient [W/m²K]
\( \Psi_r \) velocity of control surface [m/s]
\( v_x \) component of velocity in \( x \)-direction [m/s]
\( \vec{\Psi}_v \) velocity vector of the vapour phase [m/s]
\( v_y \) component of velocity in \( y \)-direction [m/s]
\( \dot{W}_c \) shear work term (Eq. 2.3) [W]
\( \dot{W}_v \) viscous work term (Eq. 2.3) [W]
\( w \) width of flashing chamber orifice [m]
\( W \) width of flashing chamber [m]
\( x \) horizontal coordinate [m]
\( X \) salinity [ppm]
\( y \) vertical coordinate [m]
\( y^* \) dimensionless wall-normal distance \( \approx y \sqrt{\rho \tau_w / \mu} \) [-]
\( Z \) vertical upward coordinate (Eq. 2.3) [m]
Greek
\(\alpha\) volume fraction [-]
\(\alpha_l\) liquid phase fraction [-]
\(\alpha_{nuc}\) nucleation site volume fraction [-]
\(\alpha_v\) vapour phase fraction [-]
\(\beta\) parameter in Lee Wen Ho phase-change model (Eq. 5) [m/s]
\(\Gamma\) diffusion coefficient (Eq 4.4) [-]
\(\varepsilon\) dissipation rate of turbulence kinetic energy [m²/s³]
\(k\) kinetic energy of turbulence [m²/s²]
\(\mu\) viscosity [Pa s]
\(\rho_l\) density of liquid [kg/m³]
\(\rho_v\) density of vapour [kg/m³]
\(\sigma\) surface tension [N/m]
\(\tau_w\) wall shear stress [Pa]
\(\Omega\) control volume [m³]

Subscripts
\(b\) bubble
\(br\) brine
\(cs\) control surface
\(cv\) control volume
\(d\) desalinated water
\(i\) number of the stage (flashing chamber) in MSF plant
\(j\) index which increases by 1 for each successive term
\(in\) inlet
\(l\) liquid phase
\(mix\) mixture
\(nuc\) nucleation
\(out\) outlet
\(op\) flashing chamber operating pressure
\(v\) vapour
\(r\) recycled brine
\(sat\) saturation
\(z\) total number of phases in the system

Abbreviations
CFD computational fluid dynamics
MSF multi-stage flash
VOF volume of fluid
1. INTRODUCTION

1.1. Objective and Thesis Overview

The multi-stage flash (MSF) desalination system is one of the most widely used. Focusing on thermal desalination techniques, understanding the system process, system components, mathematical models and system performance is the first objective of the research.

Our previous research interest was to design and develop a solar-powered seawater desalination system suitable for use in naturally sunny regions that lack other sources of energy supply, and to design a solar-powered device to improve the MSF performance. This clarifies why we focused on the solar desalination and the solar technologies.

The second objective is to investigate the flash evaporation process inside the flashing chamber in MSF. For this purpose, a computational model for the flashing process inside a flashing chamber is developed and validated. The model is applied to predict the multiphase flow field, based on the FLUENT volume of fluid (VOF) code implementation, and encapsulates two interaction mechanisms for phase change — based on saturation temperature and vapour pressure, respectively — that arise during the flashing process and thus to map the flow patterns and behaviour inside the chamber. Predicted results are compared with regard to thermo-fluid performance; gauge pressure, flow patterns, thermal performance, vapour volume fraction and mass transfer inside the flashing chamber.

Investigations of the effects of inlet flow rate and the inlet brine temperature on the thermo-fluid performance of a flashing chamber, without a baffle,
operating under infinite and finite flashing condition are the final objective of this research.

Firstly, by way of introduction and to place the study context, an overview of several topics is presented: global water, world energy, existing technologies for desalination and solar collector systems, as well as classifications of desalination technologies and solar technologies in Chapter 1.

Increasing global water demand, combined with falling supplies both of fresh water and of sources of non-renewable energy, is one of today’s more critical engineering challenges.

The provision of fresh water by means of applying a desalination process to seawater requires a large quantity of energy to be supplied. One promising solution, which deals with both the challenge of water and energy, is a solar-powered desalination system.

In this Chapter, a general overview of several topics is presented: global water, world energy, existing technologies for desalination and solar collector systems, as well as classifications of desalination technologies and solar technologies.

1.2. Global Water Supply

Water is the basis of life on earth, and clean water is an essential natural resource. It is the most important chemical compound [1]. The inability to access clean, safe drinking water can directly and negatively impact human life and the environment.

Every year, millions of adults and children die — 3,900 children per day [2] — from diseases which are caused by consuming contaminated drinking water. Over 2.6 billion people have little or no sanitation; more than 1.2
billion people suffer because of a lack of access to clean and safe water for drinking [3]. According to the World Health Organisation (WHO), 20% of the world’s population has little or no access to clean water and proper sanitation facilities [4].

In the developing world, the use of unclean water causes 80 to 90% of all diseases, and more than 30% of all deaths can be linked to the consumption of unsafe and unclean water [5]. A million and a half children die each year because of waterborne diseases [6]. In developing countries, more than 90% of raw sewage and 70% of untreated industrial wastes are dumped into surface water [7].

The exponential growth of population of these developing nations further increases pressure on limited water sources available [8]. Based on the data that is received from global population statistics and global freshwater withdrawals statistics [9, 10], a 0.5% increase in the global population leads to a 1.5% increase in global freshwater withdrawals. Figure 1.1 demonstrates the global increase in the population and water demand.

Market research states [11] that the annual demand for fresh water is increasing by 3% per annum, and this would translate into annual investment requirements of up to €400–500 billion in water infrastructure. In the meantime, Population Action International (PAI) states that more than 3 billion people, in more than 48 countries, are going to face water stress or scarcity conditions by 2025 [12].

The water situation around the world is critical, and is becoming more serious. The main question is how it can be improved? Before answering this question, certain topics need to be covered, e.g. how much water is available on Earth? How much water does a human ideally need/consume?
Figure 1.1  Global water consumption versus population growth - The relationship between global population and global fresh water withdrawals (source data [9, 10])
What is the fresh water quality? What is the impact of water quality on people and the environment?

### 1.2.1. Water availability and human requirements

The total amount of water on earth is circa 1,386,000,000 cubic kilometres (km³) [13], but only 3% of the total amount of that figure for water is considered fresh water *Figure 1.2* [14, 15] and only 2.5% [16, 17] of that 3% is available for drinking or human consumption. To equate a cubic metre of water to a tangible volume, it is equivalent to the volume of water required for: 13 baths, or 14 washing machine loads, or 28 showers, or 33 dishwasher loads, or 111 toilet flushes [18].

The Institute of Medicine defined the average drinking water required per person per day as follows. An adult male requires circa 3.5 litres per day, while an adult woman requires 2.5 litres per day [19, 20]. Note, the residential use of water varies from country to country *e.g.* the rate of using water in North America is circa 400 litres per day per person, and in Europe is circa 200 litres per day per person [5].

### 1.2.2. Water quality

Water quality is determined via the concentration of chemically dissolved materials in the water [1]. The World Health Organisation (WHO) and United States Environmental Protection Agency [1, 21, 22] state that the maximum allowable concentration for total dissolved salts in water for human consumption is 500 ppm and the maximum limit for chloride ions is 250 ppm.
Figure 1.2  The distribution of water on earth (source data [14, 15])
1.2.3. **Water impacts on human life**

The shortage of water and prevalence of human diseases via contaminated water supplies causes considerable problems of human life sectors, evident in different situations as; health, education, environment, and economic considerations. For more details the following references are recommended [22 - 29].

In short, water is seen as a health indicator. Safe water has many impacts and consequences on societies. ‘Many of the wars in this century were about oil, but wars of the next century will be over water’ – the World Bank Vice President in 1995 [29].

Water is in constant demand, but the total amount of water on earth is fixed. Global water demand and population are constantly increasing. These increases on both sides are almost impossible to stop. Thus, finding a potential solution for future water supply is a large undertaking. Water desalination [8] has the opportunity to be one of the most effective and efficient solutions. Desalination of saltwater is energy-intensive. Here, the issue of "appropriateness" of energy technology must be considered.

1.3. **Energy**

Continuous changes in human lifestyle leads to a considerable increase in energy demand. Globalisation, urbanisation, suburban sprawl, consumerism, expansion of production, and recycling all converges to an increase in demand for energy. Thus, energy demand is increasing sharply. The recent World Energy Council (WEC) [30] report states that by 2020, the world energy demand will increase by 50 – 80% more than the 1990’s level. BP’s statistical review of world energy [31] states that by 2035, the world energy
demand will increase by 60% more than the 2014’s level. This portends imminent energy problems on the horizon.

Thermodynamically, energy cannot be created or destroyed, but it can be converted from one form to another. In nature, energy is available in many different forms and can be divided into renewable and non-renewable energy. The following references go into detail on renewable [32-44], renewable energy and availability [45] and the effects of renewable sources in today’s society [46-57].

Renewable energy has the potential to be one of the main vehicles tackling the global issue of climate change, thus protecting and safeguarding the planet’s future for the generations to come. The management of finite non-renewable energy sources and the development of technologies to utilise infinite renewable energy sources is a feasible method of energy management, which will resolve issues such as climate change.

In summary, increasing global water demand, combined with falling supplies both of fresh water and of sources of non-renewable energy, is one of today’s more critical engineering challenges. Removing the salt from seawater using energy radiated from the sun is a potential solution to both problems. Thus, solar powered desalination can be considered as one of the potential solutions of the water and the energy problems. In the following section, a review of desalination technologies and solar collectors systems are presented.

1.4. Solar Powered Desalination

This section is divided into two main parts:
1- Desalination technology
2- Solar systems
1.4.1. Desalination

Desalination is a process to remove the salts and other minerals from seawater or any brackish water for producing fresh and clear water. Water is considered potable (drinkable) when salinity is below 500 ppm (Table 1.2).

<table>
<thead>
<tr>
<th>Type</th>
<th>Total Dissolved Solids (TDS)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fresh Water</td>
<td>Up to 1,000</td>
</tr>
<tr>
<td>Brackish Water</td>
<td>1,000-10,000</td>
</tr>
<tr>
<td>Salt Water</td>
<td>&gt;10,000</td>
</tr>
<tr>
<td>Seawater</td>
<td>10,000-50,000</td>
</tr>
<tr>
<td>Standard Seawater</td>
<td>35,000</td>
</tr>
</tbody>
</table>

1.4.1.1. Desalination history

Desalination has been practiced for more than a thousand years. One of the first references is found in Bible, Exodus 15:22-27. It stated the concept of producing water by desalination; fresh water can be produced from salt water. In 460-370 B.C., Hippocrates II [60] states that boiling water leads to potable water, and use of the sun’s radiation leads to provide clean water. In 1627, Francis Bacon [61] discovered that salt does not rise in vapour by evaporating seawater [62]. In 1750, the first thermal desalination systems [1] were built and used on ships for providing fresh water during sea journeys. In 1851, commercial desalination systems started to develop [1]. At that time, all systems were thermal desalination systems. In 1970, reverse osmosis (RO) membranes were developed for commercial purposes [63].

Table 1.3 summarises the historical developments of thermal and membrane desalination systems. This review in the table highlights the main
achievements which have taken place in the thermal and membrane desalination area.

**Table 1.3** Historical developments of thermal and membrane desalination systems

<table>
<thead>
<tr>
<th>Year</th>
<th>Achievement</th>
<th>Ref(s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1627</td>
<td>Francis Bacon discovered that salt does not rise in vapour by evaporating seawater</td>
<td>[61]</td>
</tr>
<tr>
<td>1790</td>
<td>Thomas Jefferson described a practical shipboard method in a report on converting saltwater to fresh</td>
<td>[62]</td>
</tr>
<tr>
<td>Early 1930’s</td>
<td>Several small seawater desalination systems were built in the Middle East</td>
<td>[1]</td>
</tr>
<tr>
<td>Late 1940’s</td>
<td>Development of oil in the Arabian Gulf gave a push to the desalination industry</td>
<td>[1]</td>
</tr>
<tr>
<td></td>
<td>Evaporator known as a flash evaporator</td>
<td>[62]</td>
</tr>
<tr>
<td>1948</td>
<td>The first patent issued for flash evaporators by J. Kirgan</td>
<td>[64]</td>
</tr>
<tr>
<td>Up to 1950’s</td>
<td>The largest desalination unit ever built had an output of less than 1,892 m³/d</td>
<td>[63]</td>
</tr>
<tr>
<td>1952</td>
<td>Worthen and Barbour obtained a patent on a two flashing chamber evaporator on the basis that the thermal economy was increased</td>
<td>[65]</td>
</tr>
<tr>
<td>1953</td>
<td>US Navy carried out a test for a five flashing chamber evaporator</td>
<td>[66]</td>
</tr>
<tr>
<td>1957</td>
<td>First industrial-scale flashing chamber by Westinghouse in Kuwait. Four flashing chamber system (3.3 performance ratio)</td>
<td>[63]</td>
</tr>
<tr>
<td>1957</td>
<td>Silver patent for multi-stage flash (MSF) configuration</td>
<td>[63]</td>
</tr>
<tr>
<td>1960</td>
<td>Silver obtained a patent for stating that a number of flash stages equal to, or greater than twice the thermal economy of the flash plant</td>
<td>[67]</td>
</tr>
<tr>
<td></td>
<td>First MSF plants commissioned in Shuwaikh, Kuwait (19 stages, 440 m³/d capacity, and performance ratio 5.7) and in Guernsey, Channel</td>
<td></td>
</tr>
</tbody>
</table>
island (40 stages, 2,775 m³/d capacity, and performance ratio 10). [63]

1962 Point Loma MSF plant capacity 4,546 m³/d [63]

1965 Deaeration of feed stream. This was used to heat feedwater and to remove oxygen and carbon dioxide from feedwater. This improved flashing chamber efficiency by increasing the feedwater temperature, and reduced the corrosion caused by oxygen and carbon dioxide. [63]

1966 Reduction in specific volume for reducing the capital cost [63]

1967 Weir Westgarth provided the first on-line ball cleaning system in the Bahamas. [63]

1967 Acid cleaning [63]

1969 Co-generation (50% reduction in energy cost) [63]

1969 Load factor of thermal & membrane desalination plants increased by 85% [63]

1970 Commercial RO membranes development [63]

1973 Cladding of partition walls for flashing chambers inside MSF systems. This reduced flashing chambers corrosion and increased their lifetime. [63]

1973 Construction of standard MSF units (36,368 m³/d, 24 stages, and performance ratio 6-8) [63]

1980 Design and run low-temperature multiple effect evaporation units combined with thermal compression [63]

Design and run low-temperature mechanical vapour compression units [63]

1985 Polymer antiscalant (chemical injection into feedwater) is used at top brine temperature of 110 ºC [63]

1996 Construction of the largest MSF unit in UAE (57,735 m³/d) [63]

1999 Construction of a large-scale RO plant in Florida, USA [63]
A comprehensive study of information and data pertaining to the history of desalination technology were reviewed in many books and reports e.g. Khan [1], El Dessouky et al. [63], Lienhard et al. [21], Lior et al. [71], and international desalination association (IDA) annual reports [70]. In the following section, we highlight the main desalination technologies; membrane and thermal.

### 1.4.1.2. Desalination technology

Desalination processes may be classified in many ways. They have been classified by method of separation as thermal (with phase change) or membrane (without phase change) methods (Figure 1.3) [59], or according to the form of the energy used to drive the process (Figure 1.4) [63].
Figure 1.3 Thermal and membrane desalination processes (source data [1, 63, 71])

Figure 1.4 Energy classification of desalination processes (source data [1, 63, 71])
There are many types of desalination technologies for producing fresh water. The most common desalination systems by global capacity of plant (Figure 1.5) are: reverse osmosis (RO), multi-stage flash process (MSF), and multi-effect distillation (MED). Relative power requirements of these technologies are represented in Figure 1.6.

**Figure 1.5** Global desalination plant capacity by technology in 2014 (source data [70, 72])

**Figure 1.6** Power requirements for different desalination technologies (source data [73])
In general, the desalination system procedure requires four steps. The basic desalination steps are: pumping of salty water, pre-treatment of pumped water, a desalination process, and post-treatment. In this study, the focus is on the desalination step of the process. In the following section we highlight the main membrane desalination technology reverse osmosis.

1.4.1.2.1. Reverse osmosis (RO)

The reverse osmosis mechanism is the opposite of osmosis. Osmosis is a natural process [1]; it is an equilibrium phenomenon, [74] which includes a semipermeable membrane (porous body) that allows small molecules to pass through but not macromolecules [74] (as shown in Figure 1.7). Figure 1.7 is schematic of the osmosis process. Water flows through semipermeable membrane from a low concentration solution (pure water) to a high concentration solution (salt solution) until it reaches the osmotic equilibrium.

To reverse the osmosis process, an external pressure is needed to be applied on the high concentration solution (salt solution) with a value greater than the osmotic pressure.

Producing fresh water by using reverse osmosis requires pumping saline feedwater, a high total dissolved solids (TDS) solution, from high pressure through permeable membranes to produce a solution with a low TDS [75, 76] (see Table 1.4), thereby separating the salts from the water. Figure 1.7 shows a schematic of the reverse osmosis process.

Table 1.4  An example of applied pressures for typical membrane filtration processes [75]

<table>
<thead>
<tr>
<th>Membrane processes</th>
<th>Applied pressure (atm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>RO seawater</td>
<td>54.4 – 68.0</td>
</tr>
<tr>
<td>RO waste and process</td>
<td>20.4 – 40.8</td>
</tr>
<tr>
<td>RO water purification</td>
<td>13.6 – 23.8</td>
</tr>
</tbody>
</table>
Figure 1.7  RO desalination process

The quality of freshwater produced by RO depends on many factors including applied pressure, concentration of salts in feedwater and also membrane type [75, 76]. The passing of water through membranes for a second time is used to enhance and improve the quality of water produced.

Low energy consumption of pumping saline feedwater is a major feature of RO technology, as well as low thermal impact discharges, fewer corrosion problems and smaller construction size relative to other desalination processes. On the other hand, the major disadvantages are sensitivity to the quality of the feedwater, membrane fouling, and low fresh water production compared with desalination plants run by a thermal approach.

1.4.1.2.2 Thermal desalination

Thermal desalination techniques can be divided into two principal categories. The first category consists of two main steps: water evaporation followed by water-vapour condensation [1, 63, 71] (Figure 1.8).
Figure 1.8  Schematic of a thermal desalination system

The second category involves freezing, followed by melting of the formed water ice crystals [63].

The first category is the most commonly used in industrial desalination. Thermal desalination plants can operate as dual-purpose, for generation of electricity and production of fresh water. This mainly happens in desalination plants such as multi-stage flash or multi-effect desalination.

Multi-stage flash (MSF)

Multi-stage flash (MSF) distillation is one of the most common techniques in use in large scale. [75]. Typically, they are built as cogeneration plants for both water and power production. [77].

The fundamental principle in the multi-stage flash process consists of boiling the feedwater and subsequently condensing the produced vapour to produce distilled water. The idea of this system is to reduce the pressure to allow the feedwater to boil at temperatures lower than the normal water boiling temperature. This happens inside a series of vessels known as stages.
There are two types of MSF processes: once-through MSF and brine-circulation MSF. Brine circulation process is the industrial standard. *Figures 1.9 and 1.10* are schematics of the process.

*Figure 1.9*  Schematic of flashing chamber desalination system

The main advantages of MSF are [76]: high distilled water quality, simplicity of operating, fewer pretreatments, and stages ability to operate semi-operation during their cleaning period. The major disadvantage is high energy consumption required when compared with the reverse osmosis process.
Figure 1.10  MSF desalination system
Multi-effect desalination (MED)

Multi-effect desalination is one of the oldest desalination technologies. The fundamental principle of this process is to employ the latent heat of condensation vapour from the first cell to provide heat to the second cell. **Figure 1.11** is a schematic of the process. The function of the thermo-ejector (steam jet pump) is to entrain the remaining part of the vapour from the stages [63], where it is compressed by the motive steam into the required pressure and temperature.

There are many advantages of this system [76]: high distilled water quality, lower energy consumption compared with MSF, and lower operation temperature, which reduces scaling and energy costs. On the other hand, the main disadvantages of this system are slow response to water demand changes and a lower capacity, *e.g.* per unit of plant volume, than MSF.

![MED desalination system](image)

**Figure 1.11** MED desalination system
1.4.2 Solar desalination

We should mention here, that our research interest was to design and develop a solar-powered seawater desalination system suitable for use in naturally sunny regions that lack other sources of energy supply, and to design a solar-powered device to improve the MSF performance. This clarifies why we focused on the solar desalination and the solar technologies.

Desalination of seawater is energy-intensive. Using typical MSF desalination technology to produce one cubic metre of fresh water requires more than 250 kJ/kg (thermal energy), and 3.5 kWh/m³ (electrical energy) [70]. Desalination by using a clean source of energy is gradually growing. It becomes economically attractive [21] as the cost of renewable technologies continues to decrease, and the cost of fossil fuels continues to increase [79].

Many studies [80-86] show that solar desalination is a successful technology to produce clean water. Tokui et al. [85] stated that running 40-50 % of the total power consumption of the desalination plants by using a renewable energy source can be enough to attain sustainable operation. Globally, however, at present less than 1% of the total water desalination uses a renewable energy source [79].

1.4.3 Solar energy

Applying the radiation of the sun, solar energy, for desalination systems can be summarised in three main ways [71]:

1. Solar radiation can be used to vaporise water directly at a moderate temperature, slightly above ambient, by using solar stills. This is the most commonly considered method and it requires very little mechanical or electrical energy.
2. Solar radiation can be used to generate steam or higher-temperature heat by using solar collectors or ponds. The generated heat is transferred to the evaporator of MED or MSF systems.

3. Solar radiation can be used to generate electricity by using either steam or photovoltaic cells. The generated electricity can be used for several desalination systems *i.e.* RO, MVC, ED.
1.4.2.1. Solar power technology

Solar technologies can be represented by two main groups: concentrating solar power (CSP) and photovoltaic (PV). Growing interest and developments are making both technology classes even more attractive. The main CSP technologies are parabolic trough and linear Fresnel reflector systems, central tower receivers, and the dish/engine arrangement [87].

The principle of CSP systems is to generate power using a vapour-driven turbine. CSP plants can be designed with a capacity of 5 MW to several hundred MW [74]. Figure 1.12 shows the principle of operation of a concentrating solar collector and a CSP plant.

*Figure 1.12* Principle of a concentrating solar collector (right and expanded) and a concentrating solar thermal power station for co-generation of electricity and process steam (left) (source data [77])
There are many factors in choosing a suitable solar collector system including application types, total required capacity, and storage options available. On choosing a collector system, the number of collectors is limited by available land. Furthermore, geographical location is the most important factor. This arises from the many critical parameters dependent on it, e.g. peak solar efficiency and concentration ratio.

Here is an example of how much one of these factors can affect a system. Using the sun tracking system instead of a fixing system in the solar still desalination application increases productivity by 22% and overall efficiency by 2% [86, 88].

Concentrating solar power systems [77] have advantages and disadvantages, *i.e.*:

- **Parabolic trough system:** long lifetime and durability. Limited temperature for heat transfer fluid, complex structure and requires level land area.
- **Linear Fresnel system:** simple structure and direct steam generation. This system needs more storage for direct steam generation.
- **Solar power tower:** high power cycle efficiency and ability to power combined cycles. However, high cost for maintenance and equipment.
- **Dish Stirling engine:** high power cycle efficiency and high modularity. However, high complexity, and no storage is available.

*Table 1.5* shows a comparison of concentrating solar power technologies.

The main disadvantages in using solar energy for desalination are cost, weather dependence and a lot of used space.
Table 1.5  Comparison of concentrating solar power technologies [77, 88]

<table>
<thead>
<tr>
<th>Technology</th>
<th>Linear Fresnel System</th>
<th>Parabolic Trough System</th>
<th>Solar Power Tower</th>
<th>Dish Stirling Engine</th>
</tr>
</thead>
<tbody>
<tr>
<td>Capacity range (MW)</td>
<td>5 - 250</td>
<td>10 - 250</td>
<td>10 - 100</td>
<td>0.1 - 1.0</td>
</tr>
<tr>
<td>Peak solar efficiency (%)</td>
<td>15</td>
<td>21</td>
<td>&lt; 20</td>
<td>31</td>
</tr>
<tr>
<td>Investment cost (€1,000/kW)</td>
<td>2.5 – 4.5</td>
<td>3.5 – 6.5</td>
<td>4 - 6</td>
<td>6 - 10</td>
</tr>
</tbody>
</table>

1.4.3.2 Flat plate photovoltaic

Photovoltaic is considered as a second type of solar technology. It can be classified as following [89]: stand-alone photovoltaic system, photovoltaic system for vehicle applications (solar vehicles), grid-connected photovoltaic system and building systems.

Photovoltaic converts solar energy into electricity. It uses semiconducting materials, e.g. silicon, to convert solar radiation to electrical power. Thus, the mechanism of a photovoltaic device is based on sunlight and electrons within the silicon. Power is produced when sunlight strikes the solar panel. The sunlight induces a flow of electrons in the silicon, which moves around and finally flows through wires built into the solar panel as an electrical current.

The cost of using flat plate photovoltaic is slightly high [90] but it is coming down and its markets are expanding at the present time [86]. This leads to focus more on the cost of producing freshwater by using a renewable energy sources, thus, an overview of solar-powered seawater desalination system production cost is presented in the following section.
1.4.4 Cost overview

Recently, seawater desalination has become more widespread in the coastal industry. During the last few years, the production capacity of all seawater desalination plants in the world has increased by 30% [71].

Only 1% of total desalination water is based on energy from a renewable source. Desalinated water market prices are between $1/m^3 and $2/m^3, and the cost of desalination is $0.5/m^3 [91]. The cost of using renewable energy in a desalination application is considered a major challenge. However, studies [78, 92] state that in the coming years, the cost of using renewable energy will decrease rapidly. Figure 1.13 shows cost reduction for concentrating solar power by years. These expectations make running the desalination plants by renewable energy sources more economically attractive.

![Figure 1.13](image)

Figure 1.13  Cost reduction for concentrating solar power by years (source data [94, 95])
Multi-stage flash desalination, specifically, has a strong position in the market share, with more than 60% of the market of world desalination [91]. The cost of renewable energy is a major factor affecting renewable desalination. The unit cost range for freshwater produced using MSF generated by a solar energy source, is from 1.785 to 2.84 $/m³ [78, 92, 93]. 

*Figure 1.14* shows a graph of unit water cost trends by MSF process.

A comprehensive review on the challenges and potential applications of choosing a renewable energy to drive proper desalination technologies can be found in Ghaffour et al. (2015) [97], Sharon. *et al.*(20015) [98] and Reif *et al.* (2015) [99].

After drawing a general look at desalinations systems, multi-stage flash is the most attractive option, especially the brine circulation MSF type. Focusing on thermal desalination techniques, understanding the system process, components and performance and the development of a mathematical model are the key research objectives of this study.

*Figure 1.14*  Unit water cost trends MSF processes (source data [96])
1.5. Organisation of This Thesis

In Chapter 2, comprehensive review of mathematical, thermodynamic, physical and material aspects are presented for the MSF system in general and for the flashing chamber in particular. Both MSF system and flashing chamber design parameters (e.g. performance ratio, specific flow rate of recirculated brine, heat transfer coefficient) are defined.

Chapter 3 presents the importance of the evaporation zone and its phase change mechanisms with new classifications of the flashing process i.e. flashing process inside the flashing chamber may not be fully completed (infinite) or may be completed (finite) somewhere inside the flashing chamber. Then, a comprehensive description of flashing with thermo-physics explanations is provided. A comprehensive review of previous work on the flashing process follows, wherein the following topics are treated: understanding the phenomena, mathematical models and correlations, experimental works, methods for improving the flashing process, and computational studies is provided.

In Chapter 4, a computational model of the flashing process inside a flashing chamber is developed and validated, insofar as possible given the sparsity of measurements available.

In Chapter 5, the computational model is applied under two different operating conditions. The first is where the flashing chamber operates under infinite flashing conditions, which is the normal operating condition in existing MSF systems. The second is where the flashing chamber operates under a finite flashing condition.

For each case, the thermodynamic process parameters of heat transfer rate, mass transfer rate, temperature distribution and vapour volume fraction
inside the evaporation zone are provided. The fluid process parameters of particle paths (streamlines in steady flow) of flashing flow and gauge pressure distribution inside the evaporation zone are studied. Non-equilibrium losses, flashing down and flashing efficiency are calculated.

Chapter 6 and 7 investigate the effects of varying the inlet flow rate, and inlet brine temperature, respectively, on the thermo-fluid performance of a flashing chamber, without a baffle, operating under an *infinite* flashing condition.

Chapter 8 and 9 investigate the effects of varying the inlet flow rate, and inlet brine temperature, respectively, on the thermo-fluid performance of a flashing chamber, without a baffle, operating under a *finite* flashing condition.

An overall summary, conclusions and recommendations for future work are presented in Chapters 10, 11, and 12 respectively.

The computational method presented in the thesis is applicable to assisting in the design procedure for MSF systems and in the investigation of improving the flashing performance in the flashing chambers. Our method is capable of predicting the fully three-dimensional, time-dependent behaviour of the evaporation zone, which exhibits oscillating, unsteady flow phenomena characteristic of open channel flows. The method can be applied on other systems for different applications *e.g.* multi-effect desolation (MED), nuclear applications.
2. MULTI-STAGE FLASH (MSF)

2.1. Introduction

Thermal desalination, particularly the brine circulation multi-stage flash (MSF) type, is one of the most attractive desalination systems, particularly the brine circulation multi-stage flash (MSF) type. Understanding MSF process, components, mathematical model and system performance are presented in the first section of this Chapter.

The flashing chamber is the space where the feedwater is desalinated in an MSF system. It can be one of the most important components of improving MSF performance since it has the highest exergy loss in the system. It is one of the main factors which affect the cost of desalination production as well.

In the second section of this Chapter, all of the details of a flashing chamber in MSF desalination system and analysis of the system from different point of views; mathematical, thermodynamic, physical and material viewpoints are presented. For further understanding of flashing chambers design parameters (e.g. performance ratio, specific flow rate of recirculated brine, heat transfer coefficient) and MSF system.
2.2. Multi-Stage Flash Process Overview

The multi-stage flash (MSF) desalination system is one of the most widely used [1]. Furthermore, it is considered as the most common and simple technique in use [100]. It is preferably to be built as a cogeneration plant for concurrent production of both power and fresh water [77, 101]. Recently, MSF accounted for 22% of total global desalination [70, 72].

The fundamental principles in the multi-stage flash process consist of producing vapour from the feed brine (feed seawater), and subsequently condensing the vapour to give the final distilled water. The mechanism behind the process is produced by changing the phase of the feed brine, by means of a reduction in the feed brine pressure below its vapour pressure. This leads to a phase change at a lower boiling point temperature than normal water boiling temperature. This process occurs inside each of a series of flashing chambers known as stages or cells.

There are two types of MSF process [69]; once-through MSF (MSF-OT) and brine circulation MSF (MSF-BC). Brine circulation (Figure 1.15) is one arrangement of a brine circulation MSF desalination system used in large scale industrial desalination plants. Various other arrangements can be found in references [1, 63, 76, 100]. Appendix A shows a set of typical images of a MSF plant and its elements. These pictures were taken from the MSF plant Al Khobar II in Saudi Arabia in 1983.

A typical MSF system [1, 76] is divided as in Figure 1.15 into three main sections; a brine heater, a heat recovery section, and a heat rejection section. The intake brine (feed seawater) enters through the heat rejection section. The heat recovery and heat rejection sections each consist of a series of flashing chambers. These flashing chambers, or stages, are connected to one
another horizontally. The total number of the flashing chambers in the modern MSF system is between 19 and 28 [63, 102-108], with top brine temperature (inlet feed brine temperature at the first flashing chamber) from 90 to 120 °C depending on the scale control method [109-121]. (Scale [122] depositions are formed by precipitation and crystal growth at the surface in contact with seawater inside the MSF).

In general, the number of flashing chambers lies between two to four times the gain ratio [123] (gain ratio or performance ratio can be defined as the mass ratio of distilled water to heated steam). The number of flashing chambers in the MSF system is determined based on:

1. The pressure difference between the flashing chambers, which is considered as the driving force of the brine. This controls the maximum number of the flashing chambers in the plant.
2. The flashing temperature range ($Flashing\ range = Inlet\ \text{brine} \ \text{temperature} (T_{in(i)}) - Saturation\ \text{seawater} \ \text{temperature} (T_{sat(seawater)(i)})$) inside the flashing chambers.
3. The cost of pumping the seawater.

The function of these flashing chambers, or stages, is to reduce the pressure inside each relative to the vapour pressure. This causes a rapid flashing of the heated feed brine that enters the individual stage and vapour is then produced. The vapour produced within each stage rises due to buoyancy and is converted to distilled water by condensation on condenser tubes. The distillate product is collected by a distillate tray within each stage. Finally, the total accumulated distilled product is withdrawn from the last stage tray.

In the brine heater, the temperature of the feed seawater is increased by the heating steam (Figure 1.15). The hot feed brine flows into the flashing chambers and it starts to vapourise. The feed seawater or the brine recycle
flows inside the condenser tubes to recover the latent heat of the produced vapour. The heated steam causes a surplus of heat, which is added to the system. This surplus heat is normally rejected to the environment in the heat rejection section, by using cooling seawater stream.

The brine flows from one flashing chamber to another in the MSF system due to the pressure difference between stages. The operating pressure inside the flashing chambers decreases from the first to the last stage respectively; similarly the brine temperature decreases from the first to the last stage.

Generally, there are several variables that affect the overall performance of the MSF system. The operating variables for a MSF desalination system are:

1. Seawater flowrate
2. Inlet/outlet seawater temperatures (in summer and in winter)
3. Top brine temperature
4. Operating pressure
5. Steam flowrate, steam temperature in brine heater
6. Recirculating brine flowrate
7. Make-up flowrate
8. Orifice height(s) at individual stage(s) inlet
9. Last stage brine level.

These variables can be divided into two categories:

1. External variables (e.g. heating steam, water quality, inlet seawater condition).
2. Controlled variables (e.g. top brine temperature and brine flow).

Thermodynamically, Darwish et al. (1993) [124] provided a comprehensive analysis of a recirculation MSF system based on the second law of thermodynamics. The study provided an exergy (exergy can be defined as
the highest work potential of energy) analysis of the different parts of MSF system. The exergy destruction analysis provides an optimisation of the MSF seawater desalination system and improvements to the system performance. Darwish et al. (1993) [124] showed that major exergy losses take place in the following parts of the MSF system: 1. the condensation process, 2. the flashing process, and 3. the feed brine heating process in the brine heater.

It should be noticed that both the condensation and flashing processes take place inside the flashing chamber. Furthermore, the flashing chamber is involved in the brine heater section also. Thus, the flashing chamber is the major part that dominates the exergy losses in MSF. Figure 2.1 shows the exergy percentages for different components of a MSF.

![Figure 2.1](image)

*Figure 2.1. Exergy percentage for different components of a MSF unit (source data: [125])*

Based on the thermodynamic analysis, the flashing chambers of MSF have the highest exergy. Thus, any improvement of the process within the flashing chamber, boiling process or condensation process will increase the system’s performance effectively. This considers the flashing chamber as the most important component in improving the overall performance of MSF and as a tool to reduce the capital cost of MSF.
2.3. Flashing Chambers

In a large scale desalination plant, a flashing stage typically [63] has dimensions of approximately 18 m width 4 m height and 3 m length relates to the horizontal direction of flow. Figure 2.2 shows a schematic diagram of both a MSF flashing chamber and a general case of the evaporation zone, which includes a horizontal flow channel with an open (free) surface and a flow baffle. The stage consists of the following basic elements: inlet orifice(s), splash baffle, flow baffle, brine pool, outlet orifice(s), condenser/preheater tubes, distillate tray, demister, venting line, and partition walls.

Each element in the stage has a specific function. An adjustable brine orifice controls the brine flow rate through the stages. A demister removes the brine droplets from the flashed-off vapour. A tube bundle of condenser, or preheat tubes, is where the water-vapour condenses on the outer surface. Latent heat released from the condensation process produces heat for the brine recycle steam which flows inside the condenser tubes. A distillate tray collects the distilled water. A venting system (not shown) is provided to allow removing any non-condensable gases since a small amount of non-condensable gases deteriorates the heat transfer rate of the condensation process from the vapour side. Instruments such as thermocouples, level sensors, and conductivity meters are used in monitoring and controlling the system.

2.3.1 Flashing chamber construction materials

Materials choices are only one of the significant parameters for designing the flashing chamber and MSF system. These materials have the ability to resist corrosion reactions, and withstand to harsh conditions of high salinity and high pressure.
Figure 2.2. Schematic of a typical flashing chamber shape in a MSF desalination plant
The major factor for stage material selection is the operating temperature of the stage (brine temperature). For this reason MSF processes operates at a range of temperatures between 30-110 °C [110] and all flashing chambers are at a temperature below 100 °C. These limitations of brine temperature are important to avoid accelerating corrosion [110-112] of metal surfaces in contact with the brine.

Appendix B shows photographs of an example of a typical case of a highly corroded flashing chamber [126] due to high brine temperature in Al Khobar MSF plant (Jeddah, Saudi Arabia – Phase II).

The formation of scale [127, 128] inside the heat exchanger tubes (brine heater or condenser tubes) inside the flashing chamber is considered as one of the major problems in operating MSF distillation plants. Scale deposits in the brine heater and heat recovery section lead to higher energy consumption or reduction of desalinated water production.

Stainless steels and its alloys are the most common materials used for flashing stage construction, because of their high resistance to corrosion and high heat losses reduction. Table 2.1 shows the limitations of operating variables of tube materials of the preheater/condenser tubes inside the flash chambers. A comprehensive study of the material selection and considerations can be found in Darwish et al. (1995) [123].
Table 2.1. Properties of the characteristics of tube materials of preheater/condenser tubes [63]

<table>
<thead>
<tr>
<th>Material</th>
<th>Brine Temperature [°C]</th>
<th>Thermal Conductivity [kW/m °C * 10^3]</th>
<th>Average Brine Speed [m/s]</th>
<th>Wall Thickness [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cu/Ni 70/30 (66% Cu, 30% Ni, 2% Fe, and 2% Mn)</td>
<td>Above 80</td>
<td>29</td>
<td>2-4</td>
<td>1.2</td>
</tr>
<tr>
<td>Aluminium Brass (76% Cu, 22% Zn, and 12% Al)</td>
<td>Below 80</td>
<td>32</td>
<td>1.5-2.5</td>
<td>1.2</td>
</tr>
<tr>
<td>Titanium</td>
<td>Below 80</td>
<td>16.5</td>
<td>3-20</td>
<td>0.5</td>
</tr>
<tr>
<td>Cu/Ni 90/10</td>
<td>Below 80</td>
<td>44</td>
<td>2-4</td>
<td>1.2</td>
</tr>
<tr>
<td>High Steel Alloy</td>
<td>Below 80</td>
<td>19.9</td>
<td>3-10</td>
<td>0.7</td>
</tr>
</tbody>
</table>

Cu = Copper, Ni = Nickel, Fe = Iron, Zn = Zinc, Al = Aluminium

Materials contribute to the capital cost of the MSF system. While using better materials or improvements in materials might increase the capital cost, it can increase the life of the plant, thus decreasing its maintenance cost which can be estimated as 6% [129] of the total cost of MSF plant. This can be a long term benefit of the plant.

In addition, analyses of the flashing chambers from a mathematical modelling, heat and mass transfer, and thermodynamics aspects are necessary to design optimal operating characteristics of the flashing chamber and of the MSF system. These calculations are used to determine the effects of operating variables and of flashing chamber characteristics [130].

Mathematical models with thermodynamic analysis are important for further understanding of flashing chamber design parameters (e.g. performance ratio, specific flow rate of recirculated brine, heat transfer coefficient) in MSF systems.
Since the heat transfer area in MSF plant represents about 30% [129] of the total cost of the plant, these parameters may be the main contributor to the cost of producing water.

2.3.2 Mathematical representation of MSF system & flashing chamber components

Many studies have developed a model by solving the governing equations of mass, energy and momentum (in steady state and in transient conditions). These models were developed to evaluate the performance of MSF desalination systems, to investigate the main design factors and to describe the dynamic behaviour of brine inside the flashing chamber [59,109, 131-169]. Table 2.2 reviews some of the mathematical models with their numerical solution methods of MSF systems.

Table 2.2. Highlights of some MSF system mathematical models with their numerical solution methods

<table>
<thead>
<tr>
<th>Author(s)</th>
<th>Achievements</th>
<th>Ref(s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Omar (1983)</td>
<td>Developed a steady state analytical mathematical model for solving the governing equations (mass and energy) for each flashing chamber. Successive approximation used in order to solve the equations. Calculations were based on stage to stage method. The model was transferred into Fortran IV computer code for either designing or simulating MSF plants.</td>
<td>[135]</td>
</tr>
<tr>
<td>Helal et al. (1986)</td>
<td>Developed a tridiagonal matrix model for solving steady state nonlinear equations which described the MSF performance. The equations solved were based on the Thomas algorithm. Calculations running time showed a significant reduction compared with calculations based on stage to stage method.</td>
<td>[136]</td>
</tr>
<tr>
<td>Al-Mutaz et al. (1989)</td>
<td>Developed a steady state analytical mathematical model for MSF plant. The numerical equations were solved by an orthogonal collocation method. The model was applied for selected number of stages in MSF. The model provided is used as a tool to optimize and predict performance of any MSF plant.</td>
<td>[137]</td>
</tr>
<tr>
<td>Author(s)</td>
<td>Description</td>
<td>References</td>
</tr>
<tr>
<td>---------------------------</td>
<td>-----------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------</td>
<td>------------</td>
</tr>
<tr>
<td>Aly et al. (1995)</td>
<td>Developed a computational model to predict MSF plants performance. The model solved the governing equation of mass, energy and momentum under transient conditions. The model was able to describe the dynamic behaviour of brine inside each flashing chamber.</td>
<td>[109]</td>
</tr>
<tr>
<td>El-Dessouky et al. (1996), (2000) &amp;. (2002)</td>
<td>Developed a steady state analytical model by solving governing equation of mass and energy for all stages in a MSF plant. Newton’s method was used to solve the equations simultaneously. The model provided significant information about designing MSF plants and flashing chambers. Furthermore, it provided an evaluation description of MSF operating performance.</td>
<td>[63, 143, 151]</td>
</tr>
<tr>
<td>Al-Shayji (1998)</td>
<td>Developed a neural network of large scale MSF desalination systems to predict the MSF plant performance. Developed steady-state and dynamic simulations of a large-scale commercial MSF plant. Optimised the performance of a large-scale MSF plant. This model was the first comprehensive study to predict and simulate MSF plants using Aspen Tech advanced simulation software tools.</td>
<td>[147]</td>
</tr>
<tr>
<td>Gambier et al. (2004)</td>
<td>Developed a hybrid modelling of MSF plant to improve the automated functions of the plant.</td>
<td>[156]</td>
</tr>
<tr>
<td>Abdel-Jabbar et al. (2007)</td>
<td>Used Newton’s method for solving the mathematical equation for each flashing chamber in MSF plants. The model added several design parameters to consider the characteristics of the large MSF capacity. The model focused on evaluation of weir loading, dimensions of the condenser tube bundle, demister dimensions, stage dimensions, brine temperature, and flow rate profiles.</td>
<td>[163]</td>
</tr>
<tr>
<td>Hala (2011)</td>
<td>Developed a detailed dynamic modelling of MSF desalination system (MSF-OT) and (MSF-BC) in order to contribute better understanding of MSF processes. The models provided can be used as tools to obtain accurate system design, evaluate system performance, and explain complex phenomena within the flashing stage. The dynamic model was coded and solved by using gPROMS software.</td>
<td>[59]</td>
</tr>
<tr>
<td>Hawaidi (2011)</td>
<td>Developed a steady state model of MSF based on the basic conservation equations: mass, energy and momentum. gPROMS software was used to develop and validate the model. The model was used to design and to optimise the operating conditions of MSF. The model was used to simulate, optimise and flexible scheduling of MSF desalination process under brine heater and demister fouling as well.</td>
<td>[169]</td>
</tr>
</tbody>
</table>
2.3.2.1. Flashing chamber mathematical model analysis

Mathematical analysis [63,101, 109, 123, 124, 170, 171, 172] can be presented based on the main fundamental dynamic equations. Conservation of mass, conservation of energy, and momentum equations are used to describe the system. Applying these equations to any system requires defining a control volume.

In a flashing stage, the system can be divided to four control volumes (as shown in Figure 2.3.); brine space, vapour space, distillate tray and condenser/preheat tubes. In each control volume the mass, salt, and energy balance and heat transfer equations are applied.

The following form the governing equation [170]. They are presented in their general forms:

Conservation of Mass

$$\frac{d}{dt} m_{\text{system}} = \frac{d}{dt} \int_{CV} \rho dV + \int_{CS} \rho (\mathbf{V}_r \cdot \mathbf{n}) dA = 0 \quad (2.1)$$

Linear Momentum

$$\frac{d}{dt} (m\mathbf{V})_{\text{system}} = \sum \mathbf{F} = \frac{d}{dt} \int_{CV} \mathbf{F} \rho dV + \int_{CS} \mathbf{F} \rho (\mathbf{V}_r \cdot \mathbf{n}) dA \quad (2.2)$$

Conservation of Energy

$$\dot{Q} - \dot{W}_s - \dot{W}_v = \frac{\partial}{\partial t} [ \int_{CV} (u + \frac{1}{2} V^2 + gZ) \rho dV] + \int_{CS} (h + \frac{1}{2} V^2 + gZ) \rho (\mathbf{V} \cdot \mathbf{n}) dA \quad (2.3)$$
Figure 2.3. Block diagram of a flashing chamber (i).
In the section below, a mathematical analysis of the brine space control volume is presented. The following assumptions are applied to simplify the analysis model [63, 101, 109, 123, 124, 171, 172, 173]:

- There is no salt in the vapour ($c_v = 0$).
- Specific heat at constant pressure $c_p$ is constant and equal 4.18 kJ/kg°C for all fluids in the stage.
- Heat losses to the surrounding are negligible.
- The vapour is saturated and sub-cooling of condensate is negligible.
- Pumps and auxiliaries powers are negligible.
- Liquid is perfectly mixed inside each stage.

Mass balance

$$\dot{m}_{b(i-1)} + \dot{m}_{d(i-1)} = \dot{m}_{b(i)} + \dot{m}_{d(i)} \quad (2.4)$$

Salt balance

$$\dot{m}_{b(i-1)} C_{b(i-1)} = \dot{m}_{b(i)} C_{b(i)} \quad (2.5)$$

Energy balance

$$\dot{m}_{b(i-1)} h_{b(i-1)} = \dot{m}_{b(i)} h_{b(i)} + (\dot{m}_{b(i-1)} - \dot{m}_{b(i)}) h_{v(i)} \quad (2.6)$$

Heat transfer equation

$$\dot{m}_{r(i)} c_p (T_{f(i)} - T_{f(i+1)}) = U_i A_i LMTD_{(i)} \quad , \quad (2.7)$$

where

$$LMTD_{(i)} = \frac{T_{f(i)} - T_{f(i+1)}}{\ln \frac{T_{d(i)} - T_{f(i+1)}}{T_{d(i)} - T_{f(i)}}} \quad (2.8)$$
2.3.3. Flashing chamber design factors

Such a model can present many important design parameters e.g. performance ratio, specific heat transfer area, specific brine feed flow rate, specific cooling water flow rate, amount of distilled water produce, and thermal stage performance.

- The performance ratio (PR) or gain ratio \([63, 123, 173]\) is an indicator of industrial scale. If the value of the performance ratio is less than one the unit cannot be used on an industrial scale. It can be defined as the mass ratio of distilled water per unit mass of heated steam.

\[
PR_{(i)} = \frac{m_{d(i)}}{m_{s(i)}}
\]  

- The specific heat transfer area \([63, 123]\) can be defined as the ratio of total heat transfer area to distillate flow rate. Increasing the specific heat transfer area leads to improved heat transfer but increases the cost. Thus an increase in the performance ratio leads to a decrease in the specific heat transfer area (inversely proportional).

\[
Specific\ heat\ transfer\ area\ [m^2/(kg/sec)] = \frac{A_{total}}{m_{d(i)}},
\]

where,

\(A_{total}\) = heat transfer area for the brine heater and all condensers in the plant.

- The ratio between feed brine flow rate \([63, 123]\) and distillate produced is used as an indicator for the provision of chemical additives and treatments for the stage.

- Cooling water flow \([63]\) rate indicates the size and cost of the intake seawater pumping unit and its power consumption.
Specific cooling water flow rate \( = \frac{m_{\text{inlet seawater}}}{m_d} \) \hspace{1cm} (2.11)

These design parameters are important because they control the cost of distilled water production from all MSF plants.

The specific heat transfer area depends on the number of flashing chambers in the MSF system. For a certain limit, this can reduce the capital cost of the MSF system. This limit is reached when the capital cost of manufacturing additional new flashing chambers exceeds the saving from the specific heat transfer area.

There is a strong relationship between performance ratio and specific heat transfer area. For an improvement in performance ratio, (energy cost saving requirement) a greater specific heat transfer area is required, and this increases the capital cost of the system.

The specific heat transfer area can be reduced by improving the heat transfer rate inside each flashing chamber. This can be attained by increasing the temperature difference between the inlet feed brine and the saturation temperature, corresponding to the same flashing chamber. This can be defined as the flashing range of the flashing chamber. Practically, this can be reached by increasing the inlet feed brine temperature at the first flashing chamber (this temperature [174] is known as the top brine temperature \( T_{BT} \)). The effect of this \( T_{BT} \) on the MSF system can be shown through a thermodynamic analysis of the system.

\[
\text{Flashing range of any flashing chamber} [\degree C] = T_{(i)} - T_{\text{sat of the seawater}(i)}
\] \hspace{1cm} (2.12)

\[
\text{Flashing range of MSF system} [\degree C] = T_{BT} - T_{\text{sat of the seawater (at the last stage)}}
\] \hspace{1cm} (2.13)
2.3.4. Thermodynamic view of flashing chamber

The inlet seawater temperature is the same as the sea temperature. Thus, it changes with respect to the weather and season. As shown in the schematic, (see Figures 2.4, 2.5) inlet seawater enters the system through preheated tubes. This increases in the inlet seawater temperature, thus it reduces the required energy at the heat input section. While the seawater enters the heat input section in the MSF system, its temperature rises until it reaches the top brine temperature level. Finally, the brine enters the flashing chambers.

Thermodynamically, the preheater tubes recover the energy added to the heating steam in the heat input section, and the preheater is further used to control the flashing chamber vapour pressure.

The complexity of the thermodynamic analyses arises due to the variation of vapour pressure inside the flashing chamber and thus the saturation temperature. For a better understanding, the flashing chamber may be divided vertically into three layers: layer a, b and c. (see Figure 2.4.).
Figure 2.4. Temperature profiles of a brine heater with first flashing chamber
Figure 2.5. The flashing cycle inside a flashing chamber shown on the (T-S) diagram for pure water
Layer (a) is the brine. Layer (b) is located above layer (a), where the vapour is located below the flashing chamber demister. Layer (c) is the space above the flashing chamber demister. Each layer has a different vapour pressure.

The vapour pressure in layer (a) is for seawater fluid ($P_{\text{sat(seawater)}}$). This vapour pressure is based on the salt concentration; it also changes along the flashing chamber based on the change in salt concentration during the flashing process (a comprehensive explanation of the flashing process is presented in Chapter 3). In general, the value of vapour pressure of seawater is greater than that of pure water.

The space above layer (a) is divided into two layers to allow for the demister, since the demister causes a pressure drop across its thickness.

\[
\text{Pressure drop in demister} = P_{\text{sat(purewater)}} - P'_{\text{sat(purewater)}}, \quad (2.14)
\]

where

($P_{\text{sat(purewater)}}$) is the vapour pressure in layer (b) for pure water fluid and ($P'_{\text{sat(purewater)}}$) is the vapour pressure in layer (c) for pure water.

The brine temperature decreases from one flashing chamber to another. The difference between outlet and inlet brine temperatures of the flashing chamber is known as the flashing down [175] $\Delta T_{\text{flash down}} = T_{\text{in}} - T_{\text{out}}$. While the brine temperature decreases horizontally along the flashing chamber, brine concentration (salinity) and saturation temperature increases. This reduces the heat transfer rate, because the flashing range decreases, and this is considered a thermal loss. Thermal losses are critical factors with regard to the design of the flashing chamber. The increase of these losses leads to reduction in the design performance ratio. Therefore, thermal losses should be considered.
For designing MSF system two types of thermal losses should be considered: the terminal heater temperature ($TTD_h$), and the terminal condensation temperature ($TTD_c$).

$TTD_h$ is the temperature difference between the steam temperature inside the brine heater and the top brine temperature, while $TTD_c$ is the temperature difference between the condensation temperature inside the flashing chamber and the preheated seawater temperature (see Figure 2.4):

\[
TTD_h = T_{steam} - T_{BT}
\] (2.15)
\[
TTD_c = T_{condensation} - T_1
\] (2.16)

The total terminal temperature difference, $TTD = TTD_h + TTD_c$, is a critical design factor of a multi stage flash desalination system. The value of $TTD$ is between 3 and 5 °C [63].

For the flashing chamber in particular, total thermal losses (thermodynamic losses) $\Delta T_{loss}$ inside any flashing chamber consists of non-equilibrium losses ($NETD$), boiling point elevation ($BPE$), and the thermal loss associate with the demister pressure drop ($P-P$). Of these, the non-equilibrium losses are the largest and the most uncertain [176].

The total thermodynamic loss ($\Delta T_{loss}$) is the temperature difference between the outlet average temperature of the brine and the condensation temperature:

\[
\Delta T_{loss} = NETD + BPE + (T_{sat} - T'_{sat})
\] (2.17)

The value of this temperature difference affects the area of the brine heater. It is desirable to minimise the loss to reduce the area of the brine heater.
2.3.4.2. Non-equilibrium losses inside the flashing chamber

Thermodynamics science [170] builds upon the concept of equilibrium states. So, in general, any thermodynamic system seeks to reach an equilibrium state at the phase interface.

Ideally, for the case of the flashing chamber as the vapourisation commences, the brine proportionately cools down, and vapourisation thus proceeds in the flash chamber until thermodynamic equilibrium between the liquid and the vapour is attained.

In reality, the brine cools down and loses temperature along the flashing chamber length, to a temperature slightly above the saturation temperature of pure water (Figure 2.4.). Due to incomplete flashing of the feed brine [177], the system cannot reach this equilibrium state. The divergence of reaching the equilibrium state at the end of the flashing chamber length is referred as non-equilibrium losses. Thus, non-equilibrium losses are an indicator of an incomplete flashing process inside the flashing chamber. These non-equilibrium losses represent residual superheating, relative to the corresponding to the saturation temperature of the seawater, of the brine at the exit of the flashing chamber.

Non-equilibrium losses are a function of the following variables:

1. Number of flashing chambers in the system.
2. Flashing chamber dimensions (stage length, width, height).
3. Flow parameters.
4. Saturation temperature of seawater.
5. Stage flash down.
6. Brine mass flow rate per unit chamber width.
7. Brine depth.
8. Boiling elevation.

10. Number of active nucleation sites.

Non-equilibrium losses can be defined as the temperature difference between the average temperature of exit brine and saturation temperature of seawater inside the flashing chamber \((T_{out} - T_{sat(seawater)})\), where \(T_{out}\) is the outlet mean temperature of the brine inside the flash chamber and, \(T_{sat(seawater)}\) is the saturation temperature of the seawater.

Several synonyms appear in the literature that refers to the same mathematical term of non-equilibrium losses viz. non-equilibrium allowance \((NEA)\) \([63,171,173]\); non-equilibrium temperature loss difference \((NETD)\) \([178]\); or non-equilibrium temperature loss \((\Delta')\) \([83]\); non-equilibrium allowance \((\Delta')\) \([179,180]\). In the present work, to ensure consistency ‘non-equilibrium temperature loss difference \((NETD)\)' will be the only term used to represent non-equilibrium losses, and is defined as:

\[
NETD \, [^\circ C] = T_{out} - T_{sat(seawater)}
\]  

(2.18)

The \(NETD\) is a good indicator of the approach to equilibrium in a flash chamber \([84]\). As the value of the \(NETD\) tends to zero, so the system inside the flashing chamber approaches the thermal equilibrium condition. Once the system reaches the thermal equilibrium conditions, no more vapour is released.

In high-performance MSF evaporators, the value of flash-down is around 2°C \([84]\) and the value of the \(NETD\) varies from 0.03 to 0.8°C \([77]\) In industrial practice, for economic reasons, it is logical to try to minimise flash-down as much as possible.

Empirical correlations were reviewed by Lior (1986) \([85]\), and Rautenbach \(et\)
al. (1996) [87]. Recently, Lior et al. (2013) [88] provided an empirical non-equilibrium correlation based on small scale experiments and as a function of:

1. The average brine temperature, $T_{out}$ [°C]
2. The saturation temperature of purewater, $T_{sat(purewater)}$ [°C]
3. The flashdown in the flashing chamber, $\Delta T_{ld}$ [°C]
4. The flashing chamber length, $l$ [m]
5. The flashing chamber load, $\omega$ [kg/(h.m)$^{-1}$]
6. The brine depth, BD [m]

$$T_{out} - T_{sat(purewater)} \ [°C] = BPE + NETD = BPE + \alpha\left(\frac{\Phi}{\alpha}\right)^{0.3281}$$  \hspace{1cm} (2.19)

Here $\alpha$ and $\Phi$ are empirical quantities given by:

$$\Phi = 0.9784^T_{sat(purewater)} \times 15.7378^{BD} \times 1.3777^{\omega \times 10^{-6}}$$  \hspace{1cm} (2.20)

$$\alpha = 0.5 \Delta T_{flashdown} + \Phi$$  \hspace{1cm} (2.21)

The overall temperature difference between flashing brine and saturation temperature of pure water comprises the sum of the boiling-point elevation ($BPE$) and non-equilibrium losses ($NETD$) (see Figure 2.4.).

2.3.4.3. **Boiling point elevation and brine concentration**

The boiling point elevation ($BPE$) is considered as the second type of irreversible thermal loss. It is an important factor in the design of flashing chambers. The brine boiling temperature is always higher than the saturation temperature of pure water, corresponding to a similar pressure inside the flashing chamber. The value of the boiling point of seawater (35w/w % salinity) at atmospheric pressure is 106 °C. More data and information about the seawater boiling point with its concentration and the seawater pressure can be found in [182].

53
The difference between the two temperatures is considered as the boiling point elevation \((BPE)\). The difference in temperatures may seem small but, in plants with a high performance ratio, a single degree in temperature is considered a significant loss [1].

\[
BPE = T_{\text{sat(seawater)}} - T_{\text{sat(purewater)}}
\]  

(2.22)

In each flashing chamber [173] there is a limited difference between the flashing brine and the condensation vapour temperatures. This reduces the mean temperature difference. This reduction affects the heat transfer inside the flashing chamber, therefore a larger heat transfer area is necessary in order to reach the required/designed heat transfer rate.

During the flashing process and phase change, the brine concentration increases along the flashing chamber length. This leads to an increase in the boiling point temperature, and hence of a BPE decrease in the rate of flashing along the flashing chamber length.

The relationship between these two variables can be seen in the boiling point elevation \((BPE)\). The following expression for estimating the value of the \(BPE\) as a function of temperature concentration is presented by Fabuss et al (1966) [182]:

\[
BPE = Ac + Bc^2
\]  

(2.23)

where, \(A\) and \(B\) are temperature dependent constants:

\[
A = A_1 + A_2T + A_3T^2, \quad B = B_1 + B_2T + B_3T^2 \quad c \text{ is the concentration expressed by the chlorinity of water factor, while } T \text{ is the average brine temperature expressed in degrees Celsius. This correlation is valid in the range of 20 to 180 °C.} 
\]
The brine concentration or the BPE are not the only liquid properties which change along the stage length, so also do the other thermal properties *i.e.* density, thermal conductivity, specific heat, viscosity, Prandtl number, surface tension. However, due to the small variations in temperature, most of the changes can be neglected in designing a flashing chamber. Typical values of MSF design parameters and performance evaluation are presented in *Table 2.3.*

**Table 2.3.** Typical values of MSF design parameters and performance evaluation

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Typical Values</th>
<th>Ref(s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of stages in MSF system</td>
<td>Twice to four times the gain ratio</td>
<td>[123]</td>
</tr>
<tr>
<td>Performance ratio (new plant)</td>
<td>8</td>
<td>[63,110]</td>
</tr>
<tr>
<td>specific heat transfer area</td>
<td>260 m²/(kg/sec)</td>
<td>[63]</td>
</tr>
<tr>
<td>Specific cooling water flow rate</td>
<td>2.5</td>
<td>[63]</td>
</tr>
<tr>
<td>Stage shape (width, height, length)</td>
<td>18 m x 4 m x 3 m</td>
<td>[63]</td>
</tr>
<tr>
<td>Minimum brine level inside any stage</td>
<td>0.2 m + orifice height</td>
<td>[63]</td>
</tr>
<tr>
<td>Maximum TBT</td>
<td>90-120°C</td>
<td>[63,110, 174]</td>
</tr>
<tr>
<td>Flashing down</td>
<td>2-5°C</td>
<td>[110]</td>
</tr>
<tr>
<td></td>
<td>1.5°C</td>
<td>[178]</td>
</tr>
<tr>
<td></td>
<td>0.6 – 4°C</td>
<td>[175]</td>
</tr>
<tr>
<td>Thermal losses (large scale MSF, 20-29 stages)</td>
<td>0.5 – 2°C</td>
<td>[63]</td>
</tr>
<tr>
<td><em>NETD</em></td>
<td>0.03 -0.8°C</td>
<td>[171]</td>
</tr>
<tr>
<td></td>
<td>1.5 - 2°C</td>
<td>[178]</td>
</tr>
<tr>
<td><em>NETD</em> for stage without baffle</td>
<td>4 – 5°C</td>
<td>[176]</td>
</tr>
<tr>
<td>TTD</td>
<td>3-5°C</td>
<td>[63]</td>
</tr>
<tr>
<td>Average velocity magnitude inside pipes</td>
<td>2 m/s</td>
<td>[63, 123]</td>
</tr>
<tr>
<td>Average velocity magnitude cooling seawater</td>
<td>1.5 m/s</td>
<td>[63, 123]</td>
</tr>
<tr>
<td>Maximum vapour release velocity magnitude per stage</td>
<td>4 m/s</td>
<td>[123]</td>
</tr>
<tr>
<td>Average vapour release velocity magnitude per stage</td>
<td>1.7 - 7 m/s</td>
<td>[1]</td>
</tr>
</tbody>
</table>

Most of the available values of the MSF operation variables are for special conditions. In MSF installations there are always some limiting values
necessary to avoid some unwanted phenomena. Thus, process control systems are required for running the MSF plant.

2.3.5. Desalination unit process control system

A process control system at the desalination unit is important for the proper functioning and system performance [159]. Running MSF system close enough to the optimal performance requires a sufficient control system for controlling brine level, flow, and temperature inside the flashing chambers. Many problems can appear because of poor control systems [63], such as thermal shock and accelerated corrosion.

Thermal shock appears when the feed seawater temperature heated to a temperature is not equal to last stage brine temperature. As a result of thermal shock, the corrosion reaction rate increases and the load on the vacuum rejecter increases also. To avoid such occurrences, a cooling control system is required. The main function of this cooling control system is to remove the excess heat added to the system.

Using corrosion control chemicals enhances the plant operation and increases its efficiency.

2.4. Summary

Flash ing chambers have the highest exergy destruction in the MSF system [125]. Thus, mathematical models with thermodynamic analysis were important for further understanding of flashing chamber design parameters (e.g. performance ratio, specific flow rate of recirculated brine, heat transfer coefficient) in MSF systems.

The evaporation zone may have the highest exergy in each flashing chamber [183]. Thus, the evaporation zone is considered [1] as the most important part
of the stage. A comprehensive study of the evaporation zone is presented in the following Chapter 3.
3. THE FLASHING PROCESS
& REVIEW OF PREVIOUS WORK

3.1. Introduction

Thermodynamically as noted in § 2.2 and Figure 2.1, the flashing chamber has the highest exergy in the MSF system, and the evaporation zone may have the highest exergy within the flashing chamber. In this respect the evaporation zone is considered the most important part of the stage.

Phase change inside a flashing chamber is a thermofluid phenomenon. It is the evaporation with self-boiling of a liquid due to a reduction in pressure, and in general it is a complex, multiphase process. In MSF desalination systems in particular, flashing flow is the key to desalination of feed brine in every stage.

This Chapter explains the importance of the evaporation zone and of its mechanism, the thermo-physics of the flashing process in horizontal open channel flow chamber (without a baffle plate). A literature review of previous work on the flashing process follows, wherein the following topics are treated: understanding the phenomena, mathematical model and correlations, experimental work, improving the flashing process, and computational studies.
3.2. Evaporation Zone

The evaporation zone, where brine flows (as shown in Figure 3.1), is located at the bottom of the flashing chamber, and it extends up to the free surface of the brine. In this zone, vapour is produced from the brine. Flashing is a thermo-fluid process. It is the surface evaporation with self-boiling of a liquid due to a reduction in pressure, involving a phase change and many interrelated variables. Some of these factors are: flow dynamics, phase interactions (heat generation and transfer between phases), mass transfer, and thermodynamics. All of them interact upon each other, therefore they effect and affect together and each other.

The evaporation zone [1, 100] is considered as the most important part of the stage, due to its key role in determining the design parameters of the flashing chamber. The overall dimensions and shape of the chambers are a function of the evaporation zone performance, as a result of the critical function of this zone. Furthermore, the evaporation zone has an economic importance. It can be an indicator of the capital and energy operating cost of the MSF system [184, 185] since the vapour load (amount of vapour coming out of a stage in a desalination process) reflects the energy cost of a thermal desalination system. This can be used for decision-making in the initial stages of the design process. Based on thermodynamic and economic aspects, the evaporation zone is worthy of further investigation, for researchers and industrial fields alike.

Many studies [7, 12, 26, 62, 63, 123, 157, 172, 175, 176, 178-181, 186-215] have been carried out to understand the mechanisms behind the evaporation zone, flashing phenomena and to propose methods for improving evaporation zone efficiency and performance.
Figure 3.1  Schematic of a flashing chamber shape with evaporation zone elements
A comprehensive description of desalination systems has been provided by El-Dessouky et al. (2002) [63]. The effects of the evaporation zone on geometry and dimensions were presented, correlations were introduced for chamber dimensions in terms of both width and length. They state that the chamber length of the stage is a function of width, vapour flow rate, vapour density, and vapour allowable velocity, while the stage width is a function of brine mass flow rate and a brine velocity.

Thermodynamically, the flashing process was described by Miyatake (1994) [175] and Fath (1997) [172]. Both stated that the produced vapour is created based on ‘an explosion of the liquid phase’ due to a reduction of the liquid brine pressure, which leads to a flashing evaporation inside the flashing chamber. In order for this to happen, the reduction of the pressure should reach values below the vapour pressure, corresponding to the liquid temperature.

Before comprehensively discussing the mechanisms of the phase change (from liquid to vapour) inside the flashing chambers in MSF, some physical mechanisms require defining. In general, there are many ways to produce a change in phase from liquid to vapour: Evaporation, boiling, cavitation, and flashing are the main processes.

3.2.1. Phase change mechanisms

*Evaporation* is a surface phenomenon. When liquid molecules have enough kinetic energy, they can escape from the liquid-vapour interfacial area as a vapour. The driving force for the phase change [216] is function of concentration, temperature or pressure gradients.

*Boiling, cavitation* and *flashing* are volume phenomena. The difference between these phenomena is based on how the liquid is superheated. In the
case of boiling the vaporisation takes place due to transfer of heat from a solid surface to a liquid. The driving force for the phase change is a function of the superheat temperature; it should be higher than 0.5-5 °C [217-220]. Cavitation and flashing are a special type of boiling. This may be defined as ‘self-boiling’ since the heat used to change phase comes from the fluid body itself. Also, it is called as ‘adiabatic boiling’ because it does not need any external heat source for the phase change (producing the vapour bubbles). Flashing is regarded as an intentional process, whereas cavitation is usually an undesirable process. Figure 3.2 shows the classification of phase change based on boiling.

![Classification of phase change based on boiling](image)

**Figure 3.2** Classification of phase change based on boiling

As engineers, we wish to know on a macroscopic level how vapour bubbles are formed and the reasons behind this, also how they grow and move. But before answering these questions we should consider the fact that vapour bubble behaviour and the heat transfer is based on the primary fluid geometric configuration, flow direction (horizontal or vertical) and orientation to the system. In our case, we are going to answer those questions assuming that the phase change takes place inside a container system. In the following section, we study the vapour bubbles behaviour inside a container.
3.2.1.1. Nucleation of vapour bubbles inside a container

Inside a container configuration part filled with a liquid, produced vapour bubbles normally are raised, e.g. to the liquid-gas interface (free surface), after a process of nucleation and growth.

Nucleation means the formation of bubbles; it can be defined as crossing phase boundaries. It requires an unstable thermodynamic state to initialise. It can take place due to many reasons [221]; for instance thermal expansion, or phase change during boiling. Four different classifications of bubble nucleation are described [222]: homogenous, heterogeneous, pseudo-classical, and non-classical nucleation.

Homogeneous bubble formation takes place in the liquid bulk of a homogeneous solution. This requires a very high level of supersaturation. Heterogeneous bubble formation requires surfaces of other materials in contact with the liquid for nucleation (see Figure 3.3). The nucleation occurs due to a sudden reduction of pressure in the system. Vapour bubbles then form either on the free surface of the liquid inside the container, on a molecularly smooth surface, or on a particle in the bulk. Bubbles then grow and detach leaving behind a portion of its gas. Jones et al. (1999) [222] stated that the main reason for bubble production continuation is due to the existence of gas filled cavities.

More details about nucleation theory, nucleation classifications, motion of bubbles and bubble growth can be found in [223-243]. After showing the mechanisms of the phase change, we are going back to our main consideration which is the flashing process. In the following section, a comprehensive review of mechanisms of phase change (from liquid to vapour) inside the flashing chambers in MSF is presented.
3.2.2. Flashing types and applications

Flashing (self-boiling) is a physical phenomenon. It represents a process of phase change from liquid to vapour phase due to the reduction in liquid pressure. In other words, the vapour bubbles are produced when the vapour pressure of a liquid is reduced to a level below the vapour pressure corresponding to the temperature of the liquid. Thus, it is clear that both of the heterogeneous and homogeneous types are the bubble formation type in the flashing phenomena case.

The flashing process can be classified [244] based on the flashing system as following: isothermal flash, adiabatic flash, three-phase flash or complex mixture.

The flashing phenomenon has a fundamental role in many industrial applications [245, 246]; desalination, vacuum freezing, refrigeration, food processing, and loss-of-coolant accidents (LOCA) of pressurised water in
nuclear reactors. In the following section, we present more details on the flashing process inside a flashing chamber in MSF.

### 3.2.3. Flashing classifications inside the flashing chamber

There is some confusion in the literature regarding flashing behaviour inside the flashing chamber. For this reason, we classify flashing into two categories based on the primary fluid (liquid phase) state inside systems (*Figure 3.4*):

1. **Pool flashing** (fixed initial amount of the primary fluid)
2. **Flow flashing** (continuous mass flow rate of the primary fluid).

*Figure 3.4*  Flashing classifications inside the flashing chamber

In the pool flashing type, there is a fixed amount of primary fluid, liquid phase, inside an enclosed container, and the phase changes only on the liquid free surface (a gravitational liquid-gas interfacial surface). This type can be also called flash evaporation. Gopalakrishna *et al.* [202] studied the dynamic behaviour of the vapour bubbles produced based on this type.

The second type is the flow flashing type. In this type the primary fluid keeps flowing through a container, the phase changes on the liquid free surface (liquid-gas interfacial) and in the bubble formation zone (close to the inlet). Thus, the phase changes are due to a combination of two different
mechanisms: \textit{self-boiling} which is located close to the inlet orifice, this region is called as bubble nucleation region, (we will talk about it in details in § 3.3); and to \textit{free surface evaporation} (at the free surface of the liquid). Lior (1973) [186] studied the behaviour of the phase change based on free surface evaporation type.

In the flashing chambers, the flow flashing type can be the more realistic one for describing the phase change process in real MSF plants. There is some confusion in the literature regarding \textit{flow flashing} type behaviours inside the flashing chamber. For this reason, we classify the flow flashing type into three categories based on the value of the \textit{NETD} inside the flashing chambers: ideal flashing ($NETD = 0$), finite flashing (\textit{i.e.} completed), and infinite flashing (\textit{i.e.} incomplete). \textit{Figure 3.5} shows the classification of the flashing process inside a flashing chamber in a MSF system.

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{classification_flashing}
\caption{Classification of the flashing process inside a flashing chamber}
\end{figure}
3.3. Flow Flashing Inside a Flashing Chamber

Phase change in a flashing chamber is a thermo-fluid process: it is the surface evaporation with self-boiling of a liquid due to a reduction in pressure. A flashing process is a flow flashing type, and it can be classified into three types as shown in Figure 3.6:

1. Ideal flashing \((\text{NETD}=0)\)
2. Finite flashing \((i.e. \text{ completed})\)
3. Infinite flashing \((i.e. \text{ incomplete})\)

Ideally, any flashing chamber is designed to work under operating conditions which leads to a zero value of the \(\text{NETD}\), and in this case the ideal flashing process is reached inside the flashing chamber. The corresponding length of the flashing chamber is considered as the optimal length. Thus, the optimal length \((L_{\text{optimal}})\) of the flashing chamber is where the value of the \(\text{NETD}\) is zero.

The optimal length of the flashing chamber is attained when the flashing process is completed just at the end of the flashing chamber. Thus, the value of the \(\text{NETD}\) is equal to zero and thermal flashing efficiency reaches its maximum corresponding to the operating conditions. However, in reality for some flashing chambers the flashing process may not be fully completed (infinite flashing) or may have completed somewhere inside the flashing chamber (finite flashing), and thus the \(\text{NETD}\) has a non-zero value. Figure 3.6 shows the three different flashing types inside a horizontal flashing chamber.

In infinite flashing, the phase change continues occurring until the end of the horizontal flashing chamber, while, for finite flashing, the phase change stops at some location within the flashing chamber when the flashing driving force, \(i.e.\) the superheat temperature, reaches zero. Depending on the flashing chamber operating conditions (assuming the rest variables are fixed \(e.g.\)
pressure, flow rate, temperature), either of the two flashing processes may occur. Thus, the operating parameters’ setting can affect the flashing chamber performance differently, depending upon the flashing type.

Figure 3.6 Flashing process classifications

Flashing can be defined as \textit{infinite flashing} when the exit brine temperature is higher than the saturation temperature of the brine, (corresponding to the flashing chamber operating pressure). In this case, the value of the \textit{NETD} is positive.

Flashing can be defined as \textit{finite flashing}, when the exit brine temperature is lower than the saturation temperature of the brine (corresponding to the flashing chamber operating pressure). In this case, the value of the \textit{NETD} is negative.
As a result, both finite and infinite flashing types are considered as thermal losses to the system. For the infinite case, there is still paid energy created, which ought to be utilised inside the flashing chamber, for the finite case which more energy needs to be expended in order for the next flashing chamber to recover waste energy generated during a no–phase–change period in the previous flashing chamber. For ideal the performance of the flashing chamber, in the case of infinite flashing, the brine needs either more time, or a longer flashing chamber, or maybe a higher flow rate, to attain the \( \text{NETD} = 0 \). Conversely, for the finite flashing case, the brine requires either less time, or a shorter flashing chamber, or higher flow rate, to attain the \( \text{NETD} = 0 \) and for thermal flashing efficiency to reach its maximum corresponding to the particular operating conditions.

3.3.2. Thermal efficiency of the flashing process

The thermal efficiency can be defined via the flashing process type. Dealing with infinite flashing, the following definition can be used (see Figure 3.6).

For infinite flashing cases, the thermal flashing efficiency is the ratio between the actual amount of brine flashing off and the maximum possible amount. It can be simply defined as the ratio of the flashing down and the flashing range:

\[
\text{Thermal flashing efficiency} = \frac{\text{Flashing down}}{\text{Flashing range}} \tag{3.1}
\]

\[
\text{Thermal flashing efficiency} = \frac{T_{in} - T_{out}}{T_{in} - T_{sat(\text{seawater})}} \tag{3.2}
\]

To deal with finite flashing, the following definition can be used:
\[ \text{Thermal flashing efficiency} = \frac{T_{\text{in}} - T_{\text{sat}(\text{seawater})}}{T_{\text{in}} - T_{\text{out}}} \quad (3.3) \]

### 3.3.3 Thermo-Physics of flashing flow inside the flashing chamber

The MSF flashing chamber maybe treated as an open system, mainly because the boundaries of the system are permeable both to energy and to mass. In general, the flow field in the evaporation zone may be classified as multiphase, complex, turbulent and unsteady, featuring a submerged orifice flow, a wall jet, flow recirculation and an unconstrained, open surface. Interactions exist between the liquid and vapour phases involving mass, momentum and energy exchanges related to vapourisation and condensation mass transfer, mean flow gradients and turbulence, buoyancy and heat transfer.

In this section we illustrate the thermo-physics of flashing in a horizontal open channel flow chamber, with a visual description of the evaporation zone. For simplicity, we consider two-dimensional flow in a flashing chamber without any flow baffles. According to the phase type and to the fluid behaviour, the chamber space may be regarded as divided into a number of layers and regions. These are indicated in Figure 3.7 adapted from Lior’s experiment work [180]. Lior 1973 [186] designed and constructed a flashing chamber (7.8 cm width, 113 cm long) to investigate the heat, mass and momentum transfer of vapour generated in the evaporation zone of the flashing chamber. (More details about Lior’s experiment and results can be found in Appendix C).
Based on Lior’s observations [180], the evaporation zone may be divided, vertically into three layers, and horizontally, into two regions, according to phase type and fluid behaviour.

In the vertical direction, the chamber space may be considered as comprising three layers: I – the brine flow layer, near the base; II – the free surface or brine-vapour interface layer; and III - the vapour layer, at the top of the evaporation zone. In the horizontal sense, there are two main regions: A – the bubble nucleation region, near the inlet, which includes the submerged jet and recirculating flow; and B - the channel flow, further downstream, which is predominantly unidirectional.

- **Vertical divisions**

**I. Brine flow layer** is the lowest layer and it consists of the bubble nucleation region and the channel flow.

**II. Free surface layer** lies between the brine flow layer and the vapour layer. It is often bubbly, highly unsteady and uneven, distinct interfacial surface. Bubbles bursting in the free surface layer, in combination with breaking
surface waves cause liquid brine droplets to the ejected into the vapour layer above.

The free surface is an unstable layer and is considered as two-phase. Here, the changing in phase from liquid to vapour, and from vapour to liquid, is very high. The process of evaporation and mass transfer takes place at the free surface. The driving forces for mass transfer [216] are functions of concentration, temperature or pressure gradients. Molecules of pure water at the free surface have the chance of escape to the vapour space due to the difference in concentration of the phases. The molecules escape from the free surface as vapour phase, and continue to escape (phase changing) until the system reaches the equilibrium situation.

III. Vapour layer (or vapour space) is located above the brine free surface. Here the volume fraction of the vapour phase is higher than that of the liquid phase, therefore the major phase in this layer is the vapour phase. Vapour molecules in the vapour layer may either continue to rise as vapour, or condense and move back to the free surface. This is determined based on the energy that the vapour particles have and on their buoyancy force. If the vapour particles rise, they pass through the demister to remove any liquid brine droplets.

- **Horizontal divisions**

A. **Bubble nucleation region (including submerged hydraulic jump region):** In this region, which is considered as two-phase flow (brine and water vapour), and where the phase changes rapidly from liquid to vapour, the movements of the fluid are violent. The temperature of the fluid is higher than the saturation temperature of the flashing chamber. The partial pressure of the fluid is higher than the vapourising pressure and vapour bubbles
form, grow and rise quickly. High nucleation rates and mass transfer take place in this region.

**B. Channel flow region** is located beyond the bubble nucleation region. The brine temperature decreases gradually along this region. Thus, the evaporation rate decreases gradually and the bubble formation rate decreases similarly until the brine reaches thermal equilibrium, when no more phase change occurs. This region is considered as a single phase region.

To summarise, phase change and mass transfer take place mainly in two regions; the nucleation region (A), and the free surface region (II). Thus, the vapourisation inside the flashing chamber can be described by two main mechanisms: self-boiling (flashing) in nucleation region (A), and evaporation process over the free surface region (II).

### 3.3.4. Flashing flow behaviour inside the evaporation zone

The flow may be generally classified as an open channel flow driven by a submerged jet [181]. The submerged orifice functions as a sluice gate, controlling the flow rate and brine level. Such flows are characterised by Froude number and Reynolds number. Here the Froude number, based on brine depth and mean inlet speed, lies around the critical value, which may result in a wide variety of surface waves and unsteady flow patterns: under these circumstances Froude number effects may be expected to dominate. Moreover, the degree of submergence of the inlet, characterised by the ratio of brine level at inlet/inlet orifice height, say, can also strongly influence the flow field, as will the geometric aspect ratio of the chamber.

Within the brine flow layer, under a process of heterogeneous flashing vapour bubbles emerge on nuclei of dissolved gas or of solid particles entrained in the brine. Vapour formation within the brine depends on the
local pressure and temperature: brine pressure $p_l$ depends on the local depth below the free surface. The local difference between the vapour pressure in a bubble and the brine pressure, the superheat term ($p_v - p_l$), is the key factor in determining whether the bubble will nucleate and grow, or collapse and disappear.

Young-Laplace equation for a spherical bubble [181], describes the equilibrium condition of an isolated spherical bubble:

$$p_v - p_l = \frac{2\sigma}{r_c}$$  \hspace{1cm} (3.4)

Where, $p_v$ is the vapour pressure inside a spherical bubble, $p_l$ is the liquid pressure surrounding the bubble, $\sigma$ is vapour-liquid surface tension and $r_c$ is the critical radius of the bubble.

The superheat term ($p_v - p_l$) is required for initiation of nucleation of the bubbles and for their formation. When the value of the superheat term is positive that means that the bubble grows. When it is zero, the bubble is in equilibrium (static), and when negative the bubble collapses and disappears.

Improving the flashing process requires reducing the value of the superheat term which responsible of initiation the nucleation of the bubbles. This can be done by several methods [26,186]:

- Increasing the bubble radius. This can be achieved by using a solid impurity or injecting non-condensable gas bubbles.
- Increasing the vapour pressure $p_v$. This can be achieved by adding non-condensable gases ($p_v + p_{non-condensable \text{ gas}}$).
- Decreasing the liquid pressure $p_l$. This happens during flow when feed brine encounters reduced pressure as it enters the flashing chamber.
• Decreasing the surface tension. This can be done by adding chemical components e.g. surfactants.

For describing the flashing process performance in the evaporation zone, the bubble nucleation frequency and flashing rate should be determined.

Shin et al. (1993) [247] explained that the nucleation behaviour in the bubble formation region can be described based on the maximum bubble nucleation frequency and on the site density. The site density is determined based on the maximum bubble nucleation frequency. Both quantities – maximum bubble frequency and site density – are used to determine the nucleation rate in the region. The maximum bubble nucleation frequency, $f_{\text{max}}$ [sec$^{-1}$] is given by $\Delta T_{\text{sup}} = T_{\text{brine}} - T_{\text{sat}}$ [°C]:

$$f_{\text{max}} = 10^4 \times \Delta T_{\text{sup}}^3 \quad (3.5)$$

$$J_{\text{w,max}} = f(n)(f_{\text{max}}, n, \text{geometry}) \quad (3.6)$$

The maximum nucleation rate, $J_{\text{w,max}}$ is given by the bubble number density, $n$.

Fath (1997) [207] defined the flashing rate as the flashing heat flux since both terms depend on the same parameters. The superheat temperature of the brine is the basic parameter which has significant influence on the flashing rate.

According to Fath [207] the following relation can be used to determine the flashing heat flux, or the flashing rate:

$$Q'' = 0.055 \times \Delta T_{\text{sup}}^3 \quad ,$$

(3.7)

where

$Q''$ is the flashing heat flux [kW/m$^2$] in the bubble formation region.
3.3.5. Visualising the flashing process on a Pressure-Enthalpy diagram

As it is noted in § 2.2.3 there are many type of flashing. Here, for simplicity, we assume the flashing inside the flashing chamber is considered as an adiabatic flashing. Adiabatic flashing [244] is a pressure drop process at constant enthalpy [248]. The process cycle of the flashing chamber can be represented by the $P - h$ diagram. Figure 3.8 shows four different states of the brine: point 1 is the state of the brine just before entering the flashing chamber, point 2 is at the inlet orifice of the flashing chamber, point 3 is located at the vena contracta, and point 4 is located at the end of the flashing chamber which can also be considered as a new enter state (as new current point 1) for the next flashing chamber.

![Diagram of flashing chamber](image)

**Figure 3.8** Flow pattern through an orifice

*Figure 3.9* shows the corresponding $p - h$ diagram. Here, point 1 is considered as a single phase component, and it is in the subcooled region. As the brine enters the flashing chamber its pressure reduces and once the pressure falls below the liquid saturation line (point 2) vapour bubbles appear and grow. At point 3, the vapourisation of the liquid becomes rapid, and this is considered as the inlet condition of the brine to the flashing chamber.
Figure 3.9  (a) The flashing cycle inside a flashing chamber on the pressure-enthalpy diagram for pure water. (b) Schematic of the phase change during the flashing in a simple chamber (adapted from Lior [180])
As illustrated in *Figure 3.9a*, the flashing process takes place along a horizontal line in the diagram, between points 3 and 4, with the initial condition of the flashing process being represented by point 3 and the final condition by point 4. Three mechanisms take place along the flashing line: nucleation, bubble growth and then reversion into single phase. Usually, the exit state (point 4) of the brine at the flashing chamber is located at the saturated liquid line.

### 3.4. Review of Previous Work on Flashing Processes in MSF Systems

There have been many studies aimed at understanding the flashing mechanisms and the effects of brine flow rate, brine level inside the flashing chamber, inlet temperature of the brine, flashing range and chamber length on the flashing process.

The following section reviews studies on the flashing process. For organisation, the review topics are grouped into five sections:

1. Understanding the phenomena
2. Mathematical models and correlations
3. Experimental works
4. Improving the flashing process
5. Computational works.

In the following section, a general review on the understanding of flashing phenomena is presented.
3.4.1 Understanding flashing phenomena

The complexity of flashing phenomena in the flashing chamber leads to some contradictory results. For instance, AMF Cuno Engineering Company (1966) [190] found experimentally that the flashing range, defined as the difference between the top brine temperature and the stage saturation temperature, does not affect the flashing efficiency. However, as part of Weir Westgarth Limited’s (1967) study [191], it was reported that increasing the flashing range leads to an increase in the flashing efficiency.

Highlights from works aimed at understanding the flashing process phenomena inside the flashing chamber in MSF systems are presented in Table 3.1.

<table>
<thead>
<tr>
<th>Reference</th>
<th>Achievements</th>
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<tbody>
<tr>
<td>Weir Westgarth Limited (1967)</td>
<td>Studied flash chamber geometry (experimentally) in MSF system. The study was divided into two main parts; an investigation of the effect of the foam heights on the flashing process performance inside the flashing chamber, and a study of the pressure losses through the demister inside the flashing chamber. The study provided the following conclusions: most of the vapour production inside the flashing chamber is released from the brine free surface. A relationship between foam height and vapour release rate was provided; the vapour release rate increases with the foam height. The foam height increases when brine level decreases. The foam height depends on brine circulation. The pressure loss through the demister increases with the square of the vapour velocity.</td>
</tr>
<tr>
<td>Gilbert (1968)</td>
<td>As a part of Gilbert’s PhD research, the mechanism behind the phase change of the brine from liquid to vapour phase was investigated. He stated that the basic difference between a boiling and a flashing process is that in the flashing process the heat used to vaporise the body of fluid must come from within the body of fluid itself, and not from a solid surface as in boiling.</td>
</tr>
</tbody>
</table>
As a part of a project that took place in Millstone Point Test Station, in Connecticut by Cuno-Maxim Division of American Machine and Foundry Corporation for the Office of Saline Water, an investigation (experimental) of multi-stage flash phenomena in a three-stage test system was provided by Williamson & Gilbert (1970). As a part of the study the following design parameters were investigated; the non-equilibrium allowance for a range of test conditions, stage temperature, liquid level, flow rate and flash down, and interstate flow characteristics for flashing stage. Several methods were used to determine an accurate expression for non-equilibrium temperature loss difference (NETD).

Based on this study the following conclusions was provided; when stage temperature and flashdown is low, the value of the NETD is large (two degrees or more).

Some experimental source errors and difficulties regarding controlling and measurement of some variables stated in the study were mentioned, e.g. the difficulty of maintain the brine level inside the flashing chamber. At the same time, the study provided methods to overcome some of these difficulties, and further methods for better measurement which allowed a more accurate and detailed analysis of the flow and the NETD. The most accurate expression of the NETD is a function of the following variables: the stage length, the brine mass flow rate, the vapour volume fraction and the stage flashdown temperature.

Lior (1973) Studied heat transfer with flash evaporation in a stream with a free surface. The study was divided into two sections; theoretical and experimental. He stated that the phase change takes place from both the free surface and from the bubble nucleation zone and the most phase change takes place at the free surface of the brine inside the flashing chamber. As part of this study, the difference between boiling process and flashing process was provided. In the boiling process, bubbles nucleate due to heat transfer from the solid surface to liquid. The nucleation can be homogeneous or heterogenous. In a flashing process, the superheat which is required for the phase change comes from the liquid itself (no solid surface is needed, i.e boiling process).

Darwish et al. (1976) A comprehensive study (experimental) was completed by Darwish (1976) to determine the effects of flow rate, brine level inside the flashing chamber, inlet temperature of the brine, flashdown range and chamber length. The study observed that the flashing efficiency increases by increasing the inlet temperature of the brine; by increasing the flashdown range; by increasing the chamber length; by decreasing the brine level; by reducing the brine flow rate.

Khan (1986) As a part of Khan’s comprehensive book of desalination processes, the flashing process was investigated. The principles of flash distillation were provided. It stated that vapours may be produced either by adding heat (boiling) or reducing pressure (flash) to a liquid. In the flashing case, the phase changes until the system
A better understanding of the flashing process is still needed.

### 3.4.2 Mathematical models and correlations

This section summarises the mathematical models and correlations studies for investigation the flashing process inside the flashing chamber (See Table 3.2).

**Table 3.2.** Highlights from mathematical models and correlations studies for investigation the flashing process inside the flashing chamber

<table>
<thead>
<tr>
<th>Reference</th>
<th>Achievements</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gilbart (1968)</td>
<td>To complement an experimental investigation of steady state flow of flashing salt solution between two submerged orifices at low pressure, a control volume analysis of the evaporation zone was provided. Energy and continuity correlation equations were applied to the flashing chamber in order to calculate the vapour production</td>
</tr>
</tbody>
</table>

Miyatake et al. (1992)  
Attempted to reach a quantitative understanding of the interaction between fluid mechanics and flash evaporation in the flashing chamber by means of a computational model. The field model dealt with isothermal, two-dimensional, steady state and turbulent flow. An empirical correlation was developed in order to estimate the shape of the free surface. The study dealt with a typical flashing chamber provided by a baffle and without a baffle. The main conclusions of this study were that the orifice coefficient is independent of the liquid superheat, of brine temperature, and of mass flowrate and the baffle plate promoted the evaporation rate. 

Miyatake et al. (1993)  
 Studied the relationship between flashing flow pattern and thermal non-equilibrium in the flashing chamber during the evaporation process. They stated that the non-equilibrium temperature difference (NETD) is reduced by increasing brine temperature. They found that the NETD is further reduced when the flow rate of the brine is increased.

Perry RH, & Green DW. (1999)  
As a part of this handbook, the flashing process was investigated. A classification of the process was provided. The flashing system can be: isothermal flash, adiabatic flash, three-phase flash or a complex mixture.
rate and heat removal from brine in the evaporation zone. In this study, the evaporation zone was divided into two layers: top layer (vapour) and bottom layer (brine). This provided a method of investigating the overall stage performance and stage efficiency by obtaining the non-equilibrium allowance \(\text{NETD}\).

**Seraq El-Din et al. (1978)** Studied a single flashing chamber experimentally. The aim of the study was to investigate the effect of interaction between operating parameters and design parameters of the flashing chamber on the thermodynamic efficiency of the flashing process. As part of this study, the thermodynamic flashing process efficiency was defined as follows:

\[
\frac{T_{\text{in}}-T_{\text{out}}}{T_{\text{in}}-T_{\text{sat}}}
\]

**Lior (1986)** Reviewed twelve empirical correlations for predicting the non-equilibrium allowance in the flashing chamber. All the correlations were dimensional. He noted the difficulties in measuring several parameters accurately, such as temperature difference of the vapour, velocity distribution at the inlet and at the outlet of the evaporator. He stated the sources of measurement errors and he showed how small values of these measurement errors can have a significantly effect on the correlations. He recommended the provision of new methods to estimate these empirical correlations.

**Miyatake et al. (1993)** Studied the relationship between flow pattern and thermal non-equilibrium inside flashing chambers in MSF system. The data used in this study were based on experimental work [180]. The numerical analysis was of turbulent isothermal flow patterns. As a result of the study, an empirical correlation was developed for obtaining the rate of temperature change along the evaporation zone inside a flashing chamber. This study showed that the \(\text{NETD}\) decreases when brine temperature increases, or when the feed brine flow rate increases.

**Darwish et al. (1995)** Worked through the flashing process of the MSF process during the development of thermal desalination. The study presented the main design features, e.g. operating temperature, flashing brine flow and vapour released velocity, gain ratio or performance ratio. It covered design of components and choice of materials of the flashing chamber in MSF system. As part of this study algebraic equations of the thermo-fluid process inside the flashing chamber were provided.

**Rautenbach et al. (1996)** Provided two empirical correlations which allow a reliable prediction of the \(\text{NETD}\) inside an open channel flow in flash chambers in a MSF system. One of the equations was for a simple rectangular orifice, while the other was for a rectangular orifice/weir. These correlations were tested and validated by data, from [249-251], which were collected based on test plants with high accuracy sensors.

As part of the study a number of correlations from previous works [1,192,193,252,253] were tested and showed different results for a given orifice configuration. They showed that six of the nine
previous correlations from literature gave unphysical results.

El-Dessouky et al. (2002) As part of a comprehensive explanation of industrial seawater desalination systems, the fundamentals of the flashing process inside a flashing chamber in a MSF system was provided. A mathematical model of the flashing chamber with several design correlations was presented. The model built was based on physical principles which included mass, energy equations, with thermodynamics and heat transfer considerations. In this study, an analysis of the flashing chamber was provided with synthesis of a flashing process inside a single flashing chamber. A mathematical model and analysis of a single flashing chamber was described, with correlations for counting thermal losses inside the flashing chamber, and for examination flashing performance inside the flashing chamber.

El-Dessouky et al. (2004) Provided empirical design correlations. The aim was to develop design correlations for the MSF system. These correlations include discharge coefficient, non-equilibrium allowance, and overall heat transfer coefficient. These correlations were based on the test data of existing MSF plants. However, the correlations can be used for a specific system capacity, dimensions, and operation temperature.

3.4.3 Experimental works

Comprehensive experimental studies aimed to understand the flashing process inside the flashing chamber in MSF system were undertaken by Gilbert (1968), Lior (1973) & (1980), Jin et al. (1999), Jin & Low (2002), Shichang et al. (2002), and Cipollina et al. (2007). Highlights of these experimental works are now presented in Table 3.3.

Table 3.3. Highlights from some experimental work for investigation the flashing process inside the flashing chamber

<table>
<thead>
<tr>
<th>Reference</th>
<th>Achievements</th>
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<tr>
<td>Gilbert (1968)</td>
<td>As part of Gilbert’s dissertation an experimental work was run to investigate the flashing behaviour inside three a series of flashing chambers. The flashing chambers were built from a steel tube 6.7 m long and 1.2m in diameter). The experiments were run under steady state conditions with a slightly reduced pressure from one flash chamber to the next. The testing chamber, where the measurement was taken, was the middle one (second flashing chamber). Measurements of the brine temperature, pressure and velocity were recorded to define the location of vapour formation, the rate of vapour formation and the amount of the vapour production. Gilbert provided a way of studying the flashing performance. The flashing</td>
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</table>
performance was a function of brine flow rate, stage flashing down, brine level, and inlet brine temperature.

Thermo-fluid behaviour of the brine inside the evaporation zone inside the testing flashing chamber was investigated by Gilbert. Based on his observations: the static temperature and heat transfer vary in both the horizontal and vertical directions inside the evaporation zone; the majority of vapour production was at the free surface of the brine inside the flashing chamber; the flow of brine in the evaporation zone was not a simple flow of pure liquid, but it consisted of two flow motions (liquid and vapour) that affect each other.

The complexity of dealing with oscillatory multiphase flow led to some experimental difficulties in Gilbert’s work. Measuring the vapour phase temperature in the vapour layer was not simple due to either the liquid dripping due to condensation or liquid drops thrown up by the brine evaporation. His analyses were based on assuming that the vapour layer was only consisted of one phase (vapour). The measured values of velocity magnitudes, and temperatures were not in steady state conditions since these required to be measured in an average time. Thus, errors were created between measured data and calculated values due to an inability of fixing the steady state conditions due to the strong oscillatory flow.

Lior (1973) Lior's PhD dissertation was aimed at investigating the thermo-fluid process under flashing conditions of seawater inside a flashing chamber in a MSF system. Therefore, he employed a scaled down (7.8 cm wide, 113 cm long) flashing chamber with glass windows for visual observation. The experiments were performed over (0.1≤ Froude number ≤0.2) and flashing down (1 °C – 3 °C ), with temperature measurement accuracy ± 0.02 °C and better.

Lior gave a visual description of the brine flow during the flashing process inside the flashing chamber. Based on these observations he divided the evaporation zone into two regions: a submerged sluice gate flow (two-phase mixture) followed by an open channel flow (one phase flow with few bubbles). He stated that the phase change of liquid inside the evaporation zone took place at the free surface of the brine and at the submerged sluice gate region but the majority was from the free surface. The vapour layer consisted of a multiphase mixture.

As part of Lior’s experimental work, a parametric study related to heat transfer and equilibrium with both flashing down and brine mass flow rate were presented. He found that brine temperature increased when flashing down of the flashing chamber increased and heat transfer can be improved when the flashing range was increased or when the brine mass flow rate was increased.

Seraq El-Din et al. Provided an experimental work for a single flashing chamber for investigating the interactions between design parameters and thermal flashing efficiency. As results of this study, the following points were
recorded: flashing thermal efficiency increased when brine flow rate increased, as a result of increased turbulence and mixing inside the evaporation zone; liquid level decreased; inlet brine temperature increased.

Lior (1980) Lior built on his previous experimental work [186] and he described the flashing process inside the flashing chamber from different aspects; heat transfer, mass transfer, thermodynamics and momentum transport. Furthermore, he investigated the effects of both brine flowrate and temperature on the hydraulic jump inside the flashing chamber. Lior achieved that bubble nucleation during the flashing evaporation process happens due to a cavitation-like phenomenon, and increasing of flash down temperature leads to bringing the brine temperature closer to the vapour saturation temperature corresponding to the stage pressure.

Jin et al., (1999) Performed experimental observations on single and multiphase flow patterns inside a flashing chamber. They used particle image velocimetry (PIV) to investigate the relationship between the flow pattern and the evaporation process during the flashing process at the evaporation zone. The flow patterns were characterised by using the velocity distribution measurements inside the evaporation zone, i.e. the back flow region near the inlet orifice, the free surface shape and the overall domain. They used a CCD camera to capture bubbles formation, motion and growth inside the bubble nucleation region. They found that during the flashing process the recirculation region size increased directly proportional with the water level in the chamber; and the bubble nucleation did not depend only on the degree of the liquid superheat but also on nuclei distribution.

Jin & Low, (2002) Investigated the single-phase flow patterns in a model flash evaporation chamber both experimentally, by making PIV measurements, and numerically. The flashing chamber dimensions had width, height, and length dimensions of approximately 40 mm x 300 mm x 823.8 mm, respectively. It was designed to be able to run at a flow rate of 1 kg/sec and at a temperature of 60 °C and an absolute pressure of 200 mbar with less than 1% measurement errors. The observations showed that a large recirculation region can be increased when the water level increased and/or when the brine flow rate increased.

Shichang et al., (2002) Provided experimental studies on the effect of the orifice configurations on the behaviour of the multiphase flow under the flashing process inside a flashing chamber. The flashing chamber dimensions had width, height, and length dimensions of approximately 0.1 m x 1 m x 2.2 m, respectively. Two types of orifices were used: plain and U-bottom orifice. The effects of the two orifices type on the thermodynamic performance were provided; the U-bottom orifice was advantageous.
3.4.4 Improving the flashing process

In this section, we summarise works aimed to provide methods and tools for improving the flashing process performance inside the flashing chamber (see Table 3.4.)

Table 3.4. Highlights from method aimed at improving the flashing process performance inside the flashing chamber

<table>
<thead>
<tr>
<th>Reference</th>
<th>Achievements</th>
</tr>
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| Desalting Systems and Services, Inc. (1971) | Developed a flash enhancer device that can be attached to each flashing chamber in a MSF system. The aim of the device was to reduce the NETD inside the flashing chamber. Full details of the device configuration can be found in [176]. The device is designed to divide the superheated feed brine into two streams; upper and lower part. The upper stream would flow under the conventional orifice of the flashing chamber. The deeper would flow through a duct along the bottom of the flashing chamber and then be lifted through a thermosiphon device above the upper part. The mechanism of introducing the feed brine into the flashing chamber by the device provided the feed brine into the evaporation zone as spraying. This improved the mixing process and increased the liquid-vapour interface area which led to minimise the NETD. The

in the vapour bubbles formation and growth this led to two opposing effects; large bubbles formation led to increase in turbulence and mixing process inside the bubble nucleation region but at the same time it caused additional energy losses. However, the U-bottom orifice had lower non-equilibrium temperature difference (NETD) in comparison to the plain orifice, thus higher thermodynamic efficiency.

Cipollina et al., (2007) investigated the flashing phenomena inside flashing chambers experimentally. They built a lab-scale MSF system consisting of three flashing chambers. Each flashing chamber dimensions had width, height, and length dimensions of approximately 0.3 m x 0.4 m x 0.6 m, respectively. The range of the operating conditions of the experimental runs were; top brine temperature range (51 – 63 °C), pressure (9 – 34 kPa), brine level (up to 0.1 m), brine flow rate (8.3x10^{-5} – 1.3x10^{-4} m/s), and feed brine velocity (0.7-1.4 m/s).

They showed that the NETD decreased dramatically when both pressure drop and/or temperature difference between inlet brine and saturation temperature of the brine, corresponding to the stage pressure, were increased. The orifice discharge coefficient was found to have decreased slightly when the pressure drop was increased.
overall capital cost of the designed MSF system was reduced by the reduction of the NETD. The improvement of 2 °F (1.1 °C) of the NETD leads to a 7% increase in the flashing performance. This saved 140,100 ft² (13015.72 m²) of heat transfer surface area at $2 per square foot, and $280,000 of the overall capital cost.

Veenman (1976) & (1977) As a way of improving a vertical multi-stage flash desalination system, a fluidised bed heat exchanger was used. It consisted of large numbers of vertical parallel condenser tubes. Due to this fluidised bed, the heat transfer co-efficient improved. Many other advantages were provided e.g. small plant size, better flash-off, increasing the flashing range.

El-Dessouky et al. (1979) They innovated an inter-stage device to reduce the sensitivity of the pressure energy losses with increasing of the feed brine flow rate inside the flashing chamber in MSF systems. The configuration of the device can be found in [254]. The idea of this device was to improve the flashing process by separating the flashing chamber, horizontally, into a vapour formation zone and a surface evaporation zone. This helped to eliminate jet submergence and partial back flow of the brine in the evaporation zone, thus pressure energy losses were reduced. The device provided a high liquid mixing between the vapour bubbles and the brine and this improved the evaporation process.

They tested this device experimentally and investigated the effect of the operating parameters on the thermodynamic efficiency and pressure energy losses in a scaled down flashing chamber. They found that overall performance was improved by using the device due to improved mixing in the evaporation zone. They found that the flashing efficiency increased when the brine temperature or the brine flowrate was increased, but the increase in the brine level led to a reduction in the flashing efficiency.

Lior et al. (1983)&(1995) The main aim of this study was an experimental investigation of the effects of generating hydrogen bubbles, electrolytically, into the flashing evaporation. By using this method, the flashing evaporation rate increased and the NETD was reduced by up to 15%.

Miyatake (1994) Performed experimental studies to compare the following different flashing systems; pool liquid exposed a sudden pressure drop in a container, superheated flowing liquid in a conventional MSF evaporator with an open channel and finally, superheated liquid jet ejected from a simple tubular nozzle or a circular orifice into a low pressure vapour zone. The flashing evaporation rate was used as an indicator of the flashing performance for each system. He concluded that the most efficient method of enhancing the flashing process was to
inject a nucleated liquid into a low-pressure vapour zone using a nozzle.

**Fath (1997)**

As part of an investigation into the flashing evaporation rate inside the flashing chamber, the author recommended four methods for improving the flashing process inside the evaporation zone. The flashing process would improve when: increasing the brine superheat, increasing the flashing surface area, increasing the number of active nucleation sites and increasing the residence time for the bubbles to nucleate, grow and transfer.

**Fath’s Patent (1998)**

He built on his previous study [172] and used a combination of MED and MSF for improving the flashing process inside the evaporation zones. By applying both systems together, the waste heat of the vapour from the MED can be used as an external heat source for brine reheat. This increased the average brine temperature inside the evaporation zone inside each flashing chamber in MSF.

**Jin et al. (2002)**

Improving the evaporation rate inside the evaporation zone can be reached by creating more turbulent flow and more recirculation regions.

**Shichang et al. (2002)**

Used a U-bottom orifice between the flashing chambers in MSF system which reduced the NETD, thus flashing efficiency improved.

**Cipollina (2005)**

Introduced a way of improving the evaporation process by using various evaporation promoters. The evaporation promoters were metallic surfaces located at the entrance of the evaporation zone. They were intended for promoting bubbles nucleation, for increasing turbulence in the evaporation zone and for improving the mixing of vapour bubbles with feed brine, thus, the flashing efficiency improved and the NETD decreased.

### 3.4.5. Computational modelling

Experimental work is always valuable; however, the cost and difficulties of constructing the experimental components and running flashing phenomena can be high. Providing accurate measurement equipment is also needed for such work. Any change in the geometry is required, often, to build a new experimental set up; this can be time consuming with an extremely expensive cost.
Thus, computational models can be useful as a good model can help to better understand the flashing phenomena and to investigate the effect of different parameters on the process and system. This can lead to innovative methods for improving not only the flashing process, but also the MSF system. Thus computational models can be a very useful design tool in the MSF desalination field.

To date, there is a shortage of available results for computational prediction of flow inside the flashing chamber, specifically for the evaporation zone. These predictions can be divided into two groups: the vapour space (a single-phase modelling) and the evaporation zone (multi-phase modelling).

3.4.5.1. Computational modelling of the vapour space

In this section, we summarise some computational modelling aimed at prediction the vapour space inside the flashing chamber (Table 3.5).

Table 3.5. Highlights from some computational modelling aimed at predict the vapour space inside the flashing chamber

<table>
<thead>
<tr>
<th>Reference</th>
<th>Achievements</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rahimi et al., (2008)</td>
<td>Attempted to determine the pressure drop in the demister in a stage of a MSF desalination system using FLUENT 6. This simulation was built for two-phase flow, turbulent and isothermal flow. The turbulent $k$-$\varepsilon$ model was used. They investigated the effect of vapour velocity on the efficiency of a demister. They found that a vapour inlet velocity 6 m/s (range of 1-7 m/s) leads to the highest demister efficiency. They predicted the pressure drop as a function of demister properties (inlet vapour velocity, and demister; diameter, thickness, and material) and operating conditions. The model was validated with about 14-21 % deviation between experiment data and computational model</td>
</tr>
<tr>
<td>Mansour et al., (2011)</td>
<td>Predicted vapour flow through a demister using FLUENT 12.1. They investigated the effects of design parameters, e.g. vapour mass flux profile, demister porosity, and stage pressure, on the MSF flash chamber efficiency. Developed a two-dimensional flow model with a single phase and a $k$-$\varepsilon$ turbulence model was used. The flash chamber geometry was taken from an existing plant. Prediction fields of vapour velocity vectors, static pressure distribution, vapour volume fraction, vapour</td>
</tr>
</tbody>
</table>

89
temperature distribution and demister pressure drop in the flashing chamber, were presented. The results showed that higher vapour velocity through the demister led to high pressure drop; less porosity in the demister mesh led to a higher pressure drop; and a lower operating pressure of the flashing chamber led to higher inlet vapour velocity through the demister.

**Janajreh et al. (2013)** presented a numerical simulation of vapour flow and pressure drop across the demister inside a flashing chamber using FLUENT 6.1. The configuration and dimensions are based on the operating Sidi-Krir desalination plant in Alexandria, Egypt. The simulation dealt with single-phase, two-dimensional, steady state, and isothermal turbulent flow. Two main areas of the demister were investigated; pressure drop and mist removal efficiency. The study investigated the effects of different velocity profiles of the inlet vapour, and viscous and inertial resistance on the pressure drop of the demister. The study showed that the inertial resistance was the most important parameter that affected the pressure drop across the demister.

**Obeid et al. (2014)** used the same flashing chamber plant as [257] to study the vapour flow numerically through its demister. The simulation was also based on FLUENT (no version was mentioned), and dealt with single phase, two-dimensional, steady state, isothermal and incompressible flow. Flow pressure drop across demister and the effect of demister wire diameter on performance were investigated in this study. The deviation of the results from the measured data was 22%. This deviation was attributed to the following assumptions: single-phase flow and deficiencies in accounting for turbulence, flow regime and vortex formulation.

### 3.4.5.2. Review of computational predictions for the evaporation zone

Seul *et al.* (1990) [259] presented a computational model for simulating the thermal-hydrodynamic behaviour of the horizontal stream inside a multi-stage flash evaporator without a flow baffle. The PSI-CELL method of Crowe *et al.* [260] for disperse particle flows was adapted and used to solve the two-dimensional, incompressible, steady, two-phase, turbulent flow conservation equations of mass, momentum and energy in conjunction with ancillary equations for motion and growth of vapour bubbles. The liquid (continuous) phase was treated using the Eulerian field approach, whereas the vapour (disperse) phase was computed with a Lagrangian model. Bubble nuclei
entered the domain at the same velocity as the surrounding liquid, with the initial size and distribution of bubble nuclei on the inlet boundary specified as parameters. Cell-averaged bubble trajectories and cell-averaged bubble size were computed as part of the solution. The method allowed for interactions between the liquid phase and the vapour bubbles, in terms of mass transfer (by evaporation), momentum transfer via interfacial forces and energy transfer by vaporisation. Calculated results of the flow patterns, in the form of liquid streamlines, temperature distributions and bubble trajectories, were presented. Seul et al. (1990) employed a standard $k - \varepsilon$ turbulence model equation for the liquid phase only. Only the brine layer was treated, and the free surface level and profile was prescribed. The temperature boundary condition at the free surface was determined from the convective heat transfer rate with surface evaporation, based on kinetic theory. An unrealistic flat (horizontal) free surface profile was specified as a boundary condition, and interfacial stresses as well as surface tension effects were ignored in the calculations. In practice, the flow is unsteady, but even in time-average [203] the free surface can deviate significantly from being flat. No consideration was given to the vapour layer, or to interactions between the brine and vapour layers. Since the model did not have the ability to predict either the brine level or the free surface shape, the imposition of unrealistic free surface conditions will have adversely affected the predicted thermo-fluid behaviour, and the pressure distribution and flow pattern in particular.

In subsequent work, Seul et al. (1992) [261] simplified and refined their bubble dynamic model, void fraction correlation and boundary conditions, and applied their improved model to investigate the effects of the liquid level on flow behaviour inside a multi-stage flash evaporator. They confirmed that bubble motions are very much dependent on the inlet
velocity profile, and showed that the evaporation performance is improved by employing lower liquid levels. Nonetheless, they retained the free surface boundary condition as horizontal with zero shear.

Miyatake et al. (1992) [203] employed a two-dimensional, incompressible, steady, single-phase, turbulent, isothermal, stream function-vorticity numerical model in an attempt to reach a quantitative understanding of the interaction between fluid mechanics and flash evaporation in a flashing chamber with and without a flow baffle. Although a simple one-equation, mixing-length turbulence model was used, the flow field contained a number of distinctly different flow zones. For accurate flow computations, each flow zone has a characteristic (mixing) length scale which needs to be calculated. However, the number, type and shape of flow zone depends on the configuration of the flashing chamber, and are not know a priori; thus determination of such length scales is cumbersome, and requires an iterative process starting, for instance, with a laminar field solution. The curved free surface shape was obtained from experimental measurements, and prescribed as a zero shear stress boundary condition.

Miyatake et al. (1993) [178] studied the relationship between the flashing flow pattern and thermal non-equilibrium in a flashing chamber with flow baffles of different locations and heights. They post-processed the velocity and pressure distribution results from their previous study [203], and took the computed streamlines as paths along which to solve for the temperature drop between inlet and exit, for which they employed an empirical correlation for the temperature decrease rate of a flashing liquid. By averaging over several streamline they computed the average temperature of the liquid at the exit. They found that the non-equilibrium temperature difference, $NETD = T_{out} - T_v$, is reduced by increasing the brine temperature, also when the flow rate of the brine is increased.
Jin and Low (2002) [209] investigated the single-phase flow patterns in a model flash evaporation chamber both experimentally, by making particle image velocimetry measurements, and numerically, by performing complementary two-dimensional, incompressible, steady, single-phase, isothermal, turbulent flow field computations. The measured free surface profiles and inlet gate velocity distributions were used as boundary conditions, and a standard $k–\varepsilon$ turbulent flow model was employed. Using streamline plots, for a simple horizontal chamber without a baffle, they showed that the shape of the free surface as well as details of the recirculating flow pattern depend strongly on the water level, flow rate, inlet gate height and inlet velocity distribution.

Mansour et al. (2013) [262] used the CFX code to predict two-dimensional, incompressible, steady, two-phase, adiabatic, turbulent flow of the flashing process inside a flashing chamber with different numbers and locations of triangular baffles. They employed a two-fluid (liquid water and water vapour) Euler–Euler, flow formulation in conjunction with a $k–\varepsilon$ turbulence model. The flow was treated as two interpenetrating continua, with individual conservation laws for mass and momentum for each phase. The space occupied by each phase within a computational cell is defined by its local volume fraction, and is part of the flow field solution. Measures of steam mass production and pressure drop of the water were used for comparison purposes between different configurations of baffle, and the computations indicated that the location of baffles strongly affects the thermo-fluid performance. Results showing contours of vapour volume fraction and of temperature, pressure and velocity for the water component were presented. However, conventional flow patterns, in the form of streamlines or pathlines, which are valuable in assimilating and assessing field results, were not provided. The bubble formation region did not appear
in the results. It is unclear what boundary conditions were employed for each of the two fluids, and the physics and mechanism for phase change, if any, were not explained.

As already observed, flashing chamber type flows are very complex and difficult to predict. From this review of previous computational work it is evident that better models are needed that are capable of treating a number of significant aspects not hitherto allowed for. The local pressure at a point in the brine layer is influenced by its submergence below the free surface. In particular, therefore, it is desirable to have an approach that enables the shape, elevation and motion of the free surface to be predicted, rather than to have to specify it empirically, or to impose it unrealistically as a flat, horizontal boundary. Prediction of phase change and mass transfer arising both from thermal effects (vapourisation and condensation) and from pressure effects is also desirable. We next present a new computational model which aims to address these aspects.
4. CFD MODEL DESCRIPTION AND VALIDATION

4.1. Introduction

Computational field modelling can be invaluable for a further understanding of combined heat transfer, mass transfer and flashing dynamics during the flashing flow evaporation processes inside the flashing chamber in multistage flash (MSF) desalination systems.

A computational model for the flashing process inside a flashing chamber is developed around a two-phase VOF formulation. Two different phase-change mechanisms are allowed for, based on the saturation temperature and on the vapour pressure, respectively. These enable the model to compute the phase change regions, also the shape of the free surface. The model is applied to solve for steady multiphase flow inside a flashing chamber without a baffle, and thus to map the chamber flow pattern and behaviour.

We note that the method is capable of predicting the fully three-dimensional, time-dependent behaviour of the evaporation zone, which exhibits oscillating, unsteady flow phenomena characteristic of open channel flows. However, for such calculations the computational time and cost is very high: moreover, assimilating and presenting the copious results of a transient analysis is time-consuming and challenging; combined with an absence of measurement data for validation, it appears premature to allocate limited resources to such a simulation. For pragmatic reasons, therefore, we confine
our attention here to running a steady-state, two-dimensional model of the evaporation zone and to analysing the results of these predictions.

This model is applied to the evaporation zone of an existing MSF desalination system for which some data are available. Processing of the results of the flow field computation enables a visualisation of the flashing process, and phase change regions can be seen over the computational domain. The fluid process parameters of gauge pressure and particle paths (streamlines in steady flow) of flashing flow inside the evaporation zone are provided and presented. The thermodynamic process parameters of temperature distribution, vapour volume fraction, heat transfer rate, and mass transfer rate inside the evaporation zone are also shown.

In the present Chapter, our computational model description for the flashing process inside a flashing chamber is developed and validated. Our simulation results are compared with the available measured values. The simulation results give average outlet temperature and average vapour temperature that closely agreed well with the values of a real MSF plant. Two main mechanisms of phase change produced by the flashing process are captured in the simulation. The vapour bubbles are formed at the entrance to the flashing chamber, and bubble production reduces along its length. At the free surface of the liquid there is also a phase change and mass transfer.

4.2. Mathematical Modelling

The choice of multiphase model is determined by the application, and by the flow phenomena of the different phases that it is desired to capture. In this application we simplify the fluids by assuming the presence of two fluid phases only: pure water (liquid) and water vapour. For the present, therefore, real fluid aspects such brine/seawater mixture properties, brine
concentration, and the presence of entrained solids and non-condensable gases are ignored. The concentration of vapour varies widely in the evaporation zone of the flashing chamber (Figure 3). We anticipate a dilute, disperse flow of vapour bubbles in the bubble nucleation region near the brine inlet, and a very high vapour concentration from the free surface upwards. Apart from predicting the flow fields in the brine and vapour layers, we wish to capture the location of the interface between the layers. The combination of a submerged sluice, open-channel flow interacting with recirculation and phase change may be strongly unsteady, so the model chosen should be capable of handling these interactions in a robust manner.

Mathematical modelling of two-phase flashing chamber flow has been undertaken by an Euler-Euler, two-fluid approach [262], and by an Euler-Lagrange approach [259, 261]. Models for these and several other multiphase formulations are now available in commercial codes. We have selected an Euler-Euler, one-fluid approach, the Volume of Fluid (VOF) method [263-287] as implemented in the FLUENT 14.5 finite volume code [263, 264], which is the most appropriate for the present requirement: in particular, it enables the prediction of free surface flows of arbitrary interface shape.

We now outline the VOF approach used, together with the thermal and mechanical models used to predict the flashing processes inside the evaporation zone.

4.2.1 Volume of fluid (VOF) multiphase model

Numerically, there are many methods of prediction the two-fluid flow e.g. Marker and Cell (MAC) method, Volume of Fluid method and Level set method. These numerical methods have the same solver methodology; a marker particle is used to represent region of a fluid or the interface then the
governing equation of that marker particle is derived from physical principles and is a pure convection equation.

Applications such as air-water dynamics, breaking surface waves, solidification melt dynamics, and combustion and reacting flows require a two-fluid flow simulation method. In any of these applications, the interaction between the interface dynamics and the surrounding fluid motion is affected by the following factors: density ratios and temperature jumps across the interface, surface tension effects, topological connectivity and finally the boundary conditions which may significantly affect the interface dynamics.

VOF methods are conservative and can deal with topological changes of the interface. A VOF method may be used to capture the interface itself, and is available in the FLUENT software. The Level Set function is used to compute the fluid properties and the interface topology which involving in the surface tension force, but it has not been employed here.

Volume of fluid [263-287] is one of the models available in multiphase options of FLUENT software. Multi-fluids, (20 different fluids is the maximum number), can be predicted using this model. The methodology is to solve a single set of momentum equations and to track the volume fraction of each fluid throughout the computational domain.

In the general Euler-Euler multiphase approach the different phases (which may include bubbles and particles as well as multiple fluid phases) are treated mathematically as interpenetrating continua on a fixed Eulerian mesh. The concept of a phasic volume fraction is applied, where the volume fractions of the different phases are assumed to be continuous functions of space and time, and the sum of the volume fractions is unity in each cell. For
each phase conservation equations are derived, with a similar equation structure for all phases.

The free surface flow is an extremely difficult class of flows involving moving boundaries [263]. The VOF interface tracking technique is a well-established [265, 267, 272, 285] and validated [266, 267, 284] approach, designed for two (or more) immiscible fluids where it is desired to locate the position of the interfaces between the fluids.

**Figure 4.1** illustrates a group of rectangular cells in the vicinity of an interface. The liquid region is shaded. For the flashing simulation, the liquid phase fraction is presented as $\alpha_l$ while the vapour phase is represented as $\alpha_v$ in each cell throughout the domain. Three cell types can be identified: empty cells (where vapour phase, $\alpha_v = 0$), cells full of vapour phase ($\alpha_v = 1$), and cells containing the interface between vapour and liquid phase ($0 < \alpha_v < 1$). Based on the local value of $\alpha_v$ and $\alpha_l$, appropriate properties and variables will be assigned to each control volume within the domain.

**Figure 4.1** Volume fraction on a discrete mesh
At the initial time of the simulation, the position of the boundary is known. Its location at later times are to be determined as part of the solution. To reach this, the space conservation law (SCL) and boundary conditions at the free surface must be used. The SCL is used to apply the mass conservation equation in the limit of zero fluid velocity. The following expression is used to describe the conservation of space when the control volume shape and/or position changes with respect to time.

\[
\frac{d}{dt} \int_{\Omega} d\Omega - \int_{S} \mathbf{v}_b \cdot \mathbf{n} \, dS = 0 \tag{4.1}
\]

where \( t \) is time, \( \Omega \) represents the control volume, \( S \) is the control surface with unit outward normal vector \( \mathbf{n} \), and \( \mathbf{v}_b \) is the control surface velocity vector.

In the VOF model the fluids share a single set of momentum equations, so that the resulting velocity field is shared among the phases. The energy equation is also shared among the phases, and energy and temperature are treated as mass-averaged variables. Likewise, arising from solution of a single set of turbulent transport equations, the turbulence variables are shared. The volume fraction of each of the fluids, in each computational cell, is computed throughout the domain and is used to identify any interfaces that emerge as part of the solution.

The results are computed in terms of field properties of the mixture, together with the phase (or void) fraction field. This means that the VOF method does not provide separate information about each individual phase — as does an Eulerian model that solves individual momentum and continuity equations for every phase — but rather of the shared properties of the (single-fluid) mixture, which may be regarded as its main limitation.
4.2.2. Conservation equations

4.2.2.1. Momentum equation [263, 288]

A single equation of conservation of momentum is solved throughout the computational domain. This equation assumes that the cells are mixed type cells, which is the general case for all cell types. The momentum equation depends on volume fractions of all phases [269]. The volume fraction term does not appear in the equation, but mixture density, \( \rho \), and molecular viscosity, \( \mu \), terms includes the volume fraction term:

\[
\frac{\partial}{\partial t} (\rho_{\text{mix}} \mathbf{V}) + \nabla \cdot (\rho_{\text{mix}} \mathbf{V} \mathbf{V}) = -\nabla p + \nabla \cdot [\mu (\nabla \mathbf{V} + \nabla \mathbf{V}^T)] + \rho_{\text{mix}} \mathbf{g} + \mathbf{F}
\]  

(4.2),

where \( \rho_{\text{mix}} \) is mixture density \((\alpha_l \rho_l + \alpha_v \rho_v)\), \( \mathbf{V} \), mixture velocity, \( p \) is static pressure, \( \Gamma \) is diffusion coefficient, \( \rho_{\text{mix}} \mathbf{g} \) is gravitational body force, \( \mathbf{F} \) are external body forces that act due to the interaction with the dispersed phase.

The limitations of this equation [263, 288] appear in applications where differences of velocities between the phases are high. Some convergence difficulties may occur when the viscosity ratio is more than \( 10^3 \).

4.2.2.2. Energy equation [263, 289]

The energy equation deals with the mixture phase:

\[
\frac{\partial}{\partial t} (\rho_{\text{mix}} E) + \nabla \cdot (\mathbf{V} (\rho_{\text{mix}} E + p)) = \nabla \cdot (k_{\text{eff}} \nabla T) + S_h
\]  

(4.3),
Where, $E$ is total energy, $k_{\text{eff}}$ is effective conductivity of the mixture, $T$ is temperature of the mixture, and $S_n$ is the source term contains contributions from radiation and any other volumetric heat sources.

The effective conductivity [263] consists of two terms; $k^*$, thermal conductivity and $k_t$ turbulent thermal conductivity:

$$k_{\text{eff}} = k^* + k_t \quad (4.4),$$

For the realisable $k, \varepsilon$ model, $k_{\text{eff}} = k^* + \frac{c_p \mu_t}{P r_t}$, where, $c_p$ is specific heat, $Pr$ molecular Prandtl numbers.

Both $E$ and $T$ are treated in FLUENT as mass-averaged variables [263]:

$$E = \frac{\sum_{q=1}^{z} \alpha_j \rho_j E_j}{\sum_{q=1}^{z} \alpha_j \rho_j} \quad (4.5),$$

where $z$ is the total number of phases in the system (e.g. two-phase system $z = 2$), subscript $j$ is an index which increases by 1 for each successive term, stopping when $j = z$, $E_j$ represents the total energy for each phase. It is based on both the specific heat of the phase and the shared temperature.

In the turbulent flow case [289], the velocities in equations (4.2 and 4.3) represent statistically average velocities. For the momentum equation, the Reynolds stress term appears as an additional term. These stresses are modelled using an eddy viscosity approach, such as the $k, \varepsilon$ model.
4.2.3. Phase change models

The phase change models based on local thermal (saturation temperature) and mechanical (vapour pressure) effects, respectively, which are used for building the flashing chamber simulation, are presented in this section.

4.2.3.1. Phase change based on thermal effect

The mechanism of interphase mass transfer between the liquid and the vapour phases is controlled by the vapour mass transfer equation [263, 290].

\[
\frac{\partial}{\partial t} (\alpha_v \rho_v) + \nabla \cdot (\alpha_v \rho_v \vec{V}_v) = \dot{m}_{l \rightarrow v} - \dot{m}_{v \rightarrow l} \tag{4.6},
\]

where subscript \( v \) = vapour phase, \( \alpha \) = volume fraction, \( \rho_v \) = vapour density, \( \vec{V}_v \) = vapour phase velocity vector, \( \dot{m}_{l \rightarrow v} \) = mass transfer rate from liquid to vapour phase, \( \dot{m}_{v \rightarrow l} \) = mass transfer rate from vapour phase to liquid.

The Lee Wen Ho model [291] is a mechanistic model representing phase change from liquid to vapour phase, as well as from vapour to liquid phase, with a physical basis. The mechanism of the vapourisation-condensation model is based on checking the temperature of the liquid phase. Since evaporation is a surface phenomenon, mass transfer of the molecules takes place at a liquid-vapour interface. When molecules have enough kinetic energy, they can escape from the free surface as a vapour phase. This kinetic energy is a function of the saturation temperature of the liquid. At higher temperatures, molecular kinetic energy is greater.

The local saturation temperature corresponding to the local pressure of the system is considered as an indicator for this model. In other words, the vapourisation process takes place when the local temperature of the liquid phase is higher than the local saturation temperature, while the condensation
process takes place when the temperature of the vapour phase is higher than
the saturation temperature. Thus

if \( T_l > T_{sat} \) (vapourisation), then

\[
m_{l\rightarrow v} = (coeff) \alpha_l \rho_l \frac{(T_l - T_{sat})}{T_{sat}}
\]  \( (4.7) \)

If \( T_v \leq T_{sat} \) (condensation), then

\[
m_{v\rightarrow l} = (coeff) \alpha_v \rho_v \frac{(T_v - T_{sat})}{T_{sat}}
\]  \( (4.8) \)

where the coefficient, \( coeff \), must be adjusted carefully in order to achieve
the best description of performance. It also should be interpreted as a
relaxation time. The coefficient factor can be described in the following
expression

\[
coeff = \frac{6}{d} \beta \sqrt{\frac{M}{2\pi RT_{sat}}} LH \left( \frac{\rho_l}{\rho_l - \rho_v} \right)
\]  \( (4.9) \)

where \( \beta \) is the accommodation coefficient, which defines the proportion of
vapour molecules being adsorbed at the liquid side of the interfacial surface:
\( \beta \) is a physical characteristic of the gas, and has a value of 1.0 near
equilibrium conditions; \( d \) = vapour bubble diameter, \( M \) = molecular weight,
\( R \) = universal gas constant, \( LH \) = latent heat, \( v \) = vapour phase, \( \alpha \) = vapour
volume fraction, \( \rho_l \) = liquid density. The vapourisation-condensation flux is
based on the Hertz Knudsen kinetic theory formula for a flat interface. In
calculating the interfacial area density it is assumed that the flow regime is
disperse, also that all vapour bubbles are spherical with the same diameter.
Here, to enable initial computations in the absence of accurate data for the diameter of the vapour bubbles, and the value of \( \text{coeff} \) is set to 0.1, thus obviating the need to define the diameter.

### 4.2.3.2. Phase change based on mechanical effect

Many studies [292-305] provided several mathematical models that describe the phase change based on pressure of a liquid. In our computational prediction, the Zwart-Gerber-Belamri model is used.

The mass transfer mechanism between liquid-vapour phases are based on the vapour transport equation [263]:

\[
\frac{\partial}{\partial t} (\alpha_v \rho_v) + \nabla \cdot (\alpha_v \rho_v \mathbf{V}_v) = R_e - R_c
\]  

(4.10),

where \( v = \) vapour phase, \( \alpha_v = \) vapour volume fraction, \( \rho_v = \) vapour density, \( \mathbf{V}_v = \) vapour phase velocity vector, \( R_e = \) mass transfer source term connected to the growth of the vapour bubbles, \( R_c = \) mass transfer source term connected to the collapse of the vapour bubbles. In the FLUENT cavitation evaporation-condensation model these mass-transfer source terms are modelled based on the Rayleigh-Plesset equation, which describes the growth of a single vapour bubble in a liquid. It is assumed that there are plenty of nuclei present for inception of cavitation, and the primary focus is on accounting for bubble growth and collapse.

The Zwart et al model [302], which is derived from the generalised Rayleigh-Plesset equation [306], is based on the assumption that bubbles in the system are of equal size. While the generalised Rayleigh-Plesset equation includes the influences of surface tension, liquid viscosity and inertia on bubble growth, these terms are absent from Zwart et al model. In the present
application, therefore, bubble growth is assumed to be dominated by thermal effects.

In the Zwart et al model the bubble density numbers \( n \) and the mass change rate of a single bubble are used for calculating the total interphase mass transfer rate per unit volume \( (R') \):

\[
R' = n \left( 4\pi R_b^2 \frac{\rho_v}{\rho_l} \frac{DR_b}{Dt} \right) \quad (4.11),
\]

where \( R_b \) is the bubble radius. To determine whether the bubble will nucleate and grow, or collapse and disappear, an indicator formula is required. A critical radius for the bubble is the key to creating such an indicator. The following indicator is used in the model for this purpose:

If the local absolute pressure \( p \leq p_v \), then

\[
R_e = F_{vap} \frac{3 \alpha_{muc} (1 - \alpha_v) \rho_v}{R_b} \frac{\sqrt{2 \frac{P_v - P}{\rho_l}}}{3 \rho_l} \quad (4.12)
\]

If \( p \geq p_v \), then

\[
R_c = F_{col} \frac{3 \alpha_v \rho_v}{R_b} \frac{\sqrt{2 \frac{P - P_v}{\rho_l}}}{3 \rho_l} \quad (4.12),
\]

Here the generalised bubble radius \( R_b \) is set at \( 10^{-6} \) m, the vapourisation coefficient \( F_{vap} \) is set to 50, and the condensation coefficient \( F_{col} \) is set to 0.01. By setting a volume fraction value for nucleation sites as input, in conjunction with the generalised bubble radius the bubble density number is determined. Here, for initial computations, the volume fraction of nucleation sites is specified as 0.0005.
4.3. Validation Case Study [262, 256]

The present work is built and validated based on modelling the evaporation zone in one stage of an existing MSF plant. The data used are for the Sidi Krir Plant at Alexandria, Egypt. The geometry used to define the computational domain and the operating conditions are taken from published papers [262, 256]. The MSF plant has 20 stages, with a total production of desalinated water of 5,000 m$^3$/day.

The shape and dimensions of the computational domain can be seen in Figure 4.2. Here we simulate the evaporation zone in the first flashing chamber of this plant. Operating conditions of the first flashing chamber are:

1. Average inlet temperature of brine 383.15 K
2. Average outlet temperature of brine 379 K
3. Mass flow rate of the brine 1847 ton/hr
4. Flashed vapour 12.5 ton/hr
5. Flashing chamber operating pressure 1.023 bar

The mass flow rate and inlet temperature of the brine are used to define inputs to the model, applied as boundary conditions, while for validation the vapour production and temperature field are computed and compared with measured values [262].
Figure 4.2 Computational domain: evaporation zone dimensions and boundary conditions (source data [262])
4.3.1. Computational case Setup

The dimensions of the flashing chamber are typical of an existing stage of a MSF desalination plant (*Figure 4.2*), with a streamwise length of 1.25 m and an inlet gate height of 60 mm. The computational domain is bounded by walls on three sides, with one inlet and one outlet each for the brine. Note that the upper (‘open’) boundary for vapour is situated within the vapour zone, 0.35 m above the floor of the chamber: the location of the free surface, which is unknown *ab initio*, will be determined by the operating conditions, and is computed as part of the solution.

Quadrilateral rectangular shape elements (see *Figure 4.3*), with six inflation layers near the walls, are used for mesh generation. The total number of elements is 201,358 (40 elements along the inlet gate).

*Figure 4.3*  Typical computational domain mesh inside the flashing chamber
Our assumptions for the computational case are as following: 2D, steady state, two-phase, adiabatic, and turbulent flow. The two phases are water liquid and water vapour phases. All walls are assumed to be smooth; the effect of surface roughness is not considered here.

In this configuration we anticipate a wall jet developing along the floor of the chamber, and wall regions experiencing flow separation, reattachment and impingement. Thus, since much of the flow may be wall-dominated, near the solid surfaces the mesh is refined and extended into the viscous sublayer. The mesh is generated such that along the walls the estimated non-dimensional wall-normal distance to the centre of the first cell, $y^+$, is less than 5 (see Figure 4.4): this enables enhanced wall functions [263] to be used along with the $k$-$\varepsilon$ turbulence model to better resolve the flows in these near-wall regions.

Figure 4.4  $y^+$ distribution along the boundary bottom wall of the flashing chamber
4.3.1.1. Boundary conditions

The boundary conditions applied are shown in Figure 4.2. A uniform inlet velocity profile of 1.5 m/s at 383.15 K is set, with a uniform vapour fraction of zero and standard default values for the turbulent parameters. The base and vertical walls have no-slip and are adiabatic. The brine outlet and open boundary are set as uniform pressure boundaries with uniform vapour fractions of zero and one, respectively.

For this low-speed, incompressible flow it is appropriate to employ a pressure-based solution algorithm. In this algorithm the pressure is derived from iterative solution of the continuity and momentum equations, and is computed as a gauge-pressure field. Thus all boundary condition pressures (Figure 4.2) are specified to FLUENT as gauge pressures \( p' \). To enable absolute pressures to be produced a reference pressure is set at a reference point in the domain. The absolute pressure is then calculated as the sum of the gauge pressure and the reference pressure. Here the reference pressure is set, on the top of the vapour layer at a point on the open boundary, to the vapour pressure of the flashing chamber. The open boundary gauge pressure is therefore set to zero. We note that when gravitational acceleration is active in FLUENT, as in the present computations, the hydrostatic component of pressure is included in the gauge pressure field. Here, we should clarify that the vapour pressure is a function of the local temperature for each cell.

The values of fluid properties are estimated from [307, 308].
Table 4.1 Water (liquid and vapour) phase properties

<table>
<thead>
<tr>
<th>Material</th>
<th>Properties</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water (liquid)</td>
<td>Density ( \rho_l ) = 950.6 kg/m³</td>
</tr>
<tr>
<td></td>
<td>Specific heat ( c_p ) = 4229 J/kg·K</td>
</tr>
<tr>
<td>( T=110 , ^\circ C )</td>
<td>Thermal conductivity ( k^* ) = 0.682 W/m·K</td>
</tr>
<tr>
<td></td>
<td>Dynamic viscosity ( \mu ) = 0.2558 x 10⁻³ kg/m·s</td>
</tr>
<tr>
<td></td>
<td>Molecular weight = 18.0152 kg/kgmol</td>
</tr>
<tr>
<td>Water (vapour)</td>
<td>Density ( \rho_v ) = 0.8263 kg/m³</td>
</tr>
<tr>
<td>( T_{st} ) = 109.32 °C</td>
<td>Specific heat ( c_p ) = 2071 J/kg·K</td>
</tr>
<tr>
<td></td>
<td>Thermal conductivity ( k^* ) = 1.261 x 10⁻⁵ W/m·K</td>
</tr>
<tr>
<td></td>
<td>Reference temperature = 298.15 K</td>
</tr>
</tbody>
</table>

4.3.1.2. Numerical schemes and convergence criteria

An implicit pressure-based solver is used in the VOF model to obtain a steady-state solution. While in reality the flow will be time-dependent, for the present, however, to simplify the analysis and processing of results we confine consideration to the imposition of an artificial steady state. To resolve the pressure-velocity coupling the FLUENT implementation of the PISO (Pressure Implicit with Splitting of Operators) algorithm is employed, and both the neighbour and skewness corrections are used to improve the efficiency of PISO calculations [287, 309]. FLUENT uses a collocated scheme in which all variables, including velocity and pressure, are stored at cell centres. Interpolation is therefore required to compute the corresponding values of variables and of their gradients at cell face locations. A least-squares cell-based interpolation method is used for computing the gradients. The PRESTO scheme is employed for pressure interpolation. For improved accuracy, second-order upwind differencing is applied for the spatial discretisation of the convective terms in the momentum, volume fraction, turbulence variables and energy equations. During each iteration, for each cell the mass-transfer and energy interaction between the two phases are calculated, then returned as source terms to the particular transport
equations. Under-relaxation factors for all variables were set to values between 0.2 and 0.5 for improving the solution stability. Appendix D provides more details about solution methodology.

The convergence criteria are as follows: for each of the $x$-velocity, $y$-velocity, $k$, $\varepsilon$ and volume-fraction-vapour equations it is $10^{-5}$, for the continuity equation it is $5 \times 10^{-4}$ and for the energy equation it is $10^{-6}$.

The simulation was run on an Intel Core2 quad-CPU Q6600 of 2.4 GHz. Three of the available four cores were used to run this model. The simulation requires about 15,000 iterations to converge, each iteration requiring about 3 secs to complete.

\begin{figure}
\centering
\includegraphics[width=\textwidth]{residuals.png}
\caption{Examples of plot of residuals: (a) solution converging, (b) solution converged}
\end{figure}
4.3.2 Validation

In this section, a visualisation of the flashing process and phase changer regions inside the flashing chamber, an evaluation of some operating data (e.g. exit temperature, vapour temperature and flashed vapour), and evaluation of the physics behind the flashing phenomena and some design factors are presented.

4.3.2.1. Mesh Resolution

Dealing with multiphase flow with phase change and free surface prediction has led us to be more critical with our choice of mesh size. The mesh size should be sufficiently small and gradual (smooth) without any sudden change in cell size, to avoid missing any phase change zones and resolving flow fields.

Initially, we started with a coarse mesh to ensure that results were independent of mesh spacing, based on Mansour’s et al. recommendation [262]. Subsequently, several meshes of quadrilateral rectangular shape elements were tested. The mesh sizes investigated were: coarse (1.6 x 1.6 mm), medium (1.4 x 1.4 mm), and fine (1.2 x 1.2 mm). Each mesh was tested using our current model (§ 4.2) with the same boundary and initial conditions as presented in Figure 4.2. Predicted values of pertinent variables were extracted along a ‘random’ vertical traverse through the flashing chamber at $x = 0.6$ m, and plotted for comparison with each other. These included temperature, velocity magnitude, and vapour volume fraction (as shown in Figures 4.6, 4.7 and 4.8). We found the maximum differences of the predicted results between the three different mesh cases are as following: for the temperatures distribution, it was less than 0.03 °C, while for the velocity distribution, it was less than
0.05 m/s, and for the vapour volume fraction distribution, it was less than 0.00625. These results are indistinguishable, independent of the mesh spacing. To ensure reaching the most accurate results, we compared the outlet average temperature for the three different mesh sizes; we found the predicted values as following: the coarse mesh case is 378.5 K, the medium mesh case is 378.6 K, and the fine mesh case is 378.7 K (the figures are provided in Appendix E). The provided outlet temperature from the real plant is 379 K [262], thus the fine mesh case was the more accurate, and so we used the fine mesh size in all our study cases.

![Figure 4.6 Influence of mesh size on temperature along a vertical traverse through the flashing chamber at x/L = 0.5](image.png)
Figure 4.7 Influence of mesh size on mixture speed along a vertical traverse through the flashing chamber at \( x/L = 0.5 \)

Figure 4.8 Influence of mesh size on vapour volume fraction along a vertical traverse through the flashing chamber at \( x/L = 0.5 \)
4.3.2.2. An existing flashing chamber of a MSF plant

The main challenges for doing such predictions lies in obtaining sufficient data to build and to validate the computational model. In particular, it is difficult to find information pertaining to the interior of the flashing chamber, i.e. bubble nucleation rate, bubble formation, recirculation zone size and length, brine level, and also the shapes of the orifices and their number. Also, for the few data that are available, there is little or no information about measurement locations, methods or uncertainties.

The VOF [267] multiphase model is well established, and has been validated for predicting liquid-gas interfaces, including the free surface shape and levels, in a variety of other applications [266, 267, 284]. Tamilarasan [287] has validated the mechanistic Lee Wen Ho vapourisation-condensation model for vapour volume fraction prediction, including its application within the Fluent VOF implementation. We have independently validated the implementation of the Zwart et al vapourisation-condensation model within the Fluent VOF code using data from detailed and extensive experiments for isothermal flashing water flow in a converging-diverging nozzle by Abuaf et al [310]. Under non-equilibrium (flashing) conditions, the measured average vapour volume fraction at the outlet of the nozzle was 0.544, for which case our predicted value is 0.543 [See Appendix F]. The validation shows that the method can reproduce some of the necessary physics with accuracy over the field, not only at inlets and outlet.

Our simulation of the evaporation zone of the flashing chamber is validated by checking values of average temperature at the stage exit, average vapour temperature above the liquid free surface and the vapour mass flow rate (more details is presented in following Chapter 5). In comparing our results
with the available data [262], we find a good agreement with the values for the real plant (*Table 4.2*).

**Table 4.2** Comparison of predicted results and measured values

<table>
<thead>
<tr>
<th></th>
<th>Measured [262]</th>
<th>Predicted</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Average outlet temperature [K]</strong></td>
<td>379</td>
<td>378.6</td>
</tr>
<tr>
<td><strong>Average vapour temperature [K]</strong></td>
<td>375</td>
<td>377.7</td>
</tr>
<tr>
<td><strong>Total flashed vapour [tonne/hour]</strong></td>
<td>12.5</td>
<td>13.6</td>
</tr>
</tbody>
</table>

The predicted flashing mechanism in the simulation agrees with the mechanism described in Chapter 3 (see *Figure 3.7*). As seen in *Figure 4.9* vapour bubbles are formed at the entrance to the flashing chamber, and bubble production diminishes along its length. At the free surface of the liquid there is also a phase change and mass transfer. Thus the two main mechanisms of phase change produced by the flashing process are captured in the simulation.

The predicted level of brine inside the flashing chamber is 0.2m above the inlet. This agrees with design recommendations by El-Dessouky *et al*. (2002) [63], who recommended that the brine pool be higher than the gate height by 0.2m.

The calculated non-equilibrium temperature difference (*NETD*) values range from 4.8 to 5.5°C, which is found to fall within the range of design recommendations [176]: the *NETD* for a flashing chamber without a baffle lies between 4 and 5°C.
Figure 4.9. Flashing regions as predicted by computational model. Compare with Figure 3.7
4.4. Summary and Discussion

A numerical procedure for flashing flow was developed and was applied to predict multiphase flow field. Our model was multiphase, based on the FLUENT VOF code implementation, and encapsulates two interaction mechanisms for phase change — based on thermal and mechanical effects, respectively — that arise during the flashing process.

To ensure mesh independence, the model was tested for three different sizes. To ensure sufficient resolution, the fine grid size was used in all study cases.

The model was evaluated for the application of MSF desalination system. For these computations the flow was treated as steady. The free surface shape and level were allowed to be estimated by the computational modelling. A visualisation of the flashing process and phase change regions can be seen over the prediction field. It was found that the simulation result agreed with the values of the real plant.

From the present work it is clear that computational field modelling can be invaluable for a better understanding of the combined heat transfer, mass transfer and the fluid dynamics during the flashing flow evaporation processes. We note that our method is capable of predicting the fully three-dimensional, time-dependent behaviour of the evaporation zone, which exhibits oscillating, unsteady flow phenomena characteristic of open channel flows. While feasible, for such calculations the computational time and cost is very high: moreover, assimilating and presenting the copious results of a transient analysis is time-consuming and challenging. Despite this potential, as already noted in §4.3.2.2 the key obstacle nonetheless remains the lack of measurements and data for validation.
In the interim, the current model, run in a steady, two-dimensional mode, will be used for studying and comparing the effects of variations in the brine temperature, velocity and flashing chamber geometry on the thermo-fluid behaviour inside the evaporation zone (as presenting in the following Chapters 5 – 9).

As already noted in Chapter 3, the flashing process may be classified as a finite or infinite (flashing) process. Depending on which of the two flashing process variations occurs — finite or infinite — may determine the thermo-fluid behaviour inside the evaporation zone inside the flashing chamber, and will also be investigated in Chapter 5. This will be presented in the following Chapter by comparing predicted results with regard to the thermo-fluid performance; gauge pressure, flow patterns, thermal performance, vapour volume fraction and the mass transfer inside the flashing chamber.
5. CFD PREDICTION OF FLASHING PROCESSES IN A MSF DESALINATION CHAMBER

5.1. Introduction

The predicted flashing mechanism in the simulation agreed with the mechanism stated in the review section. Bubbles are formed at the entrance of the flashing chamber and reduce along the length of the chamber. At the surface of the liquid, a phase change and mass transfer occurs. In this way, the two main mechanisms of phase change by the flashing process are captured in this simulation.

In this Chapter, a computational model is applied on two different operation conditions. The first, where the flashing chamber operates under infinite flashing conditions, which is the normal operating condition in an existing MSF system (operating pressure is 1.023 bar). The second where the flashing chamber operates under a finite flashing condition (operating pressure is 1.4 bar).

For each case, the thermodynamic process parameters of heat transfer rate, mass transfer rate, temperature distribution and vapour volume fraction inside the evaporation zone are provided. Furthermore, the fluid process parameters of particle paths (streamlines in steady flow) of flashing flow and gauge pressure inside the evaporation zone are studied. Non-equilibrium losses (*NETD*), flashing down and flashing efficiency are calculated.
5.2. Flashing Chamber Operating Under an Infinite Flashing Condition

Normally, in existing plants, flashing chambers are run under an infinite flashing condition. Thus, the outlet brine temperature is always higher than the saturation temperature, corresponding to the operation pressure of that flashing chamber. Therefore, non-equilibrium losses for the flashing chamber are positive.

The operating data for the current case is provided from a real plant. The first stage runs under atmospheric pressure; however the rest of the stages run under lowered pressure. The operation pressure for the present flashing chamber is 1.023 bar. The predicted steady-state results of the thermo-fluid performance (gauge pressure, flow patterns, thermal performance, vapour volume fraction and the mass transfer) inside the flashing chamber are presented in Figures 5.1 through to 5.6.

5.2.1. Gauge pressure distribution

*Figures 5.1a, 5.1b* and *5.1d* show the gauge pressure distribution throughout the flashing chamber. A discontinuity in pressure gradient is evident between the brine and the vapour layers, and the lower edge of this discontinuity is taken to be the free surface. The free surface is not well defined because there is no discontinuity in the liquid void fractions. It is difficult to determine the free surface due to the effects of evaporation and to the continuum nature of the void fraction. In all of our cases, the free surface is defined from the pressure contour, and assigned an arbitrary value of 160 Pa. This has been achieved by using a Fluent “polyline”.

There is an essentially linear increase in pressure with depth below the free surface. Along a horizontal traverse at mid-orifice level (*y* = 0.03m, *Figure*)
the calculated pressure drop is 2179 Pa, corresponding to the pressure difference required to drive the flow from the inlet to the outlet.

Figure 5.1a Predicted gauge pressure field (Case: \( v_x = 1.5 \) m/s, \( p_{op} = 1.023 \) bar, \( T_{in} = 383.15 \) K)

Figure 5.1b Gauge pressure distribution along vertical traverses at \( x = 0.31 \) m, 0.62 m, 0.93 m (Case: \( v_x = 1.5 \) m/s, \( p_{op} = 1.023 \) bar, \( T_{in} = 383.15 \) K)
Figure 5.1b  Gauge pressure distribution along horizontal traverses at $y=0.03\text{m}, 0.17\text{m}, \text{free surface line}, 0.3\text{m}$ (Case: $v_x = 1.5 \text{ m/s}$, $p_{op} = 1.023 \text{ bar}$, $T_{in} = 383.15 \text{ K}$)

5.2.2. Particle paths and velocity distribution

Figure 5.2a shows flow fields: shows flow fields: (i) velocity magnitude, (ii) vector map with fixed length vectors, (iii) vector map with vector length proportional to velocity magnitude, and (iv) particle paths (streamlines), coloured by the speed of the fluid mixture, throughout the flashing chamber. A detailed view of the flow pattern in the inlet region is also provided. As commonly observed in flashing chamber experiments [62, 193-200], a distinct brine recirculation flow zone appears in the solution.

The superheated brine enters the evaporation zone inside the flashing chamber with bubble formation in bubble nucleation region. In this region where two-phase flow (brine and vapour), and where the phase changes rapidly from liquid to vapour, the vapour bubbles rise up due to the buoyancy and grow, and it escape from the free surface, the brine flows along the channel flow region then it leaves the flashing chamber.
The mixture pathlines above the free surfaces as shown in the *Figure 5.2a* are predominantly normal to the free surface. The magnitude of the velocity near the free surface is very low since the tangential shear stress is very low. The mixture contains more vapour than liquid phase above and away from the free surface, which increases the speed due to the mixture density effect.

The movements of the fluid in the bubble nucleation region are relatively high as shown in *Figure 5.2b, 2c*. It decreases along the channel flow region but it accelerates to maximum at the brine exit due to the area reduction.

The directions of the mixture flow can be seen in *Figure 5.2a*. Flow directions are function of the free surface fluctuation and the boundary layers. The directions of the mixture particles near the walls are downwards. This is due to the return particles because of the condensation process.

The vertical traverses (*Figure 5.2b*) indicate a sharp discontinuity in velocity between the brine and vapour along the interface, with minima occurring at the free surface. Along the mid-orifice level (*Figure 5.2c*) the inlet flow decelerates due to jet spreading and subsequently accelerates to a maximum of 1.88 m/s at the brine exit.
Figure 5.2a Predicted flow fields: (i) velocity magnitude, (ii) vector map with fixed length vectors, (iii) vector map with vector length proportional to velocity magnitude, and (iv) particle paths (streamlines) (Case: $v_x = 1.5$ m/s, $p_{op} = 1.023$ bar, $T_{in} = 383.15$ K)
Figure 5.2b  Mixture speed along vertical traverses at $x = 0.31$ m, 0.62 m, 0.93 m  
(Case: $v_x = 1.5$ m/s, $p_{op} = 1.023$ bar, $T_{in} = 383.15$ K)

Figure 5.2c  Mixture speed along horizontal traverses at $y = 0.03$ m, 0.17 m, free surface line, 0.3 m  
(Case: $v_x = 1.5$ m/s, $p_{op} = 1.023$ bar, $T_{in} = 383.15$ K)
5.2.3 Thermal performance

The rate of flashing is related to the thermal performance, or brine temperature field. The flashing process across the flashing chamber causes a reduction of brine temperature, as can be seen in the overall temperature field (Figure 5.3a), and also in the vertical and horizontal temperature traverses presented in Figures 5.3b and 5.3c.

The decrease in temperature takes place in both the horizontal (x-) and vertical (y-) directions, but at different rates. The differences in range and scale between Figures 6b and 6c should be noted. The temperature gradient in vertical (y-) directions is large compared to the (x-) horizontal. The calculated flashing down \((T_{in} - T_{out})\) is 4.55 °C.

\[
\begin{align*}
  v_x & = 1.5 \text{ m/s} \\
  p_{op} & = 1.023 \text{ bar} \\
  T_{in} & = 383.15 \text{ K}
\end{align*}
\]

**Figure 5.3a** Predicted temperature field (Case: \(v_x = 1.5 \text{ m/s}, p_{op} = 1.023 \text{ bar}, T_{in} = 383.15 \text{ K}\)
Figure 5.3b  Temperature distribution along vertical traverses at $x = 0.31\text{m}$, $0.62\text{m}$, $0.93\text{m}$ (Case: $v_x = 1.5 \text{ m/s}$, $p_{op} = 1.023 \text{ bar}$, $T_{in} = 383.15 \text{ K}$)

Figure 5.3c  Temperature distribution along horizontal traverses at $y = 0.03\text{m}$, $0.17\text{m}$, free surface line, $0.3\text{m}$ (Case: $v_x = 1.5 \text{ m/s}$, $p_{op} = 1.023 \text{ bar}$, $T_{in} = 383.15 \text{ K}$)
5.2.4. Vapour volume fraction and mass transfer rate

*Figure 5.4a* presents the distribution of vapour volume fraction \((\alpha_v)\) throughout the flashing chamber, and for clarity two field images are provided. The upper image (i) indicates the full range of vapour volume fraction scale from 0 to 1, and shows that above the free surface there is a mixture of vapour and liquid phases. The lower set of images (ii) focuses on a restricted range, between 0 and about 0.0001, to better illustrate the vapour phase distribution throughout the evaporation zone. The peak values of vapour volume fraction occur at the free surface and in the bubble nucleation region.

Along vertical traverses above the free surface (*Figure 5.4b*), the vapour volume fraction increase monotonically. Horizontally, (*Figure 5.4c*) the values of vapour volume fraction decrease. The average vapour volume fraction along the free surface is \(6.84 \times 10^{-3}\). *Figure 5.4d* shows the distribution of vapour volume fraction along \(y = 0.03\) m, which passes through the bubble nucleation region. This happens because of the reduction in brine temperature, so the phase change rate is decreased.

*Figure 5.5* shows the mass transfer rate in the bubble nucleation region and at the free surface layer inside the flashing chamber. The positive value means that the phase changes from liquid to vapour phase (vapourisation process is occurred) while the negative value means the phase changes from vapour to liquid phase (condensation process has occurred).
Figure 5.4a  Vapour volume fraction ($\alpha_v$) fields with two different ranges of scale; (i) upper field range $[0 - 1]$, and (ii) lower field range $[0 - 9 \times 10^{-5}]$ (Case: $\nu_i = 1.5$ m/s, $p_{op} = 1.023$ bar, $T_{in} = 383.15$ K)
Figure 5.4b  Vapour volume fraction distribution along vertical traverses at $x=0.31\text{m}$, $0.62\text{m}$, and $0.93\text{m}$ (Case: $v_x = 1.5 \text{ m/s}$, $p_{op} = 1.023 \text{ bar}$, $T_{in} = 383.15 \text{ K}$)

Figure 5.4c  Vapour volume fraction distribution along horizontal traverses at $y = 0.03\text{m}$, $0.17\text{m}$ and $0.3\text{m}$ (vapour region), and along free surface ($y$ varies) (Case: $v_x = 1.5 \text{ m/s}$, $p_{op} = 1.023 \text{ bar}$, $T_{in} = 383.15 \text{ K}$)
Figure 5.4d  Vapour volume fraction distribution along horizontal travers $y=0.03\text{m}$ (Case: $v_x = 1.5 \text{ m/s}$, $p_{op} = 1.023 \text{ bar}$, $T_{in} = 383.15 \text{ K}$)
Figure 5.5  Mass transfer rate field with three different ranges of scale; (a) $[-190, +34]$ kg/m³·sec, (b) $[-26, +8]$ kg/m³·sec, and (c) $[-0.0016, +2.5]$ kg/m³·sec (Case: $v_x = 1.5$ m/s, $p_{op} = 1.023$ bar, $T_{in} = 383.15$ K)
5.2.5. Non-equilibrium losses

*Figure 5.6a* shows the non-equilibrium loss (NETD) distribution over the flashing chamber outlet orifice. The NETD increases almost linearly from the top to the bottom of the orifice (i.e. with depth of the brine), ranging from 4.8 to 5.59°C. *Figure 5.6b* shows the relationship between the NETD ($T_{in} - T_{sat}$) and flashing down ($T_{in} - T_{out}$). When flashing down increases, the NETD decreases. The range of flashing down is from 4.17 to 4.88°C. *Figure 5.6c* shows the relationship between the NETD and flashing efficiency. The flashing efficiency increases when the NETD decreases. Thermal flashing efficiency ranges from to 0.425 to 0.5.

![NETD distribution over outlet orifice](image)

*Figure 5.6a  NETD distribution over outlet orifice (Case: $v_x = 1.5$ m/s, $p_{op} = 1.023$ bar, $T_{in} = 383.15$ K)*
Figure 5.6b  NETD distribution over flashing down (Case: $v_x = 1.5 \text{ m/s}$, $p_{op} = 1.023 \text{ bar}$, $T_{in} = 383.15 \text{ K}$)

Figure 5.6c  NETD distribution over flashing efficiency (Case: $v_x = 1.5 \text{ m/s}$, $p_{op} = 1.023 \text{ bar}$, $T_{in} = 383.15 \text{ K}$)
5.3. Discussion

5.3.1 Gauge pressure distribution

The pressure distribution decreases dramatically (large pressure change over a very small height) across the free surface as it is shown on the vertical traverses in Figure 5.1b, and this pressure distribution behaviour agrees with what was observed by [318].

Gauge pressure resists the flashing process [261] by increasing vapour pressure with depth, which affects the saturation temperature. Thus mass transfer and phase change decrease with depth due to the increase in pressure.

The gauge pressure has the same effect as the brine level on the flashing performance. The increase of the brine level leads to higher thermodynamic non-equilibrium losses in the flashing chamber. Also, the increase of the brine level leads to an increase in the pressure with depth.

Generally, gauge pressure [261] affects both the recirculation zone shape and locations (it increases linearly with the brine level), and bubble growth and rise locations. However, in our simulation case, the gauge pressure is not enough to block the bubbles formation and the reduction in brine temperature.

5.3.2. Particle paths (streamlines) and velocity distribution

Using the VOF in our simulations does not allow one to study the velocity distribution separately for each phase in the mixture case. Thus the velocity distribution in Figure 5.2 is for the mixture only. However, the VOF model makes it possible to deal with the vapour layer as well with the brine layers.
The velocity magnitude at the free surface is the minimum. This happens in order to keep conservation of mass concept over the free surface valid. For $x = 0.93$ (Figure 6b) the value of the mixture speed is higher comparing with $x = 0.31, 0.62$ m. This is due to the higher phase changes rate and thus more vapour phase included in the mixture as it can be noticed in Figure 5.2a at the lower field. Horizontally, the maximum speed is found to be 1.88 m/s at the end of the flashing chamber. This happens to keep the conservation of energy concept valid, since pressure in this point has reached the minimum value.

The recirculation zone promotes bubble transport from the bubble nucleation region to the free surface. Other information about the relationship between fluid flow and its thermal behaviour can be reached by using such simulations, i.e. rate at which turbulence kinetic energy converts into thermal internal energy. This can provide information about vertical heat flow, which is required in order for evaporating liquid close to the free surface.

5.3.3. Thermal performance

Generally, thermodynamics of the brine in the evaporation zone inside the flashing chamber is difficult to predict [311]. However, our computational model for the evaporation zone can be used to obtain more information about the thermodynamics of the brine inside the flashing chamber. We found that when the horizontal rate of decrease of temperature is greater than the vertical rate of decrease of temperature ($\frac{dT}{dx} > \frac{dT}{dy}$), the non-equilibrium temperature difference (NETD) will be positive, while for $\frac{dT}{dx} < \frac{dT}{dy}$, the NETD will be negative.

The reduction in brine temperature in the evaporation zone occurs due to the change phase. This reduction in temperature affects the mass transfer rate
and vapour volume fraction distribution (see Figures 5.4, 5.5). Also the mixing process affects the rate of temperature decrease.

In our simulation, the temperature gradient in the vertical (y-) direction is large compared to the (x-) horizontal. The turbulent shear mixing between the upper and lower brine layers influences the local temperature gradient. The upper layer is colder than the lower, and since the horizontal temperature gradient is small compared to the vertical gradient, the preferred direction of heat transfer is vertical rather than horizontal.

In our current simulation (flashing chamber operating under an infinite flashing condition) it is evidenced that the driving force of the phase change — the temperature difference between the brine temperature and the saturation temperature — along the evaporation zone has remained positive. In other words, while the phase change reduces the brine temperature, its values are still higher than the saturation temperature corresponding to the flashing chamber pressure, thus maintaining the phase change along the evaporation zone length.

5.3.4. Vapour volume fraction and mass transfer

The vapour volume fraction distribution can be used as an indicator for tracking the vapour phase and bubble formation. Figure 5.4a shows that the vapour fraction at the outlet is relatively high, this happens due to the reduction in the brine pressure while it flows through the outlet orifice. The vapour void fraction at the wave crest is high, this happens because the produced vapour’s bubbles inside the nucleation region travel and raise to reach the wave crest, due to the jet flow.

It should be mentioned that the value of the vapour fraction is low (74 x 10⁻⁶); we do not have enough data to compare our predicted vapour fraction inside
the flashing chamber. These values may be contaminated by numerical error, thus further studies would be required to determine whether this is indeed the case.

A continuum of volume fraction above the free surface, and this is referred to as foam (Figure 5.4a). This agreed with what was observed experimentally by [176]. It may be a numerical effect, therefore, further studies are required to determine the effects of mesh sensitivity over the free surface zone.

Maximum value of vapour volume fraction in the evaporation zone is 74 x 10^{-6} (Figure 5.4d) and it is located in the bubble nucleation region. The small value obtained is due to gauge pressure (brine level) effects. On the other hand, the average value of vapour volume fraction at the free surface is 6.84 x 10^{-3}, which is higher than found in the bubble nucleation region. Thus phase change and mass transfer occurs predominantly at the free surface (Figure 5.5), similar to results observed in the experiments of Gilbert (1968) [62].

5.3.5. Total vapourisation mass flow

The total vapourisation mass flow rate can be calculated by computing the net produced vapour flux through the open boundary. From the field results along the open boundary, we obtain a mean product of y-component velocity times average fluid density of 0.433 kg/m^3 s. The flashing chamber length is 1.25 m, so that for the corresponding width of 7 m [256], the estimated vapour mass flow rate is 3.79 kg/sec (13.6 tonne/hour).

5.3.6. Non-equilibrium temperature losses

The average calculated value of the NETD is 3.8°C. This value may look slightly higher than typical values (see Table 2.3.) of operating plants.
However, it is still within the typical range of the categories of flashing chambers without a baffle.

There are several factors to consider. The simulated flashing chamber is the first flashing chamber of the MSF system: this means that it has the highest value of thermal losses. The simulated flashing chamber does not contain a baffle and there is not enough time for complete flashing to occur. Finally, the length of the flashing chamber is not sufficient to allow the system to reach a close enough value to thermal equilibrium conditions, which can then reduce the value of thermal losses.
5.4. Flashing Chamber Operating Under a Finite Flashing Condition

In this section, the computational model is applied to a flashing chamber operating under a finite flashing condition. In this case, the brine temperature reaches at the saturation temperature corresponding to the operation pressure of that flashing chamber, somewhere inside the flashing chamber. Thus, the outlet brine temperature is less than the saturation temperature corresponding to the vapour pressure of the flashing chamber. Furthermore, the non-equilibrium losses of the flashing chamber are negative.

For the current predicted flashing chamber the operation pressure is 1.4 bar. Note, the same boundary conditions and assumptions as the case in § 5.2 are applied.

The predicted results of the thermo-fluid performance (gauge pressure, flow patterns, thermal performance, vapour volume fraction and mass transfer) inside the flashing chamber are presented in Figures 5.7 through to 5.12.

5.4.1. Gauge pressure distribution

Gauge pressure distribution inside the flashing chamber is shown in Figures 5.7a, 5.7b, and 5.7d. The pressure increases with depth of brine below the free surface. Gauge pressure decreases along the horizontal, and this is responsible for driving the flow from the inlet orifice to the outlet orifice. The calculated pressure drop along horizontal traverses at $y = 0.3m$ inside the flashing chamber is 2150 Pa.
Figure 5.7a  Predicted gauge pressure field (Case: $v_x = 1.5$ m/s, $p_{op} = 1.4$ bar, $T_{in} = 383.15$ K)

Figure 5.7b  Gauge pressure distribution along vertical traverses at $x = 0.31$m, 0.62m, 0.93m (Case: $v_x = 1.5$ m/s, $p_{op} = 1.4$ bar, $T_{in} = 383.15$ K)
Figure 5.7c  Gauge pressure distribution along horizontal traverses at $y = 0.03\text{m}, 0.17\text{m},$ free surface line, $0.3\text{m}$ (Case: $v_x = 1.5 \text{ m/s}, p_{op} = 1.4 \text{ bar}, T_{in} = 383.15 \text{ K}$)

5.4.2. Particle paths and velocity distribution

Figure 5.8a shows flow fields: (i) velocity magnitude, (ii) vector map with fixed length vectors, (iii) vector map with vector length proportional to velocity magnitude, and (iv) particle paths (streamlines), coloured by the speed of the fluid mixture, throughout the flashing chamber. A detailed view of the flow pattern in the inlet region is also provided. A distinct brine recirculation flow zone appears in the solution. The dividing line between forward and reverse flow at the free surface lies just beyond $x = 0.6 \text{ m}$, as evident from Figure 5.8c.

The vertical traverses (Figure 5.8b) indicate a sharp discontinuity in velocity between the brine and vapour along the interface, with minima occurring at
the free surface. At $x=0.62$ m, the velocity magnitude at the free surface reaches 0 m/s, this due to a stagnation point located in that flow field.

Along the mid-orifice level (Figure 5.8c) the inlet flow decelerates due to jet spreading and subsequently accelerates to a maximum of 1.9 m/s at the brine exit.
Figure 5.8a  Predicted flow fields: (i) velocity magnitude, (ii) vector map with fixed length vectors, (iii) vector map with vector length proportional to velocity magnitude, and (iv) particle paths (streamlines) (Case: \( v_x = 1.5 \text{ m/s}, p_{op} = 1.4 \text{ bar}, T_{in} = 383.15 \text{ K} \) )
Figure 5.8b  Mixture speed along vertical traverses at \(x = 0.31\text{m}, 0.62\text{m}, 0.93\text{m}\) (Case: \(v_x = 1.5\text{ m/s}, p_{op} = 1.4\text{ bar}, T_{in} = 383.15\text{ K}\))

Figure 5.8c  Mixture speed along horizontal traverses at \(y = 0.03\text{m}, 0.17\text{m}, \text{free surface line}, 0.3\text{m}\) (Case: \(v_x = 1.5\text{ m/s}, p_{op} = 1.4\text{ bar}, T_{in} = 383.15\text{ K}\))
5.4.3. Thermal performance

The flashing process across the flashing chamber causes a reduction of brine temperature, as can be seen in the overall temperature field (Figures 5.9a), and also in the vertical and horizontal temperature traverses presented in Figures 5.9b, and 5.9c.

The decrease in temperature takes place in both the horizontal ($x$-) and vertical ($y$-) directions, but at different rates. The differences in range and scale between Figures 5.9b and 5.9c should be noted. The temperature gradient in vertical ($y$-) directions is large compared to the ($x$-) horizontal.

**Figure 5.9a** Predicted temperature field (Case: $v_x = 1.5$ m/s, $p_{op} = 1.4$ bar, $T_{in} = 383.15$ K)
**Figure 5.9b**  Temperature distribution along vertical traverses at $x = 0.31$ m, 0.62 m, 0.93 m (Case: $v_x = 1.5$ m/s, $p_{op} = 1.4$ bar, $T_{in} = 383.15$ K)

**Figure 5.9c**  Temperature distribution along horizontal traverses at $y = 0.03$ m, 0.17 m, 0.3 m and on free surface (vapour region), and along free surface ($y$ varies) (Case: $v_x = 1.5$ m/s, $p_{op} = 1.4$ bar, $T_{in} = 383.15$ K)
5.4.4. Vapour volume fraction and mass transfer rate

*Figure 5.10a* shows the vapour volume fraction ($\alpha_v$) in the flashing chamber. For clarity two field images are provided. The upper image (a) indicates the full range of vapour volume fraction scale from 0 to 1, and the lower set of images (b) focuses on a restricted range, between 0 and 0.00002. The peak values of vapour volume fraction occur at the free surface and in the bubble nucleation region. Vertically (*Figure 5.10b*) the values of vapour volume fraction increase. While, horizontally (*Figure 5.10c*) the values of vapour volume fraction decrease. This happens because of the reduction in brine temperature and increased saturation temperature.

*Figure 5.10d* shows the distribution of vapour volume fraction along $y = 0.03$ m, which passes through the bubble formation region. As shown in *Figure 5.10d* there is vapour formed, inside the nucleation bubble formation, where the brine temperature is higher than the saturation temperature. *Figure 5.10e* shows the average vapour volume fraction along the saturation temperature line where it is considered as the boundary of the bubble nucleation region. The saturation line is a contour of the temperature, approximated as $T_{\text{sat}}$ at the operating pressure. But the local pressure is not the same as the operating pressure. However, since the gauge pressure is small (3 kPa), the error in $T_{\text{sat}}$ is small (less than 1°C). The value of $T_{\text{sat}}$ is 382.47K.

*Figure 5.11* shows the mass transfer rate in the bubbles formation region and at the free surface layer inside the flashing chamber.
Figure 5.10a Vapour volume fraction ($\alpha_v$) fields with two different ranges of scale; (i) upper field range $[0 - 1]$, and (ii) lower field range $[0 - 2.2 \times 10^{-5}]$

(Case: $v_x = 1.5 \text{ m/s}$, $p_{op} = 1.4 \text{ bar}$, $T_{in} = 383.15 \text{ K}$)
Figure 5.10b  Vapour volume fraction distribution along vertical traverses at \( x = 0.31 \text{m}, 0.62 \text{m}, \text{and} 0.93 \text{m} \) (Case: \( v_\text{x} = 1.5 \text{ m/s}, p_\text{op} = 1.4 \text{ bar}, T_\text{in} = 383.15 \text{ K} \))

Figure 5.10c  Vapour volume fraction distribution along horizontal traverses at \( y = 0.03 \text{m}, 0.17 \text{m} \text{ and} 0.3 \text{m} \) (vapour region), and along free surface (\( y \) varies) (Case: \( v_\text{x} = 1.5 \text{ m/s}, p_\text{op} = 1.4 \text{ bar}, T_\text{in} = 383.15 \text{ K} \))
Figure 5.10d  Vapour volume fraction distribution along horizontal traverses \( y = 0.03 \text{m} \) (Case: \( v_x = 1.5 \text{ m/s}, \ p_{op} = 1.4 \text{ bar}, \ T_{in} = 383.15 \text{ K} \))

Figure 5.10e  Average vapour volume fraction along the saturation temperature line (\( y \) varies) (Case: \( v_x = 1.5 \text{ m/s}, \ p_{op} = 1.4 \text{ bar}, \ T_{in} = 383.15 \text{ K} \))
Figure 5.11 Mass transfer rate field with three different ranges of scale; (a) $[-130, +16]$ kg/m$^3$sec, (b) $[-28, +0.08]$ kg/m$^3$sec, and (c) $[-0.0034, +0.2]$ kg/m$^3$sec (Case: $v_x = 1.5$ m/s, $p_{op} = 1.023$ bar, $T_{in} = 383.15$ K) (Case: $v_x = 1.5$ m/s, $p_{op} = 1.4$ bar, $T_{in} = 383.15$ K)
5.4.5. Non-equilibrium losses

Figure 5.12a shows non-equilibrium losses (NETD) over the flashing chamber outlet orifice. The NETD decreases almost linearly from the top to the bottom of the orifice (with depth of the brine), with ranging from \(-4.01\) to \(-3.48\)°C. Figure 5.12b shows the relationship between the NETD and flashing down. The NETD is proportional to flashing down. The range of the flashing down is from 4.16 to 4.69°C. Figure 5.12c shows the relationship between the NETD and flashing efficiency. The flashing efficiency increases when the NETD decreases. The thermal flashing efficiency ranges from to 0.171 to 0.192.

Figure 5.12a  NETD distribution over outlet orifice (Case: \(v_x = 1.5\) m/s, \(p_{op} = 1.4\) bar, \(T_{in} = 383.15\) K)
Figure 5.12b  NETD distribution over flashing down (Case: $v_x = 1.5 \text{ m/s}, p_{op} = 1.4 \text{ bar}, T_{in} = 383.15 \text{ K}$)

Figure 5.12c  NETD distribution over flashing efficiency (Case: $v_x = 1.5 \text{ m/s}, p_{op} = 1.4 \text{ bar}, T_{in} = 383.15 \text{ K}$)
5.5. Discussion

5.5.1 Gauge pressure distribution

Gauge pressure in the finite flashing case has similar effect as the in the infinite flashing case; it resists the flashing process by increasing vapour vapour pressure with depth, which affects the saturation temperature. Thus, both mass transfer and phase change decrease due to an increase in pressure with depth.

5.5.2. Particle paths and the velocity distribution

Maximum speed is found to be 1.9 m/s at the end of the flashing chamber. This happens in order to keep conservation of energy concept valid, since pressure in this point has reached the minimum value.

The recirculation zone promotes bubble transportation from the bubble formation region to the free surface. Other information about the relationship between fluid flow and its thermal behaviour can be predicted using such simulations, i.e. rate at which turbulent kinetic energy converts into thermal internal energy. This can provide information about vertical heat flow, which is required for evaporating liquid close to the free-surface.

The flashing process inside the flashing chamber is a very sensitive process. Based on the relationship between fluid pattern and flashing process, the inlet velocity profile is one of the most important fluid dynamic parameter effects on the flashing process. Thus, the inlet velocity profile should be carefully considered for the inter-stages and the last flashing chamber. Here, we are dealing with the first flashing chamber where the brine flows into it through pipes. To test the effect of inlet velocity profiles, for the same mass flow rate we run three different, unidirectional flow cases with different inlet
profiles. We tested three different inlet velocity profiles (see Appendix H), we found that the uniform inlet velocity profile leads to closest average outlet temperature compared to the only data we have available from the plant [256].

5.5.3. Thermal performance

This reduction of brine temperature is attributed to the vapourising enthalpy that is used to change phase. As noticed in Figures 5.10 and 5.11, the reduction in temperature affects the mass transfer rate and the vapour volume fraction distribution.

The temperature gradient is affected due to the turbulent mixing process between the upper and lower brine layers. The heat transfers in the vertical direction rather than in the horizontal direction because the horizontal temperature gradient is smaller compared to the vertical gradient. Thus, the rate of temperature decrease in y-direction is larger than in x-direction.

The phase change reduces the brine temperature to the point where it reaches the saturation temperature (over the saturation line in Figure 5.10) This leads to no more change of phase taking place inside the evaporation zone after the saturation line, and thus flashing is finished.

5.5.4. Vapour volume fraction and mass transfer

The phase keeps changing in the evaporation zone as long as the phase change driving force is positive. However, the brine temperature inside the flashing chamber reached the saturation temperature, corresponding to the flashing chamber pressure, at $x = 0.54m$ and kept reducing. Thus (Figure 5.10a) the vapour phase change takes place in a certain region in the
evaporation zone. The major phase change takes place in the bubble nucleation region (*Figure 5.10d & 5.11*).

The maximum value of vapour volume fraction inside the bubble nucleation region is $6 \times 10^{-6}$. The small value obtained may be due to gauge pressure effects. On the other hand, the average value of vapour volume fraction at the free surface is 0.00684, which is higher than found in the bubble nucleation region. Thus, phase change and mass transfer occurs predominantly at the free surface (*Figure 5.11*). The predicted mass flow rate from the model is 12.62 tonne per hr (see §5.3.5 for method used).

5.5.5. Non-equilibrium temperature losses

Ideally, the value of the NETD should be close to the zero as it was mentioned earlier. For the predicted case, the average calculated value of the NETD is $-3.74^\circ C$. This is considered as a thermal loss since more energy is needed to apply in the next flashing chamber.
5.5.6. Flashing regions of the flashing chamber operating under a finite flashing condition

The predicted flashing mechanism in the simulation agrees with the mechanisms stated in the review section. The bubbles are formed at the entrance of the flashing chamber (see Figure 4.15) and bubble formation reduces along its length. At the surface of the liquid, there is also a phase change and mass transfer. Thus, the two main mechanisms of phase change by the flashing process are captured in this simulation.

Figure 5.13  (a) Flashing regions of the flashing chamber (adapted from Lior [47]), (b) Flashing regions in the flashing chamber as predicted by computational model
5.6. Summary

A flashing process can be classified into two types: infinite flashing (incomplete), and finite flashing (completed). Under infinite flashing the phase change continues occurring until the end of the flashing chamber, while for finite flashing the phase change stops once the flashing driving force, the superheat temperature, reaches zero.

Each of the two flashing processes may occur depending upon the flashing chamber operating conditions. Thus, operating parameter variables can affect the flashing chamber performance differently, based on the flashing type.

As shown in the predicted results, flashing can be defined as infinite flashing when the exit brine temperature is higher than the saturation temperature of the brine, (corresponding to the flashing chamber pressure). In this case, the value of the $NETD$ is positive.

When the exit brine temperature is lower than the saturation temperature of the brine (corresponding to the flashing chamber pressure), flashing can be defined as finite flashing. In this case, the value of the $NETD$ is negative.

The ideal performance of the flashing chamber is attained when the flashing process completes just at the exit of the flashing chamber. Here, the $NETD$ is equal to zero. Thus infinite flashing needs either more time or a longer flashing chamber length or maybe a higher flow rate to attain the ideal case, the $NETD = 0$, of flashing. Conversely finite flashing requires either a short flashing chamber length or higher flow rate to attain the ideal case. As a result, both non ideal flashing types are considered as thermal losses to the system. For the infinite case, there is still energy existed, which needs to be utilised inside the flashing chamber and for the finite case, more energy needs to be expended for the next flashing chamber to recover waste.
generated energy during a no phase change period in the previous flashing chamber.

Generally, a non-equilibrium loss (NETD) is used as an indicator of the flashing chamber performance. However, checking the NETD is not enough to judge that the flashing chamber performs sufficiently. The NETD is merely an indicator to establish how far the flashing chamber operation conditions of the system deviate from the system from thermal equilibrium conditions.

Many factors affect the NETD of the flashing chamber, e.g. operating pressure (as is presented in this Chapter), inlet flow parameters, feed brine mass flow rate, brine temperature, and flashing residence time. In Chapters 6, 7, 8 and 9 a parametric study of these variables and the effect on the flashing chamber thermo-fluid process are presented under infinite and finite flashing conditions.
6. EFFECTS OF INLET FLOW RATE – INFINITE FLASHING

6.1. Introduction

The inlet brine flow to the first flashing chamber is provided through parallel pipes [1, 63]. The brine velocity inside the pipes should not be low to avoid sludge deposition, scale accumulation, and low heat transfer coefficients in the pipe side [1, 63, 123]. Also, brine velocity should not be high to avoid excessive brine pumping power [1, 63, 123]. Thus, in MSF plant the range of the average brine speed [123] entry the flashing chamber is between 1 to 2 m/s.

The objective of this Chapter is to study the effect of different inlet brine velocities ($\bar{v}_{xin}$ = 1, 1.5, and 2 m/s), corresponding to varieties in the inlet mass flow rate, on the thermo-fluid performance of a flashing chamber, without a baffle, operating under an infinite flashing condition ($p_{op} = 1.023$ bar).

This Chapter is divided into two sections; the first part presents the predicted thermo-fluid results (particle paths of flashing flow, gauge pressure, temperature distribution, vapour volume fraction, and mass transfer rate) due to varying the inlet brine velocity. In the absence of information on the details of the geometric or velocity profile at the inlet, a uniform, horizontal velocities distribution is applied.

The second part is a comparison between the different inlet velocity cases in terms of both thermo-fluid behaviour and flashing chamber performance.
factors e.g. non-equilibrium losses (NETD), flashing down, flashing efficiency, flashing rate, and maximum bubble nucleation.

The computational model is applied to a flashing chamber, without a baffle and operating under an infinite flashing condition. The same computational domain configuration is used as in Figure 4.2. The defined value of the inlet velocity of the boundary conditions has been stated based on the study case. Three cases are investigated, with mean inlet brine velocities of \( v_{xin} \) = 1, 1.5, and 2 m/s respectively. Otherwise the same boundary conditions, operating conditions and assumptions as used in § 5.2, are used for running the simulations.

In Chapter 5 at § 5.2, the predicted results of the thermo-fluid performance of \( v_{xin} \) = 1.5 m/s were presented. The remaining two cases are studied in the in the following § 6.2. and 6.3.

6.2. Effects of \( v_{xin} \) = 1 m/s on Thermo-Fluid

In this section, the predicted results of the thermo-fluid performance (gauge pressure, flow patterns, thermal performance, vapour volume fraction and mass transfer) inside the flashing chamber are summarised in Figures 6.1 – 6.5.

6.2.1. Gauge pressure distribution

The gauge pressure inside the flashing chamber increases with depth of the brine below the free surface (see Figures 6.1) and it decreases along the x-direction. The calculated pressure drop along a horizontal traverse at mid-orifice level \((y = 0.03\text{m})\) is 2140 Pa.
Figure 6.1a  Predicted gauge pressure field (Case: \(v_x = 1\) m/s, \(p_{op} = 1.023\) bar, \(T_{in} = 383.15\) K)

Figure 6.1b  Gauge pressure distribution along vertical traverses at \(x = 0.31\) m, 0.62m, 0.93m (Case: \(v_x = 1\) m/s, \(p_{op} = 1.023\) bar, \(T_{in} = 383.15\) K)
Figure 6.1c  Gauge pressure distribution along horizontal traverses at $y = 0.3\text{m}, 0.17\text{m},$ free surface line, $0.3\text{m}$ (Case: $v_x = 1 \text{ m/s}, p_{op} = 1.023 \text{ bar}, T_{in} = 383.15 \text{ K}$)

6.2.2. Particle paths and velocity distribution

Figure 6.2a shows particle paths (streamlines) with magnitude of mixture velocity inside the flashing chamber, and it shows the recirculation zone which is created below the free-surface with its size and length.

As noted in Figures 6.2b & 6.2c, in the $y$-direction, the speed value decreases until it reaches the minimum value at the free-surface and subsequently, it increases again in the vapour layer. In the $x$-direction, the maximum speed is found to be $1.89 \text{ m/s}$ at the end of the flashing chamber.
Figure 6.2a  Predicted particle paths (streamlines) (Case: \(v_x = 1 \text{ m/s}, p_{op} = 1.023 \text{ bar}, T_{in} = 383.15 \text{ K}\))

Figure 6.2b  Mixture speed along vertical traverses at \(x = 0.31\text{m}, 0.62\text{m}, 0.93\text{m}\) (Case: \(v_x = 1 \text{ m/s}, p_{op} = 1.023 \text{ bar}, T_{in} = 383.15 \text{ K}\))
6.2.3. Thermal performance

The reduction of brine temperature inside the flashing chamber can be seen in Figures 6.3a, 6.3b, & 6.3c. The rate of the brine temperature decrease is not the same in the $x$- or $y$-direction.

*Figure 6.2c* Mixture speed along horizontal traverses at $y = 0.03$ m, 0.17 m, free surface line, 0.3 m (Case: $v_x = 1$ m/s, $p_{op} = 1.023$ bar, $T_{in} = 383.15$ K)

*Figure 6.3a* Predicted temperature field (Case: $v_x = 1$ m/s, $p_{op} = 1.023$ bar, $T_{in} = 383.15$ K)
Figure 6.3b  Temperature distribution along vertical traverses at $x = 0.31\text{m}$, $0.62\text{m}$, $0.93\text{m}$ (Case: $v_x = 1 \text{ m/s}$, $p_{op} = 1.023 \text{ bar}$, $T_{in} = 383.15 \text{ K}$)

Figure 6.3c  Temperature distribution along horizontal traverses at $y = 0.3\text{m}$, $0.17\text{m}$, free surface line, $0.3\text{m}$ (Case: $v_x = 1 \text{ m/s}$, $p_{op} = 1.023 \text{ bar}$, $T_{in} = 383.15 \text{ K}$)
6.2.4. Vapour volume fraction and mass transfer rate

Figure 6.4a shows the vapour volume fraction in the flashing chamber. The upper image (a) indicates the full range of vapour volume fraction scale from 0 to 1, and shows that above the free surface there is a mixture of vapour and liquid phases. The lower set of images (b) focuses on a restricted range, between 0 and about 0.0001.

The average vapour volume fraction along the free surface \( (\alpha_v) \) is 0.0063, while in the bubble nucleation region does not exceed \( 7.2 \times 10^{-4} \). Three different areas are appeared and numbered in the bubble nucleation zone. The maximum bubble formation takes place in area (1), in the inlet jet just below the top edge of the inlet orifice, then the bubble formation decreases from area (2) into area (3).

In the \( y \)-direction, (Figure 6.4b) the values of vapour volume fraction starts increasing above the free surface, while, in the \( x \)-direction (Figure 6.4c) the values of vapour volume fraction decrease along the flashing chamber length.

The distribution of vapour volume fraction along \( y = 0.03 \) m, which passes through the bubble formation region can be seen in Figure 6.4d.

Figure 6.5 shows the mass transfer rate in the bubble nucleation region and at the free surface layer inside the flashing chamber. The vapourisation process occurs when the mass transfer value is positive, while the condensation process occurs when the value is negative.
Figure 6.4a  Vapour volume fraction ($\alpha_v$) fields with two different ranges of scale; (i) upper field range [0 – 1], and (ii) lower field range [0 – 9x10^{-5}] (Case: $u_x = 1$ m/s, $p_{op} = 1.023$ bar, $T_{in} = 383.15$ K)
Figure 6.4b  Vapour volume fraction distribution along vertical traverses at $x$ = 0.31m, 0.62m, and 0.93m (Case: $v_x = 1$ m/s, $p_{op} = 1.023$ bar, $T_{in} = 383.15$ K)

Figure 6.4c  Vapour volume fraction distribution along horizontal traverses at $y$ = 0.03m, 0.17m and 0.3m (vapour region), and along free surface ($y$ varies) (Case: $v_x = 1$ m/s, $p_{op} = 1.023$ bar, $T_{in} = 383.15$ K)
Figure 6.4d  Vapour volume fraction distribution along horizontal travers $y$ =0.03m (Case: $v_x = 1$ m/s, $p_{op} = 1.023$ bar, $T_{in} = 383.15$ K)
Figure 6.5  Mass transfer rate field with three different ranges of scale; (a) $[-190, +34]$ kg/m$^3$sec, (b) $[-26, +8]$ kg/m$^3$sec, and (c) $[-0.0014, +2.5]$ kg/m$^3$sec
(Case: $v_x = 1$ m/s, $p_{op} = 1.023$ bar, $T_{in} = 383.15$ K)
6.3. Effects of $\bar{v}_{xin} = 2\text{m/s}$ on Thermo-Fluid

In this section, the predicted results of the thermo-fluid performance (gauge pressure, flow patterns, thermal performance, vapour volume fraction and mass transfer) inside the flashing chamber are presented in Figures 6.6 through to 6.11.

6.3.1. Gauge pressure distribution

*Figures 6.6a, 6.6b & 6.6c* show the gauge pressure distribution inside the flashing chamber. The pressure increases with the depth of brine below the free surface, while, gauge pressure decreases along the $x$-direction. The calculated pressure drop along the flashing chamber is 2070 Pa.

![Figure 6.6a](image)

*Figure 6.6a* Predicted gauge pressure field (Case: $v_x = 2\text{ m/s}$, $p_{op} = 1.023$ bar, $T_{in} = 383.15$ K)
Figure 6.6b  Gauge pressure distribution along vertical traverses at \( x = 0.31 \text{m}, \ 0.62 \text{m}, \ 0.93 \text{m} \) (Case: \( v_x = 2 \text{ m/s}, p_{op} = 1.023 \text{ bar}, T_{in} = 383.15 \text{ K} \))

Figure 6.6c  Gauge pressure distribution along horizontal traverses at \( y = 0.03 \text{m}, \ 0.17 \text{m}, \ \text{free surface line}, \ 0.3 \text{m} \) (Case: \( v_x = 2 \text{ m/s}, p_{op} = 1.023 \text{ bar}, T_{in} = 383.15 \text{ K} \))
6.3.2. Particle paths and velocity distribution

The streamlines with magnitude of mixture velocity inside the flashing chamber is shown in Figures 6.7a, 6.7b & 6.7c.

Figure 6.7a shows the recirculation zone’s location, size and length. In the y-direction, the speed value decreases until it reaches the minimum value, at the free-surface and subsequently, increases again. In the x-direction, speed decreases along the flashing chamber length, however at the end of the flashing chamber exit, it increases to 1.9 m/s. The maximum speed is found to be 2 m/s at the inlet of the flashing chamber.

\[ v_x = 2 \text{ m/s} \]
\[ p_{op} = 1.023 \text{ bar} \]
\[ T_{in} = 383.15 \text{ K} \]

Figure 6.7a  Predicted particle paths (streamlines) (Case: \( v_x = 2 \text{ m/s}, p_{op} = 1.023 \text{ bar}, T_{in} = 383.15 \text{ K} \))
Figure 6.7b  Mixture speed along vertical traverses at $x = 0.31\text{m}, 0.62\text{m}, 0.93\text{m}$

(Case: $v_x = 2\text{ m/s}$, $p_{op} = 1.023\text{ bar}$, $T_{in} = 383.15\text{ K}$)

Figure 6.7c  Mixture speed along horizontal traverses at $y = 0.03\text{m}, 0.17\text{m}$, free surface line, 0.3m

(Case: $v_x = 2\text{ m/s}$, $p_{op} = 1.023\text{ bar}$, $T_{in} = 383.15\text{ K}$)
6.3.3. Thermal performance

*Figures 6.8a, 6.8b, & 6.8c* show the thermal performance inside the flashing chamber. The brine temperature decreases in the $x$ and $y$-directions. It is observed that the rate of decrease is not the same in both directions.

*Figure 6.8a* Predicted temperature field (Case: $v_x = 2 \text{ m/s}$, $p_{op} = 1.023 \text{ bar}$, $T_{in} = 383.15 \text{ K}$)

$v_x = 2 \text{ m/s}$

$p_{op} = 1.023 \text{ bar}$

$T_{in} = 383.15 \text{ K}$
Figure 6.8b  Temperature distribution along vertical traverses at $x = 0.31$ m, 0.62 m, 0.93 m (Case: $v_x = 2$ m/s, $p_{op} = 1.023$ bar, $T_{in} = 383.15$ K)

Figure 6.8c  Temperature distribution along horizontal traverses at $y = 0.03$ m, 0.17 m, free surface line, 0.3 m (Case: $v_x = 2$ m/s, $p_{op} = 1.023$ bar, $T_{in} = 383.15$ K)
6.3.4. Vapour volume fraction and mass transfer Rate

The vapour volume fraction and the mass transfer rate in the flashing chamber are shown in Figures 6.9 & 6.10, respectively.

For Figure 6.9a, the upper image (a) indicates the full range of vapour volume fraction scale from 0 to 1, while the lower set of images (b) focuses on a restricted range, between 0 and about 0.0001. The peak values of vapour volume fraction occur at the free surface (average \( \alpha_v = 0.019 \)) and in the bubble nucleation region (maximum \( \alpha_v = 7.4 \times 10^{-4} \)). In the bubble nucleation region three different areas are numbered; the maximum bubble formation takes place in area (1), in the inlet jet just below the top edge of the inlet orifice.

Along vertical traverses above the free surface (Figure 6.9b), the vapour volume fraction increase monotonically. Horizontally, (Figure 6.9c) the values of vapour volume fraction decrease. Vapour volume fraction along \( y = 0.03 \) m can be seen in Figure 6.9d.
Figure 6.9a  Vapour volume fraction \( (\alpha_v) \) fields with two different ranges of scale; (i) upper field range \([0 – 1]\), and (ii) lower field range \([0 – 9 \times 10^{-5}]\) (Case: \( \nu_x = 2 \text{ m/s}, p_{op} = 1.023 \text{ bar}, T_{in} = 383.15 \text{ K} \))
Figure 6.9b  Vapour volume fraction distribution along vertical traverses at 
  \( x = 0.31 \text{m}, 0.62 \text{m}, \text{and } 0.93 \text{m} \) (Case: \( v_x = 2 \text{ m/s}, \ p_{op} = 1.023 \text{ bar}, T_{in} = 383.15 \text{ K} \))

Figure 6.9c  Vapour volume fraction distribution along horizontal traverses 
  at \( y = 0.03 \text{m}, 0.17 \text{m and } 0.3 \text{m} \) (vapour region), and along free surface (\( y \) 
  varies) (Case: \( v_x = 2 \text{ m/s}, \ p_{op} = 1.023 \text{ bar}, T_{in} = 383.15 \text{ K} \))
Figure 6.9d  Vapour volume fraction distribution along horizontal traver $y = 0.03m$
(Case: $v_r = 2 \text{ m/s}, p_{op} = 1.023 \text{ bar}, T_{in} = 383.15 \text{ K}$)
Figure 6.10  Mass transfer rate field with three different ranges of scale; (a) \([-170, +24]\) kg/m\(^3\)sec, (b) \([-26, +8]\) kg/m\(^3\)sec, and (c) \([-0.0015, +2.5]\) kg/m\(^3\)sec
(Case: \(v_x = 2\) m/s, \(p_{op} = 1.023\) bar, \(T_{in} = 383.15\) K)
6.4. Discussion of Effects of Inlet Flow Rate under Infinite Flashing

The change of the inlet brine velocity (or inlet flow rate) affects the thermo-fluid performance inside the flashing chamber significantly. These effects are presented in the following sections

6.4.1 Gauge pressure distribution

For all cases, the gauge pressure increases linearly with depth below the free surface. A discontinuity in pressure gradient is noticed between the brine and the vapour layers.

It is found that increasing inlet velocity affected the magnitude of the pressure distribution. As it can be noticed in Figures 6.1b, 5.1b, & 6.6b the magnitude of the gauge pressure along a vertical traverse at x = 0.93m is increased by increasing the inlet flow rate. This occurred mainly due to the local increase in brine level compared with x = 0.31m and x = 0.62m. At the same time, the total pressure at x = 0.93m is increased due to the increasing in the vapour phase by increasing the flow rate. This happened due to the increasing in the flashing rate, the bubble nucleation frequency, and the length of the bubble nucleation region (as it is evident in Figures 6.13, 6.14, & 6.15) which leads to an increase of mass transfer and phase change and generated more vapour bubbles inside the evaporation zone.

The brine level does not change for all cases. The free surface shape is uniform for all cases as it can be noticed in Figure 6.15a. This happened due to the brine level and the flow rate; the brine level is high enough and the flow rate is low enough.
Here we model a steady-state, two-dimensional representation of the evaporation zone, with no baffles and with a uniform inlet velocity profile. Because the calculation is mathematically constrained to be time-independent, the predicted interface shape represents an (artificial) equilibrium solution where no unsteadiness is allowed. Without performing a full transient (‘dynamic’) solution, and comparing results with the steady-state case, we cannot know how much the dynamic effects influence the performance.

6.4.2. Particle paths and velocity distribution

The recirculation zone promotes bubble transportation from the bubble formation region to the free surface. The streamlines of the three investigated cases can be shown in Figure 6.11. The recirculation zone is created at the first half of the flashing chamber. The size, length and location of the recirculation zone changes based on the inlet flow rate. The flow rate clearly affects the size and length of the recirculation zone; e.g. for the case of $v_{\text{in}} = 1 \text{ m/s}$ the recirculation zone has the smallest size. The higher the brine velocity the higher the recirculation zone size and length, and the farther recirculation zone centre from the inlet flashing chamber. The velocity magnitude in the recirculation is small, in comparison to the inlet velocity of the flashing chamber. This agrees with results provided by [209] during the investigation of flow patterns in flashing chamber by using PIV measurements. Inlet brine velocity (inlet flow rate) clearly has a significant effect on flow patterns inside the flashing chamber.

The mixture velocity at any point along the streamlines in the flow has a single value and it cannot take more than one direction at the same time (streamlines cannot cross). The point where the velocity magnitude is zero is defined as a stagnation point. It is noticed that the stagnation point over the
free surface is located vertically from the end edge of the bubble nucleation zone.

In the vapour layer, it is found that the magnitude of the mixture speed decreases by increasing inlet brine velocity as it is evident in Figures 6.2b, 5.2b & 6.7b.
Figure 6.11  Effect of inlet brine velocity on predicted particle paths – infinite flashing.
6.4.3. Thermal performance

Inlet brine velocity (inlet flow rate) affects the thermal performance inside the flashing chamber. The reduction rate of brine temperature in both \( x \) and \( y \) directions, is affected by the change of the inlet flow rate. Inlet brine velocity (inlet flow rate) affects the thermal inertia, a measure of thermal mass and velocity of the thermal wave (see Figure 6.12). A higher inlet brine velocity leads to a higher thermal wave. This can be observed in Figures 6.3b, 5.3b, & 6.8b, for the traverse \( x = 0.62 \) m.

As a result of the changes in thermal performance, mass transfer rate and vapour volume fraction distribution are affected. Bubble formation is increased by increasing the flow rate (see Figure 6.15). The higher the inlet brine velocity (or the higher the flow rate), the higher flashing rate (or phase change rate) that occurs. This happens due to the lowered pressure in the bubble formation, and the increasing of the mixing and turbulence that is created in the evaporation zone by increasing the inlet flow rate. Figure 6.13 shows the effect of inlet brine velocity on the flashing rate along the flashing chamber length. It is clear that the flashing rate is increasing by increasing the inlet brine velocity (inlet flow rate). As it can be noticed in Figure 6.13 the reduction point of the flashing rate is delayed by increasing the flow rate, this is due the higher thermal wave which is created. In other words, the length of the bubble nucleation region is increased but the maximum magnitude of the flashing rate is equal for the three cases (the maximum flashing rate is 51 kW/m\(^2\)).

As a result of changes in the flashing rate, the brine temperature at the outlet orifice and the \( NETD \) changed. This will be further discussed in § 6.4.5.

In fact, these effects on the thermal performance can be predicted by checking the flow pattern and the inlet flow rate. Since increasing mass flow
rates leads to higher turbulent mixing but less in residence time for bubble nucleation and growing. So, increasing the flow rate improves the mass transfer and phase change rate along the flashing chamber; but it may not give the bubbles a reasonable time to nucleate and grow. Therefore, the flashing chamber system can be farther away from thermal equilibrium conditions corresponding to the flashing chamber operating conditions.
Figure 6.12  Effect of inlet brine velocity on predicted temperature field – infinite flashing
6.4.4. Vapour volume fraction and mass transfer rate

Inlet brine velocity (or inlet flow rate) affects the vapour volume fraction distribution inside the evaporation zone in general and the bubble nucleation region in particular. This happens because of its significant effect on many variables which are responsible for driving the force for changing the phase from liquid to vapour such as: bubbles pressure difference, superheat temperature, and residence time.

It is found that a higher inlet brine velocity (higher flow rate) leads to a higher nucleation frequency and phase change rate along the flashing chamber length as it is evident in Figure 6.14. The values of the maximum bubble nucleation frequency in Figure 6.14 are extracted based on Eq 3.5. It is clear that the maximum bubble nucleation frequency decreases along the flashing chamber length. The maximum bubble nucleation frequency depends on \((T_b - T_{sat})\), as mentioned in §3.3.4. The brine temperature

Figure 6.13 Effect of inlet brine velocity on flashing rate along horizontal traverses at \(y = 0.03\text{m}\) – infinite flashing.
decreases along the flashing chamber length due to the change in the brine phase into vapour. Thus, the maximum bubble nucleation frequency along the flashing chamber length is expected to be decreased in all cases. The maximum magnitude is the same for the three cases (9.269 x 10^7) but the maximum length of the peak of the bubble nucleation frequency increases proportionally with the inlet flow rate. Table 6.1 summarises the effects of the inlet flow rate on the flashing process.

\[ \text{Table 6.1: Summary of the inlet brine flow rate and the maximum length for both the peak of flashing rate and bubble nucleation frequency} \]

<table>
<thead>
<tr>
<th>( \bar{v}_{xin} ) [m/s]</th>
<th>Maximum length of the peak flashing [m]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.16</td>
</tr>
<tr>
<td>1.5</td>
<td>0.31</td>
</tr>
<tr>
<td>2</td>
<td>0.39</td>
</tr>
</tbody>
</table>

\[ \text{Figure 6.14: Effect of inlet brine velocity on maximum bubble nucleation frequency along horizontal traverses at } y = 0.03 \text{ m – infinite flashing} \]

\[ \text{Figure 6.15: Vapour volume fraction distribution and mass transfer rate inside the flashing chamber for different inlet brine velocities (inlet flow} \]
rate). At $v_{xin} = 2$ m/s, the maximum value of vapour volume fraction and bubble formation region length are found.

Higher flow rate leads to higher turbulent flow in the evaporation zone, as mentioned previously. Turbulent flow promotes the rate of conversion of mechanical energy to heat, within feed brine inside the flashing chamber and this is the reason behind the promotion of the phase changing rate due to turbulent flows.
**Figure 6.15a** Effect of inlet brine velocity on predicted vapour volume fraction field – infinite flashing

\[ v_x = 1 \text{ m/s} \]
\[ p_{op} = 1.023 \text{ bar} \]
\[ T_{in} = 383.15 \text{ K} \]

\[ v_x = 1.5 \text{ m/s} \]
\[ p_{op} = 1.023 \text{ bar} \]
\[ T_{in} = 383.15 \text{ K} \]

\[ v_x = 2 \text{ m/s} \]
\[ p_{op} = 1.023 \text{ bar} \]
\[ T_{in} = 383.15 \text{ K} \]
Figure 6.15b  Effect of inlet brine velocity on predicted mass transfer – infinite flashing
Figure 6.15c Effect of inlet brine velocity on predicted mass transfer – infinite flashing
Mixing improves the phase change rate by increasing the vapour pressure \((p_v)\) inside the bubbles to a summation of both the vapour pressure and the non-condensable gas pressure \((p_v + p_{\text{non-condensable gas}})\). Thus the energy required for changing the phase from liquid to vapour is reduced (as discussed in §3.3.3.1). Thus, the flashing rate is improved.

Increasing the brine speed creates more turbulent mixing flow inside the flashing chamber, which improves the rate of both mass transfer and phase change from liquid to vapour, but it reduces the residence time for bubble formation. As a result, the flashing process cannot be able to be completed inside the flashing chamber, and it cannot be able to reach the thermal equilibrium condition. Thus, the thermal losses \((NETD)\) maybe increase inside the flashing chamber by increasing the inlet flow rate.

6.4.5. Non-equilibrium losses and thermal flashing efficiency

The flashing process inside the flashing chamber is affected by the inlet brine velocity (or inlet flow rate). Figure 5.16a shows the non-equilibrium losses \((NETD)\) as a function of flashing chamber outlet orifice for different inlet brine velocity numbers. The \(NETD\) values range of \(\overline{v}_{xin} = 1\text{m/s}, \overline{v}_{xin} = 1.5\text{m/s}, \overline{v}_{xin} = 2\text{m/s}\) from 4.95 to 5.69 °C, 4.8 to 5.59 °C, and 5.093 to 4.7 °C respectively.

It is found that \(\overline{v}_{xin} = 1.5 \text{ m/s}\) has the lowest \(NETD\). However, it is found that \(\overline{v}_{xin} = 2 \text{ m/s}\) has the highest \(NETD\). This may be referred to as not having enough time (residence time) for using the whole energy for changing the phase from liquid to vapour. The \(NETD\) is an indicator of an incomplete flashing process inside the flashing chamber. It represents residual superheating, corresponding to the saturation temperature of the brine, of the brine exit of the flashing chamber.
The effect of inlet brine velocity on the relationship between the $NETD$ and flashing down can be seen in Figure 6.16b. When flashing down increases, the $NETD$ decreases in all cases. The range of flashing down of $v_{xin} = 1\text{ m/s}$, $v_{xin} = 1.5\text{ m/s}$, $v_{xin} = 2\text{ m/s}$ is from 4.11 to 4.77°C, 4.17 to 4.88°C, and 4 to 4.65°C respectively.

The flashing thermal efficiency increases when the $NETD$ decreases as it shown in Figure 6.16c. Thermal flashing efficiency ranges of $v_{xin} = 1\text{ m/s}$, $v_{xin} = 1.5\text{ m/s}$, $v_{xin} = 2\text{ m/s}$ is from 0.42 to 0.49, 0.425 to 0.5, and 0.41 to 0.47 respectively. Table 6.2 summarises the effects of the inlet velocity on the MSF design factors i.e. the $NETD$, flashing down and thermal flashing efficiency.

<table>
<thead>
<tr>
<th>$v_{xin}$ [m/s]</th>
<th>$NETD$ Range [°C]</th>
<th>Flashing Down Range [°C]</th>
<th>Thermal Flashing Efficiency Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>4.95 − 5.69</td>
<td>4.11 − 4.77</td>
<td>0.42 − 0.49</td>
</tr>
<tr>
<td>1.5</td>
<td>4.80 − 5.59</td>
<td>4.17 − 4.88</td>
<td>0.43 − 0.50</td>
</tr>
<tr>
<td>2</td>
<td>5.09 − 4.70</td>
<td>4.00 − 4.65</td>
<td>0.41 − 0.47</td>
</tr>
</tbody>
</table>
In general, it is hard to ascertain which variable (pressure, brine level, velocity, mixing, turbulence, temperature, residence time and flashing shape) affects more the flashing process inside the flashing chamber. This is because the thermo-fluid factors mutually interact. For example, increasing the brine speed creates more
turbulence inside the flashing chamber but it reduces the residence time for the bubble formation. To fully investigate the hydraulic and the thermodynamic effects on the thermo-fluid behaviour inside the evaporation zone would require a parametric study or sensitivity analysis.

6.5. Summary and Conclusions

Generally, the inlet flow rate affects thermal efficiency simultaneously in opposite directions by both increasing it and decreasing it. Increasing the flow rate may lead to an increase in the level of brine in some cases, and also to reduced residence time. These lead to a reduction in thermal efficiency of the flashing chamber system. Increasing the flow rate promotes both turbulence and the mixing process and these improve the thermal efficiency. This can explain the reasons behind the different inlet flow rate effects on the flashing chamber in previous studies [176, 189, 197, 254].

In cases where the brine level is fixed at a certain level, (Chapter 6) thermal efficiency is a function of residence time and both turbulence and mixing processes. Here, a dimensionless quantity that consisted of the ratio between turbulence and mixing effects to the residence time effect on the thermal efficiency can be the best indicator to determine the effect of the inlet flow rate on the thermal flashing efficiency (whether the thermal efficiency improves or not).

For the flashing chamber, it is clear that it is important to improve the mixing process as much as possible, as this leads to an improvement in the phase change rate. This can be attained in many ways and increasing the flow rate is one of these. Furthermore, it is important to give the feed brine enough time for phase change and utilise the entire extended (paid) energy inside the flashing chamber. This can be obtained by using baffles inside the flashing chambers.
The flashing chamber simulated is the first flashing chamber of the MSF system; this means that it has the highest value of thermal losses. The simulated flashing chamber does not contain a baffle and there is not enough time for complete flashing to occur. The length of the flashing chamber is not sufficient to allow the system to reach a close enough value for thermal equilibrium conditions, which can then reduce the value of thermal losses. Thus, increasing the flow rate can increase the mixing but reduces the residence time.

Inlet flow rate affects the size (the length) of phase change, flashing rate and bubble nucleation frequency inside the bubble formation region, but not the maximum magnitude of the nucleation.

Merely checking the $NETD$ is not enough to judge that the flashing chamber performs sufficiently. The $NETD$ is simply an indicator to establish how far the flashing chamber operating conditions deviate from thermal equilibrium conditions. Thus, the flashing rate and bubble nucleation frequency along the flashing chamber are necessary to evaluate the thermo-fluid behaviour inside the flashing chamber.

In Chapter 7, the effects of the inlet brine temperature on the flashing chamber thermo-fluid process are presented under infinite flashing conditions.
7. EFFECTS OF INLET BRINE TEMPERATURE – INFINITE FLASHING

7.1. Introduction

Thermo-fluid behaviour of the flashing process is been influenced by small difference of temperatures [176, 178]. The main goal of this Chapter is to investigate the effect of different inlet brine temperatures ($T_{in} = 383.15$ K, & $\Delta T_{in} = \pm 0.55$K) on thermo-fluid behaviour inside a flashing chamber, without a baffle, operating under an infinite flashing condition ($p_{op} = 1.023$ bar).

The Chapter is organised as follows: we next present the predicted results of the thermo-fluid performance due to $T_{in}=382.6$ K. In § 7.3, the thermo-fluid predicted results of $T_{in}=383.7$ K are provided. Comparisons with discussions of the differences between the cases are presented in § 7.4. Finally, a summary and conclusions are presented in § 7.5.

For each case, thermodynamic process parameters of heat transfer rate, mass transfer rate, temperature distribution and vapour volume fraction inside the evaporation zone are provided. Furthermore, fluid process parameters of particle paths (streamlines in steady flow) of flashing flow and gauge pressure inside the evaporation zone are studied. Non-equilibrium losses ($NETD$), flashing down, flashing efficiency, flashing rate, and maximum bubble nucleation are calculated and plotted.
The same computational model as used in § 5.3 is applied for the current studies, with the same boundary conditions, as Figure 4.2, excepts for the inlet brine temperature.

We have already shown the predicted results of the thermo-fluid performance of $T_{in}=383.15$ K in § 5.2. The remaining two cases are studied in the following § 7.2., and 7.3.

7.2. Effects of $T_{in}=382.6$ K on Thermo-Fluid Performance

The predicted results of the thermo-fluid performance are presented in this section. Figures 7.1 to 7.5 show gauge pressure, flow patterns, thermal performance, vapour volume fraction and mass transfer inside the flashing chamber, respectively.

7.2.1. Gauge pressure distribution

Figures 7.1 shows the gauge pressure distribution inside the flashing chamber. Gauge pressure increases in proportion to depth below the free surface because of the increasing weight of fluid exerting downward force from above. The calculated pressure drop along a horizontal traverse at mid-orifice level is 2179 Pa.
Figur 7.1a  Predicted gauge pressure field (Case: $v_x = 1.5 \text{ m/s}$, $p_{op} = 1.023 \text{ bar}$, $T_{in} = 382.6 \text{ K}$)

Figur 7.1b  Gauge pressure distribution along vertical traverses at $x = 0.31\text{m}$, $0.62\text{m}$, $0.93\text{m}$ (Case: $v_x = 1.5 \text{ m/s}$, $p_{op} = 1.023 \text{ bar}$, $T_{in} = 382.6 \text{ K}$)
Figure 6.1c  Gauge pressure distribution along horizontal traverses at $y = 0.03\text{m}, 0.17\text{m},$ free surface line, $0.3\text{m}$ (Case: $v_x = 1.5 \text{ m/s}, p_{op} = 1.023 \text{ bar}, T_{in} = 382.6 \text{ K}$)

7.2.2. The particle paths and velocity distribution

The particle paths (streamlines) with magnitude of mixture velocity inside the flashing chamber are shown in Figures 7.2a, 7.2b & 7.2c. In the $y$-direction, the mixture speed value decreases until it reaches the minimum at the free-surface and then it increases again. In the $x$-direction, the maximum speed is found to be $1.88 \text{ m/s}$ at the end of the flashing chamber.

A recirculation zone is created below the free surface. The size and length of the recirculation zone can be seen in Figure 7.2a.
Figure 7.2a  Predicted particle paths (streamlines) (Case: $v_x = 1.5 \text{ m/s}$, $p_{op} = 1.023 \text{ bar}$, $T_{in} = 382.6 \text{ K}$)

Figure 7.2b  Mixture speed along vertical traverses at $x = 0.31\text{m}$, $0.62\text{m}$, $0.93\text{m}$

(Case: $v_x = 1.5 \text{ m/s}$, $p_{op} = 1.023 \text{ bar}$, $T_{in} = 382.6 \text{ K}$)
Figure 7.2c  Mixture speed along horizontal traverses at y = 0.03m, 0.17m, free surface line, 0.3m (Case: \( v_s = 1.5 \text{ m/s} \), \( p_{op} = 1.023 \text{ bar} \), \( T_{in} = 382.6 \text{ K} \))

7.2.3. Thermal performance

Figures 7.3a, 7.3b, and 7.3c show the thermal performance inside the flashing chamber. The decrease in brine temperature takes place in both the x and y-direction with different reduction gradient rate.
Figure 7.3a  Mixture speed along horizontal traverses at $y = 0.03\text{m}, 0.17\text{m}$, free surface line, 0.3m (Case: $v_x = 1.5\text{ m/s}, p_{op} = 1.023\text{ bar}, T_{in} = 382.6\text{ K}$)

Figure 6.3b  Temperature distribution along vertical traverses at $x = 0.31\text{m}, 0.62\text{m}, 0.93\text{m}$ (Case: $v_x = 1.5\text{ m/s}, p_{op} = 1.023\text{ bar}, T_{in} = 382.6\text{ K}$)
7.2.4. Vapour volume fraction and mass transfer rate

Figure 7.4a shows the vapour volume fraction in the flashing chamber. In the y-direction, (Figure 7.4b) the values of vapour volume fraction increase, while in the x-direction (Figure 7.4c) the values of vapour volume fraction decrease. The predicted average vapour volume fraction along the free-surface is 0.0063, while the predicted vapour void fraction inside the bubble nucleation region can be seen in Figure 7.4d.

Mass transfer rate in the bubbles nucleation region and over the free surface layer inside the flashing chamber can be seen in Figure 7.5.
Figure 7.4a  Vapour volume fraction (\(\alpha_v\)) fields with two different ranges of scale; (i) upper field range [0 – 1], and (ii) lower field range [0 – 9x10^{-5}] (Case: \(v_x = 1.5\) m/s, \(p_{op} = 1.023\) bar, \(T_{in} = 382.6\) K)
Figure 7.4b  Vapour volume fraction distribution along vertical traverses at $x = 0.31$ m, $0.62$ m, and $0.93$ m (Case: $v_x = 1.5$ m/s, $p_{op} = 1.023$ bar, $T_{in} = 382.6$ K)

Figure 7.4c  Vapour volume fraction distribution along horizontal traverses at $y = 0.03$ m, $0.17$ m and $0.3$ m (vapour region), and along free surface ($y$ varies) (Case: $v_x = 1.5$ m/s, $p_{op} = 1.023$ bar, $T_{in} = 382.6$ K)
Figure 7.4d  Vapour volume fraction distribution along horizontal traverse

$y = 0.03m$ (Case: $v_x = 1.5 \text{ m/s}$, $p_{op} = 1.023 \text{ bar}$, $T_{in} = 382.6 \text{ K}$)
Figure 7.5 Mass transfer rate field with three different ranges of scale; (a) \([-190, +27]\) kg/m³·sec, (b) \([-26, +8]\) kg/m³·sec, and (c) \([-0.0016, +2.4]\) kg/m³·sec
(Case: \(v_s = 1.5\) m/s, \(p_{op} = 1.023\) bar, \(T_{in} = 382.6\) K)
7.3. Effects of $T_{in}=383.7$K on Thermo-Fluid Performance

The predicted results of the thermo-fluid performance (gauge pressure, flow patterns, thermal performance, vapour volume fraction and mass transfer) inside the flashing chamber are presented in Figures 7.6 - 7.11.

7.3.1. Gauge pressure distribution

Figures 7.6a, 6.6b & 6.6c show the distribution of the gauge pressure inside the flashing chamber. The same as the previous cases, the gauge pressure increases with the depth of brine below the free surface, and it decreases along the $x$-direction. The calculated pressure drop along the flashing chamber (at traverse $y = 0.03$ m) is 2179 Pa.

![Gauge Pressure Distribution](image)

**Figure 7.6a** Predicted gauge pressure field (Case: $v_s = 1.5$ m/s, $p_{op} = 1.023$ bar, $T_{in} = 383.7$ K)
Figure 7.6b  Gauge pressure distribution along vertical traverses at \( x = 0.31 \text{m}, 0.62 \text{m}, 0.93 \text{m} \) (Case: \( v_x = 1.5 \text{ m/s}, p_{op} = 1.023 \text{ bar}, T_{in} = 383.7 \text{ K} \))

Figure 7.6b  Gauge pressure distribution along horizontal traverses at \( y = 0.3 \text{m}, 0.17 \text{m}, \text{free surface line}, 0.3 \text{m} \) (Case: \( v_x = 1.5 \text{ m/s}, p_{op} = 1.023 \text{ bar}, T_{in} = 383.7 \text{ K} \))
7.3.2. Particle paths and velocity distribution

Figure 7.7a shows streamlines with a magnitude of mixture velocity inside the flashing chamber. It shows the recirculation zone that is created below the free surface with its location, size and length.

Similar as before, in the y-direction, the speed value decreases until it reaches its minimum value at the free-surface and then increases again in the vapour layer (Figures 7.7b). In the x-direction, speed decreases along the flashing chamber length, however at the end of the flashing chamber exit, it increases to 1.88 m/s and this is the maximum speed found inside the flashing chamber (Figure 7.7c).

Figure 7.7a  Predicted particle paths (streamlines) (Case: \(v_x = 1.5\, \text{m/s} \), \(p_{op} = 1.023\, \text{bar} \), \(T_{in} = 383.7\, \text{K} \))
Figure 7.7b  Mixture speed along vertical traverses at $x = 0.31$ m, $0.62$ m, $0.93$ m
(Case: $v_x = 1.5$ m/s, $p_{op} = 1.023$ bar, $T_{in} = 383.7$ K)

Figure 7.7c  Mixture speed along horizontal traverses at $y = 0.03$ m, $0.17$ m, free surface line, $0.3$ m (Case: $v_x = 1.5$ m/s, $p_{op} = 1.023$ bar, $T_{in} = 383.7$ K)
7.3.3. Thermal performance

The thermal performance inside the evaporation zone can be seen in Figures 7.8a, 7.8b, and 7.8c. The bubble nucleation zone has the highest temperature, and thus the highest vapourisation enthalpy. The flashing process across the flashing chamber causes a reduction in vapourisation enthalpy and then in the brine temperature. This reduction occurs (as shown in Figures 7.8) in both the $x$ and $y$-directions, with different rates.

*Figure 7.8a*  Predicted temperature field (Case: $v_x = 1.5$ m/s, $p_{op} = 1.023$ bar, $T_{in} = 383.7$ K)
Figure 7.8b  Temperature distribution along vertical traverses at $x = 0.31\text{m}$, 0.62m, 0.93m (Case: $v_x = 1.5 \text{ m/s}$, $p_{op} = 1.023 \text{ bar}$, $T_{in} = 383.7 \text{ K}$)

Figure 7.8c  Temperature distribution along horizontal traverses at $y = 0.03\text{m}$, 0.17m, free surface line, 0.3m (Case: $v_x = 1.5 \text{ m/s}$, $p_{op} = 1.023 \text{ bar}$, $T_{in} = 383.7 \text{ K}$)
7.3.4. Vapour volume fraction and mass transfer rate

Figure 7.9a shows the vapour volume fraction in the flashing chamber. In the $y$-direction above the free surface (Figure 7.9b) the values of vapour volume fraction increase linearly, while, in the $x$-direction, (Figure 7.9c) the values of vapour volume fraction decrease along the flashing chamber. Figure 7.9d shows the distribution of vapour volume fraction along $y=0.03$ m, which passes through the bubble nucleation region.

Figure 7.10 shows the mass transfer rate over the free surface, and in the bubble nucleation region in the flashing chamber. Condensation occurs above the free surface at the flashing chamber walls for this reason the mass transfer rate is negative (phase changes from vapour to liquid).
Figure 7.9a  Vapour volume fraction (\(\alpha_v\)) fields with two different ranges of scale; (i) upper field range \([0 – 1]\), and (ii) lower field range \([0 – 9\times10^{-5}]\) (Case: \(v_x = 1.5\) m/s, \(p_{op} = 1.023\) bar, \(T_{in} = 383.7\) K)
Figure 7.9b  Vapour volume fraction distribution along vertical traverses at \( x = 0.31 \text{m}, 0.62 \text{m}, \) and 0.93m (Case: \( v_x = 1.5 \text{ m/s}, p_{op} = 1.023 \text{ bar}, T_{in} = 383.7 \text{ K} \))

Figure 7.9c  Vapour volume fraction distribution along horizontal traverses at \( y = 0.03 \text{m}, 0.17 \text{m} \) and 0.3m (vapour region), and along free surface (\( y \) varies) (Case: \( v_x = 1.5 \text{ m/s}, p_{op} = 1.023 \text{ bar}, T_{in} = 383.7 \text{ K} \))
Figure 7.9d  Vapour volume fraction distribution along horizontal traverse
\( y = 0.03 \text{m} \) (Case: \( v_x = 1.5 \text{ m/s}, \ p_{op} = 1.023 \text{ bar}, \ T_{in} = 383.7 \text{ K} \)
Figure 7.10 Mass transfer rate field with three different ranges of scale; (a) \([-200, +32]\) kg/m\(^3\)sec, (b) \([-26, +8]\) kg/m\(^3\)sec, and (c) \([-0.0016, +2.7]\) kg/m\(^3\)sec

(Case: \(v_x = 1.5\) m/s, \(p_{op} = 1.023\) bar, \(T_{in} = 383.7\) K)
7.4. Discussion of Effects of Inlet Brine Temperature under Infinite Flashing

Inlet brine temperature affects the thermo-fluid performance inside the flashing chamber. These effects are presented in the following sections.

7.4.1 Gauge pressure distribution

It is found that the increasing of the brine temperature does not affect the magnitude of the gauge pressure significantly in the evaporation zone. For all cases, the gauge pressure increases in proportion to the depth below the free surface because of increasing weight with depth. The gauge pressure is not enough to resist the flashing process inside the evaporation zone. Thus, phase change continues across the evaporation zone, and this can be noticed in Figures 5.4, 5.5, 7.4, 7.5, 7.9 & 7.10.

7.4.2. Particle paths (Streamlines) and velocity distribution

The increasing inlet brine temperature affects the streamlines inside the flashing chamber. Figure 7.11 shows the particle paths of the three investigated cases. A recirculation zone is created for the whole cases; it is located at the first half of the flashing chamber and below the free surface. But the size, length and location of recirculation zone changes based on the inlet brine temperature. The streamlines around the recirculation zones are changed based on the inlet brine temperature as it shown in Figure 7.11. For better illustration we divided the first half of the flashing chamber into three areas; (a) shows the streamlines before the recirculation zone into the flashing wall, (b) the centre of the recirculation zone, and (c) is the stream lines after the recirculation zone and before the middle of the flashing chamber.
In area (a), for $T_{in} = 382.6$ K, mixture speed along streamlines has the lowest values compared to the other two cases. While for $T_{in} = 383.15$ K, an intensive streamlines are created just before the recirculation zone. Therefore, streamlines’ distances and velocity magnitude are increased with higher inlet brine temperature. However, the mixture speed in the vapour layer is found to be decreased by increasing of the inlet brine temperature (see Figure 7.2b, & 7.7b).

The shape of the recirculation zone is changed, as it is noticed in area (b). For the different cases of inlet brine temperature, the centre of the recirculation zone keeps located at the same location along the flashing chamber. The higher inlet brine temperature leads to more stretching in the elliptical shape of the circulation zone (for $T_{in} = 382.6$ K, it has the smallest size).

In area (c), for $T_{in} = 382.6$ K, the distances between the streamlines are increased. This means the mixture speed decreased comparing to the other two cases.

The increasing in mixture speed can be a good indicator that the mixture has more vapour phase, in other word more vapourisation occurs, this also can be evident in vapour void fraction, phase change and flashing rate plotted (see Figure 7.13 & 7.15). This means that the flashing is become more violent by increasing the inlet brine temperature. This observation does agree with [318].
Figure 7.11  Effect of inlet brine temperature on predicted particle paths – infinite flashing
7.4.3. Thermal performance

Inlet brine temperature affects the thermal performance as follows; increasing the inlet brine temperature improves the mass transfer and phase change rate along the flashing chamber due to the increase in the flashing range. This can be seen in Figure 7.15. However, the outlet brine temperature increased by increasing the inlet brine temperature, and this leads to move thermal outlet state further away from thermal equilibrium conditions, corresponding to the flashing chamber operation conditions.

Due to the changing of both of mass transfer and phase change rate, the reduction rate of the brine temperature inside the evaporation zone, in both the $x$ and $y$ directions, is changed based on the inlet brine temperature (see Figure 7.3b,c & 7.8b,c).

Inlet brine temperature affects the thermal inertia (see Figure 7.12). A higher inlet brine temperature leads to a higher magnitude of the thermal wave or the vapourisation enthalpy but it does not affect the size (or length) as the inlet mass flow rate does. This proportionally affects the driving force (superheated temperature) of phase change from liquid to vapour thus the mass transfer rate, heat transfer rate vapour volume fraction distribution and bubbles nucleation rate and growth by increasing them as well.

The higher the inlet brine temperature, the higher flashing rate (or phase change rate) occurs and higher flashing range and superheated temperature. Figure 7.13 shows the effect of inlet brine temperature on the flashing rate along the flashing chamber length. It is clear that the flashing rate is increased by an increase in the inlet brine temperature e.g. for $T_{in} = 383.7$ K the maximum calculated flashing rate is 62.76 kW/m².
Figure 7.12 Effect of inlet brine temperature on predicted temperature field—
infinite flashing
7.3.4. Vapour volume fraction and mass transfer

The vapour volume fraction distribution inside the evaporation zone is affected by increasing the inlet brine temperature, due to the increase of the phase change driving force (superheat temperature). Thus, the vapour void fraction and the mass transfer rate are increased by increasing the inlet brine temperature as shown in Figure 7.15.

In the bubble nucleation region, the magnitude of the maximum bubble formation increases in proportion to the inlet brine temperature (see Figure 7.15a). The magnitude of the vapour volume fraction increased throughout three different areas inside the bubble nucleation region. This is evident in the plotted distribution of vapour volume fraction along $y = 0.03$ m, which passes through the bubble nucleation region (see Figure 7.4d & 7.9d). $T_{in} = 383.7$ K has the maximum vapour volume fraction comparing to the other two cases.
In the three studied cases, the effect of the inlet temperature affects the magnitude of phase change inside the bubble formation region, not the size (length). This may be because of the small inlet temperature difference between them (± 0.55 °C). The maximum phase change rate in the bubble formation region is found at $T_{in}=383.7$ K case. On the other hand, the average value of vapour volume fraction at the free surface does not change significantly.

*Figure 7.14* shows the effect of inlet brine temperature on the maximum bubble nucleation frequency along the flashing chamber length. A higher inlet brine temperature leads to a higher nucleation frequency and phase change rate. *Table 7.1* summarises the effects of the inlet brine temperature on the flashing. As previous cases (§6.4.3.), the maximum bubble nucleation frequency and the flashing rate are computed based on Eq. 3.5 and 3.7, respectively. The brine temperatures through the fluid field were extracted and used for this purpose.

*Figure 7.14* Effect of inlet brine temperature on maximum bubble nucleation frequency along horizontal traverses at $y = 0.03$m– infinite flashing
Table 7.1  Summary of the inlet brine temperature effects on the flashing

<table>
<thead>
<tr>
<th>$T_{in}$ [K]</th>
<th>Maximum vapour volume fraction at $y = 0.03$</th>
<th>Maximum flashing rate [kW/m²]</th>
<th>Maximum bubble nucleation frequency [sec⁻¹]</th>
</tr>
</thead>
<tbody>
<tr>
<td>382.6</td>
<td>$7 \times 10^{-5}$</td>
<td>43.53</td>
<td>$7.914 \times 10^6$</td>
</tr>
<tr>
<td>383.15</td>
<td>$7.4 \times 10^{-5}$</td>
<td>50.97</td>
<td>$9.268 \times 10^{10}$</td>
</tr>
<tr>
<td>383.7</td>
<td>$7.95 \times 10^{-5}$</td>
<td>62.76</td>
<td>$1.141 \times 10^{11}$</td>
</tr>
</tbody>
</table>
Figure 7.15a  Effect of inlet brine temperature on predicted vapour volume fraction field— infinite flashing
Figure 7.15b  Effect of inlet brine temperature on predicted mass transfer—infinit flashing
Figure 7.15c  Effect of inlet brine temperature on predicted mass transfer—
infinte flashing
As observed in Figures 7.13-15, the increasing of the inlet temperature leads to greater vapour generation. But, it moves the flashing chamber system to further away from thermal equilibrium conditions, corresponding to the flashing chamber operation conditions, and from reaching the completed flashing. This leads to increase the thermal losses (NETD) inside the flashing chamber.

### 7.4.5. Non-equilibrium losses and thermal flashing efficiency

The inlet brine temperature affects non-equilibrium losses and thermal flashing efficiency inside the flashing chamber. Figure 7.16a shows the distribution of the non-equilibrium losses (NETD) over the flashing chamber outlet orifice for the three inlet brine temperatures. The NETD values range of $T_{\text{in}} = 382.6 \text{ K}$, $T_{\text{in}} = 383.15 \text{ K}$, and $T_{\text{in}} = 383.7 \text{ K}$ from 4.87 to 5.55 °C, 4.8 to 5.59 °C, and 4.9 to 5.61 °C respectively. It is found that $T_{\text{in}} = 382.6 \text{ K}$ has the lowest NETD and $T_{\text{in}} = 383.7 \text{ K}$ has the highest NETD. This indicates that increasing the inlet brine temperature increases the thermal losses, but it still increases the vapour production inside the flashing chamber. Thus, checking the NETD is not enough to judge that the flashing chamber performs sufficiently: the flashing rate and bubble nucleation frequency along the flashing chamber are needed to evaluate the flashing process. As mentioned in Chapter 6, The NETD is an indicator of an incomplete flashing process inside the flashing chamber and it represents residual superheating, corresponding to the saturation temperature of the brine, of the brine exit of the flashing chamber.

The flashing range increases proportion to the inlet brine temperature. However, the same is not necessarily the case for the flashing down which also depends on the outlet brine temperature, may increase, decrease or remain the same. Figure 7.16b shows the effect of inlet brine temperature on the relationship between the NETD and flashing down. When flashing down
increases, the \( NETD \) decreases in all cases. The range of flashing down of \( T_{in}=382.6 \text{ K}, \ T_{in}=383.15 \text{ K}, \text{ and } T_{in}=383.7 \text{ K} \) is from 3.69 to 4.36 °C, 4.17 to 4.88 °C, and 4.84 to 5.53 °C respectively.

**Figure 7.16a** Effect of inlet brine temperature on \( NETD \) distribution over outlet orifice

**Figure 7.16b** Effect of inlet brine temperature on \( NETD \) distribution and flashing down
Figure 7.16c shows the effect of inlet brine temperature on the relationship between the NETD and flashing thermal efficiency. The flashing thermal efficiency increases when the NETD decreases. Thermal flashing efficiency ranges of $T_{in} = 382.6$ K, $T_{in} = 383.15$ K, and $T_{in} = 383.7$ K is from 0.4 to 0.47, 0.43 to 0.5, and 0.46 to 0.53 respectively. It is found that $T_{in} = 382.6$ K has the lowest flashing thermal efficiency and $T_{in} = 383.7$ K has the highest flashing thermal efficiency. These results agree with the experimental works [172, 175, 189, 197, 214, 254]. A summary of the effects of the inlet brine temperature on the MSF design factors i.e. NETD, flashing down and thermal flashing efficiency can be found in Table 7.2.

Table 7.2  Summary of the inlet velocity effects on the MSF design factors

<table>
<thead>
<tr>
<th>$T_{in}$ [K]</th>
<th>NETD range [°C]</th>
<th>Flashing down range [°C]</th>
<th>Thermal flashing efficiency range</th>
</tr>
</thead>
<tbody>
<tr>
<td>382.6</td>
<td>4.87 – 5.55</td>
<td>3.69 – 4.36</td>
<td>0.40 – 0.47</td>
</tr>
<tr>
<td>383.15</td>
<td>4.80 – 5.59</td>
<td>4.17 – 4.88</td>
<td>0.43 – 0.50</td>
</tr>
<tr>
<td>383.7</td>
<td>4.90 – 5.61</td>
<td>4.84 – 5.53</td>
<td>0.46 – 0.53</td>
</tr>
</tbody>
</table>

Figure 7.16c  Effect of inlet brine temperature NETD distribution over flashing thermal efficiency– infinite flashing

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The values of the *NETD* for the three cases might look slightly high: this is because the simulation is of the first flashing chamber of the MSF system which has the highest value of thermal losses. The simulated flashing chamber does not contain a baffle and there is not enough time for complete flashing to occur. The length of the flashing chamber is not sufficiently long enough to allow the system to reach a close enough value of thermal equilibrium conditions.

### 7.5. Summary and Conclusions

Thermodynamically, the amount of flashed off vapour is limited by the available energy. The required energy of phase change from liquid into vapour is known as the enthalpy of vaporization (or latent heat of vaporization). Increasing the brine temperature affects proportionally in thermal internal energy and this helps to overcome the liquid surface resistance of phase change. Thus, energy transfer from thermal to phase change is promoted.

For the flashing chamber, the higher inlet brine temperature leads to a higher flashing range and driving force of phase change from liquid to vapour (superheated temperature) in the evaporation zone. This improves the mass transfer rate, flashing rate, heat transfer rate vapour volume fraction distribution and bubble nucleation rate and growth.

The inlet temperature affects the magnitude of phase change, flashing rate and bubble nucleation frequency inside the bubble formation region, but not the size (length).

Flashing thermal efficiency in the flashing chamber can be improved by increasing the inlet brine temperature. However, the thermal loss (*NETD*)
increases inside the flashing chamber and the flashing is infinite and is moved further away from the completed flashing (the ideal case).

Checking the NETD is not enough to evaluate the flashing process inside the flashing chamber. The NETD is merely an indicator to establish how far the flashing process is from the ideal or the completed flashing case. Therefore, the flashing rate and bubble nucleation frequency along the flashing chamber are necessary for better evaluation of the flashing process.

In Chapter 8, the effects of the inlet flow rate on the thermo-fluid performance of a flashing chamber operating under a finite flashing condition are discussed.
8. EFFECTS OF INLET FLOW RATE – FINITE FLASHING

8.1. Introduction

Investigating the effects of varying the inlet flow rate on the thermo-fluid behaviour of a flashing chamber, without a baffle, operating under a finite flashing condition \((p_{op} = 1.4 \text{ bar})\) is the main objective of this Chapter.

The same flashing chamber described in Chapter 6 except using finite flashing condition, as determined by the operating pressure \((p_{op} = 1.4 \text{ bar})\). Thus, the computational model is applied to a flashing chamber, without a baffle and operating under a finite flashing condition, with same computational domain configuration is used as in Figure 4.2, and with the same three mean inlet velocities profiles \((\overline{v}_{xin} = 1, 1.5, \text{ and } 2 \text{ m/s})\).

Otherwise, the same boundary, operating conditions and assumptions as used in § 5.2, are used for running the simulation of each case.

This Chapter is organised as follows: the predicted thermo-fluid results for \(\overline{v}_{xin} = 1 \text{ m/s} \) case, and \(\overline{v}_{xin} = 2 \text{ m/s} \) case are presented in the first part. Then discussions with comparisons between the different inlet mass flow rate cases are provided.

As before in Chapter 6, for each case, the flashing flow particle paths and gauge pressure inside the evaporation zone were provided. The thermodynamic and thermal performance of the flashing flow, \(e.g.\) heat
transfer rate, mass transfer rate, temperature distribution and vapour volume fraction inside the evaporation zone are provided. Flashing performance factors, e.g. non-equilibrium losses (NETD), flashing down, flashing efficiency, flashing rate, and maximum bubble nucleation are calculated.

In Chapter 5, § 5.4, the predicted results of the thermo-fluid behaviour of \( \nu_{x_{in}} = 1.5 \) m/s were studied and discussed. The remaining two cases are studied in the following § 8.2, § 8.3.

8.2. Effects of \( \nu_{x_{in}} = 1 \) m/s on Thermo-Fluid

Gauge pressure, flow patterns, thermal performance, vapour volume fraction and mass transfer inside the flashing chamber are presented in Figures 8.1 through to 8.5.

8.2.1. Gauge pressure distribution

The gauge pressure distribution inside the flashing chamber is shown in Figures 8.1. Below the free surface, the pressure increases with the depth of brine. A gradient of gauge pressure along the \( x \)-direction for driving the flow from the inlet orifice to the exit orifice can be noticed in Figure 8.1c. The calculated average pressure drop along the flashing chamber is 2146 Pa.
**Figure 8.1a** Predicted gauge pressure field (Case: \(v_x = 1\) m/s, \(p_{op} = 1.4\) bar, \(T_{in} = 383.15\) K)

\(v_x = 1\) m/s  
\(p_{op} = 1.4\) bar  
\(T_{in} = 383.15\) K

**Figure 8.1b** Gauge pressure distribution along vertical traverses at \(x = 0.31\) m, 0.62 m, 0.93 m (Case: \(v_x = 1\) m/s, \(p_{op} = 1.4\) bar, \(T_{in} = 383.15\) K)
8.2.2. The particle paths and the velocity distribution

Flashing flow streamlines with magnitude of mixture velocity inside the flashing chamber can be seen in Figure 8.2a. The free surface separates the brine layer from the vapour layer. A recirculation zone is created below the free surface; its size and length have been predicted.

Figures 8.2b & 8.2c show that the speed value decreases in the y-direction until it reaches the minimum value at the free-surface, and then in the vapour layer, the mixture speed increases again. Horizontally in the brine layer, the maximum speed is found to be 1.82 m/s at the end of the flashing chamber. It is noticed that the mixture speed changes slightly along the bubble nucleation region (Figure 8.2c, $y = 0.03$ m).
Figure 8.2a  Predicted particle paths (streamlines) (Case: \( v_x = 1 \) m/s, \( p_{op} = 1.4 \) bar, \( T_{in} = 383.15 \) K)

Figure 8.2b  Mixture speed along vertical traverses at \( x = 0.31 \) m, 0.62 m, 0.93 m

(Case: \( v_x = 1 \) m/s, \( p_{op} = 1.4 \) bar, \( T_{in} = 383.15 \) K)
8.2.3. Thermal performance

The thermal performance of the flashing flow inside the flashing chamber can be seen in Figure 8.3. Brine temperature is an inductor for enthalpy of vapourisation inside the flashing chamber. It is noted that the brine temperature decreases along the flashing chamber (Figures 8.3a, 8.3b, & 8.3c). The reduction of the brine temperature takes place in both the x and y-direction, but with a different reduction rate. It is noted that the brine temperature distribution does not change along the bubble nucleation region (Figure 8.3c, y = 0.03 m).
Figure 8.3a  Predicted temperature field (Case: $v_x = 1$ m/s, $p_{op} = 1.4$ bar, $T_{in} = 383.15$ K)

Figure 8.3b  Temperature distribution along vertical traverses at $x = 0.31$ m, 0.62 m, 0.93 m (Case: $v_x = 1$ m/s, $p_{op} = 1.4$ bar, $T_{in} = 383.15$ K)
8.2.4. Vapour volume fraction and mass transfer rate

Vapour volume fraction ($\alpha_v$) in the flashing chamber is shown in Figure 8.4, while the mass transfer rate distribution inside the flashing chamber is shown in Figure 8.5.

In the vapour layer (upper the free surface) in the $y$-direction, the values of vapour volume fraction increases, while, in the $x$-direction (Figure 8.4c) the values of vapour volume fraction decrease until it reaches the minimum value at $x = 0.5$ m, where subsequently increases again.

In the brine layer, along the flashing chamber length, the vapour volume fraction decreases, and this occurs because of the reduction in brine temperature as discussed previously (Figure 8.4b, Figure 8.4d).
Figure 8.4d shows the distribution of vapour volume fraction along $y = 0.03 \text{m}$, which passes through the bubble nucleation region. It is noticeable that, in the $x$-direction, the vapour volume fraction is reduced dramatically outside the bubble nucleation region, and it reaches the minimum value (or near to a zero value). This occurs because the phase change driving force has reached a minimum (almost zero), and therefore there is no more phase change occurring and the flashing is finite. Figure 8.5c shows the mass transfer rate in the bubble nucleation region, and it is evident that there is no further mass transfer outside the bubble nucleation region.

The peak values of vapour volume fraction, in the evaporation zone, occur at the free surface and in the bubble nucleation region. Figure 8.4c shows that the value of the vapour volume fraction along the bubble nucleation region is constant. The vapour volume fraction distribution along the saturation temperature line ($y$ varies) is shown in Figure 8.4e.
Figure 8.4a  Vapour volume fraction ($\alpha_v$) fields with two different ranges of scale; (i) upper field range [0 – 1], and (ii) lower field range [0 – 2.2x10^{-5}]

(Case: $v_x = 1$ m/s, $p_{op} = 1.4$ bar, $T_{in} = 383.15$ K)
Figure 8.4b  Vapour volume fraction distribution along vertical traverses at $x = 0.31$ m, 0.62 m, and 0.93 m (Case: $v_x = 1$ m/s, $p_{op} = 1.4$ bar, $T_{in} = 383.15$ K)

Figure 8.4c  Vapour volume fraction distribution along horizontal traverses at $y = 0.03$ m, 0.17 m and 0.3 m (vapour region), and along free surface ($y$ varies) (Case: $v_x = 1$ m/s, $p_{op} = 1.4$ bar, $T_{in} = 383.15$ K)
Figure 8.4d  Vapour volume fraction distribution along horizontal travers $y = 0.03\text{m}$ (Case: $v_x = 1\text{ m/s}$, $p_{op} = 1.4\text{ bar}$, $T_{in} = 383.15\text{ K}$)

Figure 8.4e  Vapour volume fraction distribution along the saturation temperature line ($y$ varies) (Case: $v_x = 1\text{ m/s}$, $p_{op} = 1.4\text{ bar}$, $T_{in} = 383.15\text{ K}$)
Figure 8.5  Mass transfer rate field with three different ranges of scale; (a) 
[-130, +16] kg/m³·sec, (b) [-28, +0.08] kg/m³·sec, and (c) [-0.004, +0.21] kg/m³·sec
(Case: \( v_x = 1 \) m/s, \( p_{op} = 1.4 \) bar, \( T_{in} = 383.15 \) K)
8.3. Effects of $\bar{v}_{xin} = 2\text{m/s}$ on Thermo-Fluid

In this section, the predicted results (Figures 8.6 - 8.10) of the thermo-fluid performance gauge pressure, flow patterns, thermal performance, vapour volume fraction and mass transfer inside the flashing chamber are presented.

8.3.1. Gauge pressure distribution

The gauge pressure distribution inside the flashing chamber is shown in Figures 8.6. In the brine layer, the pressure increases with the depth of brine below the free surface, while gauge pressure decreases along the $x$-direction for driving the brine flow to the outlet orifice. Pressure drop along the flashing chamber is calculated along a horizontal traverse at mid-orifice level and it is found to be 2215 Pa.

![Gauge Pressure Distribution](image)

*Figure 8.6a* Predicted gauge pressure field (Case: $\bar{v}_x = 2\text{ m/s}$, $p_{op} = 1.4\text{ bar}$, $T_{in} = 383.15\text{ K}$)
Figure 8.6b  Gauge pressure distribution along vertical traverses at $x = 0.31\text{m}$, $0.62\text{m}$, $0.93\text{m}$ (Case: $v_x = 2 \text{ m/s}$, $p_{op} = 1.4 \text{ bar}$, $T_{in} = 383.15 \text{ K}$)

Figure 8.6c  Gauge pressure distribution along horizontal traverses at $y = 0.03\text{m}$, $0.17\text{m}$, free surface line, $0.3\text{m}$ (Case: $v_x = 2 \text{ m/s}$, $p_{op} = 1.4 \text{ bar}$, $T_{in} = 383.15 \text{ K}$)
8.3.2. The particle paths and velocity distribution

*Figures 8.7a, 8.7b & 8.7c* show particle paths (streamlines) with magnitude of mixture velocity inside the flashing chamber. The predicted field shows that a recirculation zone is created below the free surface; *Figure 8.7a* shows its location, and size. It is clear that the flow pattern in *Figure 8.7a* has a remarkable difference compared with *Figure 6.7a*. These significant effects are occurred due to the change in the flashing process type from infinite into finite.

In the $y$-direction of the evaporation zone, the speed value decreases until it reaches the minimum value, at the free-surface line and subsequently, increases again. In the brine layer, along the $x$-direction, the mixture speed decreases along the flashing chamber length. However, it increases to 2.11 m/s at the end of the flashing chamber exit and this is the maximum speed found along the evaporation zone. *Figure 8.7c* shows that the mixture speed inside the bubble nucleation region is almost constant.

\[ v_x = 2 \text{ m/s} \]
\[ p_{op} = 1.4 \text{ bar} \]
\[ T_{in} = 383.15 \text{ K} \]

*Figure 8.7a*  Predicted particle paths (streamlines) (Case: $v_x = 2$ m/s, $p_{op} = 1.4$ bar, $T_{in} = 383.15$ K)
Figure 8.7b  Mixture speed along vertical traverses at $x = 0.31\text{m}, 0.62\text{m}, 0.93\text{m}$

(Case: $v_x = 2\text{ m/s}$, $p_{op} = 1.4\text{ bar}$, $T_{in} = 383.15\text{ K}$)

Bubble nucleation region

Figure 8.7c  Mixture speed along horizontal traverses at $y = 0.03\text{m}, 0.17\text{m}$, free surface line, $0.3\text{m}$ (Case: $v_x = 2\text{ m/s}$, $p_{op} = 1.4\text{ bar}$, $T_{in} = 383.15\text{ K}$)
8.3.3. Thermal performance

The thermal performance of the flashing process across the flashing chamber can be seen in Figures 8.8a, 8.8b, & 8.8c. The brine temperature decreases in the $x$ and $y$-directions. Figure 8.8c ($y = 0.03$ m) shows that there is no reduction rate along the bubble nucleation region, thus the brine temperature along the bubble nucleation region is constant.

**Figure 8.8a** Predicted temperature field (Case: $v_x = 2$ m/s, $p_{op} = 1.4$ bar, $T_{in} = 383.15$ K)
Figure 8.8b  Temperature distribution along vertical traverses at $x = 0.31$m, $0.62$m, $0.93$m (Case: $v_x = 2$ m/s, $p_{op} = 1.4$ bar, $T_{in} = 383.15$ K)

Bubble nucleation region

Figure 8.8c  Temperature distribution along horizontal traverses at $y = 0.03$m, $0.17$m, free surface line, $0.3$m (Case: $v_x = 2$ m/s, $p_{op} = 1.4$ bar, $T_{in} = 383.15$ K)
8.3.4. Vapour volume fraction and mass transfer rate

*Figure 8.9a* shows the vapour volume fraction in the evaporation zone inside the flashing chamber. In the brine layer, in the $x$-direction, (*Figure 8.9c*) the values of vapour volume fraction decrease. This happens because of the temperature reduction in the brine. While in the vapour layer, (*Figure 8.9b*) the values of vapour volume fraction increase in the $y$-direction.

*Figure 8.9d* shows the distribution of vapour volume fraction along $y=0.03\text{m}$, which passes through the bubble nucleation region. It is noticed that the vapour volume fraction along the bubble nucleation region is constant. However, it reduces dramatically outside the region. *Figures 8.9e* shows average vapour volume fraction along the saturation temperature line.

The mass transfer rate distributions in the bubble nucleation region and over the free surface layer are shown in *Figure 8.9*. 
Figure 8.9a  Vapour volume fraction ($\alpha_v$) fields with two different ranges of scale; (i) upper field range [0 – 1], and (ii) lower field range [0 – 2.2x10⁻⁵]

(Case: $v_x = 2$ m/s, $p_{op} = 1.4$ bar, $T_{in} = 383.15$ K)
Figure 8.9b  Vapour volume fraction distribution along vertical traverses at 
x=0.31m, 0.62m, and 0.93m (Case: \(v_x = 2 \text{ m/s}, \ p_{op} = 1.4 \text{ bar}, T_{in} = 383.15 \text{ K}\))

Figure 8.9c  Vapour volume fraction distribution along horizontal traverses 
at \(y = 0.03\text{m}, 0.17\text{m} \) and 0.3m (vapour region), and along free surface (\(y \) varies) (Case: \(v_x = 2 \text{ m/s}, \ p_{op} = 1.4 \text{ bar}, T_{in} = 383.15 \text{ K}\))
**Figure 8.9d** Vapour volume fraction distribution along horizontal traverse \(y = 0.03\) m (Case: \(v_x = 2\) m/s, \(p_{\text{op}} = 1.4\) bar, \(T_{\text{in}} = 383.15\) K)

**Figure 8.9e** Vapour volume fraction distribution along the saturation temperature line (\(y\) varies) (Case: \(v_x = 2\) m/s, \(p_{\text{op}} = 1.4\) bar, \(T_{\text{in}} = 383.15\) K)
Figure 8.10  Mass transfer rate field with three different ranges of scale; (a) $[-113.8, +17.71] \text{ kg/m}^3\text{sec}$, (b) $[-28, +0.08] \text{ kg/m}^3\text{sec}$, and (c) $[-0.003, +0.2] \text{ kg/m}^3\text{sec}$(Case: $v_x = 2 \text{ m/s}, p_{op} = 1.4 \text{ bar}, T_{in} = 383.15 \text{ K}$)
8.4. Discussion of Effects of Inlet Flow Rate under Finite Flashing

Inlet flow rate affects the thermo-fluid performance inside the flashing chamber significantly. These effects are presented in the following.

8.4.1 Gauge pressure distribution

The gauge pressure below the free surface increases with depth for all of the studied cases. The change of inlet flow rate affects the gauge pressure; increasing the inlet flow rate increases the gauge pressure slightly (see Figure 8.1b, 5.1b, & 8.6b). The pressure drop along the flashing chamber length is increased by increasing the inlet flow rate.

The level of brine is increased by increasing the inlet flow rate and this can be observed in Figures 8.1b, 5.6b, & 8.6b in the range between 0 to 1000 Pa. The highest brine level occurred when $\overline{v}_{in} = 2$ m/s case and it is located at $x = 0.93$m. The peak brine level location changed as a result of the flow rate case e.g. for the $\overline{v}_{in} = 1$ m/s case, the highest brine level is located at $x = 0.62$, while for $\overline{v}_{in} = 1.5$ m/s and $\overline{v}_{in} = 2$ m/s cases the highest brine level is located at $x = 0.93$m.

8.4.2. Particle paths (streamlines) and velocity distribution

The streamlines of the three investigated cases are shown in Figure 8.11. Inlet flow rate has a significant effect on the flow patterns inside the flashing chamber. The recirculation zone is created at the first half of the flashing chamber. The size, length and location of the recirculation zone changes based on the inlet flow rate. The recirculation zone centre location is shifted forwards away from the inlet orifice due to the increasing of the inlet flow rate as observed in Figures 8.2a, 5.8a, & 8.7a.
In the brine layer, the brine velocity profile at $x = 0.31\text{m}$ can be observed in Figures 8.2b, 5.8b, & 8.8b. A parabolic velocity profile is developed for $\overline{v}_{xin}=1\text{ m/s}$ case, while the uniform velocity profile is remained for both cases $\overline{v}_{xin}=1.5\text{ m/s}$ and $2\text{ m/s}$. At $x = 0.62\text{m}$, it is noticed that a parabolic velocity is developed for $\overline{v}_{xin}=1.5\text{ m/s}$ and $2\text{ m/s}$, while for $\overline{v}_{xin}=1\text{ m/s}$ case the velocity profile is uniform.

In the vapour layer at $y = 0.3\text{m}$ (Figure 8.2c, 5.8c, 8.8c), it is noticed that the behaviour of the mixture speed decreases until it reaches the minimum point and then it increases again, but for $\overline{v}_{xin}=2\text{m/s}$ the mixture speed keeps decreasing along the flashing chamber. This happens because the horizontal traverse ($y = 0.3\text{m}$) crosses the free surface at $x = 0.9\text{m}$ and passes through the brine layer where the mixture contains liquid phase more than the vapour phase thus the speed of the mixture has to be low.
Figure 8.11  Effect of inlet flow rate on predicted particle paths – finite flashing
8.4.3. Thermal performance

Inlet flow rate affects the thermal performance inside the flashing chamber. Increase in the flow rate creates higher mixing and turbulence in the evaporation zone. A higher inlet flow rate leads to a higher thermal (heat) wave inside the evaporation zone. This affects the mass transfer rate and vapour volume fraction distribution (Figure 8.12). Thus, the increase in the flow rate leads to higher phase change and flashing rate (Figure 8.13).

The rate of the brine temperature reduction in the $x$ and $y$-direction changes based on the inlet flow rate case. Figures 8.3b, 5.8b, & 8.7b demonstrate that along vertical traverse $x = 0.31$ m; for $\overline{v}_{xin}=1.5$ m/s and 2 m/s cases the temperature is constant at below $y = 0.05$ m then it starts to reduces dramatically, while for $\overline{v}_{xin}=1$ m/s the temperature reduces gradually. This can be explained as the vertical traverse $x = 0.31$ m passes through the bubble nucleation region for $\overline{v}_{xin}=1.5$ m/s and 2 m/s cases and the temperature inside the region is constant.

The maximum flashing rate for the three studied cases is found to be 0.031 kW/m$^2$. Increasing the flow rate does not affect the magnitude of the flashing rate but it does increase the maximum flashing rate length (Figure 8.13) i.e. the maximum bubble nucleation region size. Thus, $\overline{v}_{xin} = 2$ m/s has the longest bubble nucleation region, and thus the highest vapourisation rate comparing to the other two cases, while $\overline{v}_{xin} = 1$ m/s has the shortest length.

In Figure 8.13 the negative flashing values means that brine temperature is less than the saturation temperature thus there is no more flashing occurs. For showing the flashing rate areas we have focused on a restrict range [0 – 0.6] m along horizontal traverses at $y= 0.03$ m.
In the brine layer along traverse \( y = 0.03 \text{m} \), Figures 8.3c, 5.9c, & 8.8c show that the increase in the inlet flow rate reduces the rate of brine temperature reduction in the \( x \)-direction, thus the outlet temperature is higher when the flow rate is increased, therefore the \( NETD \) is reduced.
Figure 8.12  Effect of inlet flow rate on predicted temperature field – finite flashing
Figure 8.13  Effect of inlet flow rate on flashing rate along horizontal traverses at $y = 0.03$ m; (i) upper field range $[0 – 1.4]$ m, and (ii) lower field range $[0 – 0.6]$ m – finite flashing
8.4.4. Vapour volume fraction and mass transfer

Increasing the flow rate improves the mass transfer and phase change rate along the flashing chamber and increases the bubble nucleation region size (Figures 8.14, & 8.15). The higher flow rate leads to higher mixing and turbulence in the evaporation zone and turbulent flow promotes the rate of conversion of mechanical to heat energy.

Figure 8.14 shows the effect of inlet flow rate on maximum bubble nucleation frequency along the flashing chamber length. In Figure 8.14 we have focused on a restrict range [0 – 0.6] m along horizontal traverses at y = 0.03m to show the maximum bubble nucleation areas. The negative values mean that there is no more bubble nucleation.

A higher inlet flow rate leads to increase the area of the nucleation frequency and phase change rate (Figure 8.14). It is clear that the value of the maximum bubble nucleation frequency (5717 sec⁻¹) for the three studied cases is not affected by the increasing flow rate.

The vapour void fraction increases in the brine layer by increasing the inlet flow rate (Figures 8.4b, 5.10b, & 8.4b). In the bubble region, the maximum vapour void fraction is found to be 6 x 10⁻⁶ for the three cases (Figure 8.4d, 5.10d, & 8.9d), however the area under the curve (maximum vapour void fraction x bubble nucleation region length) is found to increase (bigger) by increasing the inlet flow rate, thus the total vapour void fraction increased. So, \( v_{in} = 2 \text{m/s} \) has the maximum value of the total vapour volume fraction and bubble nucleation region size. Table 8.1 summarises the effects of the inlet flow rate on the flashing process for the three studied cases.
Table 8.1  Finite flashing: summary of the inlet brine flow rate and the maximum length for both the peak of flashing rate and bubble nucleation frequency

<table>
<thead>
<tr>
<th>$\bar{v}_{\text{sin}}$ [m/s]</th>
<th>Length of the peak flashing region [m]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.18</td>
</tr>
<tr>
<td>1.5</td>
<td>0.31</td>
</tr>
<tr>
<td>2</td>
<td>0.41</td>
</tr>
</tbody>
</table>
Figure 8.14  Effect of inlet flow rate on maximum bubble nucleation frequency along horizontal traverses at $y = 0.03$ m; (i) upper field range $[0 – 1.4]$ m, and (ii) lower field range $[0 – 0.6]$ m – finite flashing
Figure 8.15a  Effect of inlet flow rate on predicted vapour volume fraction field– finite flashing
Figure 8.15b  Effect of inlet flow rate on predicted mass transfer—finite flashing
8.4.5. Non-equilibrium losses and thermal flashing efficiency

The NETD is a good indicator of the approach to equilibrium in a flash chamber. As the value of the (NETD) trends to zero, as the system inside the flashing chamber approaches the thermal equilibrium.

It is found that the NETD decreases by increasing the inlet flow rate; \( \dot{\nu}_{xin} = 2 \text{m/s} \) has the lowest NETD (Figure 8.16a). This is referred to the improved mass transfer rate due to the increase in mixing and turbulence. The NETD values range of \( \dot{\nu}_{xin} = 1 \text{m/s}, \dot{\nu}_{xin} = 1.5 \text{m/s} \) and \( \dot{\nu}_{xin} = 2 \text{m/s} \) from -5.43 to -5.17 °C, -4.01 to -3.48 °C, and -2.55 to -1.87 °C respectively.

![Figure 8.16a](image)

**Figure 8.16a** Effect of inlet flow rate on NETD distribution over outlet orifice – finite flashing

**Figure 8.16b** shows the effect of inlet flow rate on the relationship between the NETD and flashing down. When flashing down increases, the NETD decreases in all cases. The range of flashing down of \( \dot{\nu}_{xin} = 1 \text{m/s}, \dot{\nu}_{xin} = 1.5 \text{m/s} \) and \( \dot{\nu}_{xin} = 2 \text{m/s} \) is from 6.06 to 5.85°C, 4.16 to 4.69°C, and 3.20 to 2.55°C respectively.
Figure 8.16b  Effect of inlet flow rate on NETD distribution over flashing down – finite flashing

Figure 8.16c shows the effect of inlet flow rate on the relationship between the NETD and flashing thermal efficiency. The flashing thermal efficiency increases when the NETD is closer to the zero value. Thermal flashing efficiency ranges for $\bar{v}_{xin} = 1$ m/s, $\bar{v}_{xin} = 1.5$ m/s and $\bar{v}_{xin} = 2$ m/s is from 0.13 to 0.14, 0.17 to 0.19, and 0.25 to 0.31 respectively. Table 8.2 summarises the effects of inlet flow rate on MSF design factors i.e. NETD, flashing down and thermal flashing efficiency.

Table 8.2  Finite flashing inlet flow rate effects on MSF design factors

<table>
<thead>
<tr>
<th>$\bar{v}_{xin}$ [m/s]</th>
<th>NETD Range [°C]</th>
<th>Flashing Down Range [°C]</th>
<th>Thermal Flashing Efficiency Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>-5.43 - -5.17</td>
<td>6.06 - 5.85</td>
<td>0.13 - 0.14</td>
</tr>
<tr>
<td>1.5</td>
<td>-4.01 - -3.48</td>
<td>4.16 - 4.69</td>
<td>0.17 - 0.19</td>
</tr>
<tr>
<td>2</td>
<td>-2.55 - -1.87</td>
<td>3.20 - 2.55</td>
<td>0.25 - 0.31</td>
</tr>
</tbody>
</table>
Figure 8.16c  Effect of inlet flow rate NETD distribution over flashing thermal efficiency – finite flashing

### 8.5. Finite Flashing – Summary and Conclusions

The inlet flow rate effects flashing rate, bubble formation rate, and thermal efficiency. Increasing the flow rate leads to an increase in phase change, flashing rate, bubble formation rate and thermal efficiency. This happened due to a promotion of both the turbulence and the mixing process.

Varying the inlet flow rate affected the thermo-fluid behaviour as follows:

- The level of the brine is increased by increasing the inlet flow rate.
- The higher the inlet flow rate, the higher the recirculation zone size and length, the higher mixing and turbulence in the evaporation zone.
- Increasing the flow rate does not affect the magnitude of the flashing rate but it does increase the size of the flashing region.
- The increase in the inlet flow rate reduces the rate of brine temperature gradient along the chamber.
• A higher inlet flow rate leads to a higher nucleation frequency and phase change rate size.
• A higher inlet flow rate leads to a longer bubble nucleation region.
• Increasing the inlet flow rate improves mass transfer, heat transfer and phase change rate.
• Increasing the flow rate improves the rate of mass transfer and phase change along the flashing chamber.
• Increasing the inlet flow rate brings the flashing chamber closer to thermal equilibrium conditions, corresponding to the flashing chamber operation conditions.

The exit brine temperature for finite flashing studied cases was always lower than the saturation temperature of the brine (corresponding to the flashing chamber pressure). Thus, the value of the \( NETD \) is negative.

The ideal performance of the flashing chamber is attained when the flashing process completes just at the exit of the flashing chamber. Here, the \( NETD \) is equal to zero. Thus, finite flashing requires a short flashing chamber length or higher flow rate for reaching the ideal case.

Next, in Chapter 9, the effects of the inlet brine temperature on the flashing chamber thermo-fluid behaviour under finite flashing conditions are presented and discussed.
9. EFFECTS OF INLET BRINE TEMPERATURE – FINITE FLASHING

9.1. Introduction

The objective of this Chapter is to study the effect of varying the inlet brine temperatures on the thermo-fluid performance of the same flashing chamber described in Chapter 7, and the same three inlet temperatures as used in Chapter 7 are applied but now operating under a finite flashing condition \( p_{op} = 1.4 \text{ bar} \).

This Chapter is divided into two sections: the predicted results of the thermo-fluid performance due to different inlet brine temperature cases and then a comparative discussion between the three cases.

In the first section, for \( \Delta T_{in} = \pm 0.55K \) cases, the gauge pressure and the particle paths of flashing flow inside the evaporation zone are studied. The thermodynamic process parameters of temperature distribution vapour volume fraction and mass transfer rate inside the evaporation zone are provided.

In the second section, comparative discussions between the three different temperature cases are presented. These include calculated MSF design factors such as non-equilibrium losses \( NETD \), flashing down, flashing efficiency, flashing rate, and maximum bubble nucleation.
The same computational model as used in § 5.3 is applied for the current studies, with the same boundary conditions, as Figure 4.2, with the exception of the inlet brine temperature, which varies based on the study case $T_{in} = 382.6 \, \text{K}, T_{in} = 383.15 \, \text{K}, \text{and } T_{in} = 383.7 \, \text{K}$.

In § 5.2, we provided the predicted results of the thermo-fluid performance of $T_{in} = 383.15 \, \text{K}$ case. The remaining two cases are studied in the following § 9.2., and 9.3.

9.2. Effects of $T_{in} = 382.6 \, \text{K}$ on Thermo-Fluid Performance

In this section, the predicted results of the thermo-fluid performance (gauge pressure, flow patterns, thermal performance, vapour volume fraction and mass transfer) inside the flashing chamber are presented in Figures 9.1 through to 9.5.

9.2.1. Gauge pressure distribution

Figure 9.1 shows the gauge pressure distribution inside the flashing chamber. It is clear that in the brine layer, the pressure increases with depth of brine, while it decreases along flashing chamber length (Figure 9.1b). The calculated pressure drop along the flashing chamber is 2150 Pa.

In the vapour layer, the gauge pressure is constant and equal to zero Pa (Figure 9.1b). While, over the free surface, the gauge pressure decreases dramatically.
Figure 9.1a  Predicted gauge pressure field (Case: $v_x = 1.5 \text{ m/s}, \ p_{op} = 1.4 \text{ bar}, \ T_{in} = 382.6 \text{ K}$)

Figure 9.1b  Gauge pressure distribution along vertical traverses at $x = 0.31\text{m}, 0.62\text{m}, 0.93\text{m}$ (Case: $v_x = 1.5 \text{ m/s}, \ p_{op} = 1.4 \text{ bar}, \ T_{in} = 382.6 \text{ K}$)

$v_x = 1.5 \text{ m/s}$
$p_{op} = 1.4 \text{ bar}$
$T_{in} = 382.6 \text{ K}$

Crossing the free surface
Figure 9.1b  Gauge pressure distribution along horizontal traverses at $y = 0.03m, 0.17m, \text{free surface line, 0.3m}$ (Case: $v_x = 1.5 \text{ m/s, } p_{op} = 1.4 \text{ bar, } T_{in} = 382.6 \text{ K}$)

9.2.2. Particle paths and velocity distribution

Particle paths (streamlines) with magnitude of mixture velocity inside the flashing chamber are shown in Figure 9.2. Figure 9.2a shows the recirculation zone size, length and location; for the present case it is formed below the free surface in the first half of the flashing chamber.

In the $y$-direction, the mixture speed value decreases until it reaches the minimum value at the free surface line and subsequently, increases again (Figure 9.2b). In the $x$-direction throughout the brine layer, the maximum speed is found to be $1.98 \text{ m/s}$ and located at the end of the flashing chamber (Figure 9.2c). It is shown that the mixture speed through the bubble nucleation region is constant.
Figure 9.2a  Predicted particle paths (streamlines) (Case: $v_x = 1.5 \text{ m/s}$, $p_{op} = 1.4 \text{ bar}$, $T_{in} = 382.6 \text{ K}$)

Figure 9.2b  Mixture speed along vertical traverses at $x = 0.31\text{m}, 0.62\text{m}, 0.93\text{m}$

(Case: $v_x = 1.5 \text{ m/s}$, $p_{op} = 1.4 \text{ bar}$, $T_{in} = 382.6 \text{ K}$)
Figure 9.2c  Mixture speed along horizontal traverses at $y = 0.03\text{m}$, $0.17\text{m}$, free surface line, $0.3\text{m}$ (Case: $v_x = 1.5 \text{ m/s}$, $p_{op} = 1.4 \text{ bar}$, $T_{in} = 382.6 \text{ K}$)

9.2.3. Thermal performance

Figures 9.3a shows the brine temperature distribution of the evaporation zone inside the flashing chamber. It is clear that the brine temperature is decreasing in both $x$ and $y$-directions. Figure 9.3b shows that in the $y$-direction, the brine temperature decreases gradually and it shows that there is no discontinuity in the temperature distribution caused by the free surface. In the bubble nucleation region as shown in Figure 9.3c., the brine temperature is constant along the region, while it starts decreasing outside the region.
Figure 9.3a  Mixture speed along horizontal traverses at \( y = 0.3\,\text{m}, 0.17\,\text{m}, \) free surface line, 0.3m (Case: \( v_x = 1.5\,\text{m/s}, p_{op} = 1.4\,\text{bar}, T_{in} = 382.6\,\text{K} \))

\[ v_x = 1.5\,\text{m/s} \]
\[ p_{op} = 1.4\,\text{bar} \]
\[ T_{in} = 382.6\,\text{K} \]

Figure 9.3b  Temperature distribution along vertical traverses at \( x = 0.31\,\text{m}, 0.62\,\text{m}, 0.93\,\text{m} \) (Case: \( v_x = 1.5\,\text{m/s}, p_{op} = 1.4\,\text{bar}, T_{in} = 382.6\,\text{K} \))
9.2.4. Vapour volume fraction and mass transfer rate

The vapour volume fraction ($\alpha_v$) in the flashing chamber is shown in Figure 9.4a. In the vapour layer, (Figure 9.4b) the values of vapour volume fraction increase in the $y$-direction, while in the $x$-direction (Figure 9.4c) the values of vapour volume fraction decrease until it reaches the minimum value (at $x=0.8$ m) where it increases again.

In the brine layer, the peak vapour volume fraction occurs inside the bubble nucleation region (Figure 9.4a). In the bubble nucleation region two different areas appear when we focused on a restricted range, between 0 and about 0.0001. These areas are numbered area (1) and (2) as shown in Figure 9.4a. It is clear that the vapour volume fraction varies inside the bubble nucleation region; the maximum bubble formation takes place in area (1), in the inlet jet
just below the top edge of the inlet orifice. In area (2) the vapour formation decreases.

*Figure 9.4d* shows the distribution of vapour volume fraction along \( y = 0.03\)m, which passes through the bubbles formation region. The vapour volume fraction distribution along the bubble nucleation region boundaries is shown in *Figures 9.4e*.

Mass transfer in the vapour layer is very active in both directions (from liquid to vapour and the opposite) as shown in *Figure 9.5*. The values of the phase change rate are positive in some places and negative in others especially close to the flashing chamber walls which are used as surfaces for condensate the vapours.

While in the brine layer; the phase change rate values inside the bubble nucleation region are positive (*Figure 9.5c*). This is an indicator that the fluid phase changes from liquid into vapour. It is evident that no further phase change occurs outside the bubble nucleation zone, and thus the flashing process is finite.
Figure 9.4a  Vapour volume fraction (\( \alpha_v \)) fields with two different ranges of scale; (i) upper field range [0 – 1], and (ii) lower field range [0 – 8x10^{-6}] (Case: \( v_x = 1.5 \text{ m/s}, p_{op} = 1.4 \text{ bar}, T_{in} = 382.6 \text{ K} \)
Figure 9.4b Vapour volume fraction distribution along vertical traverses at $x = 0.31\text{m}, 0.62\text{m},$ and $0.93\text{m}$ (Case: $v_x = 1.5\text{ m/s}, p_{op} = 1.4\text{ bar}, T_{in} = 382.6\text{ K}$)

Figure 9.4c Vapour volume fraction distribution along horizontal traverses at $y = 0.03\text{m}, 0.17\text{m}$ and $0.3\text{m}$ (vapour region), and along free surface ($y$ varies) (Case: $v_x = 1.5\text{ m/s}, p_{op} = 1.4\text{ bar}, T_{in} = 382.6\text{ K}$)
Figure 9.4d  Vapour volume fraction distribution along horizontal traverse $y = 0.03 \text{m}$ (Case: $v_x = 1.5 \text{ m/s}$, $p_{op} = 1.4 \text{ bar}$, $T_{in} = 382.6 \text{ K}$)

Figure 9.4e  Average vapour volume fraction along the saturation temperature line ($y$ varies)
Figure 9.5  Mass transfer rate field with three different ranges of scale; (a) \([-120, +14]\) kg/m\(^3\)sec, (b) \([-26, +8]\) kg/m\(^3\)sec, and (c) \([-0.0035, +0.07]\) kg/m\(^3\)sec (Case: \(v_x = 1.5\) m/s, \(p_{op} = 1.4\) bar, \(T_{in} = 382.6\) K)
9.3. Effects of $T_{in}=383.7$ K on Thermo-Fluid Performance

The thermo-fluid predicted results, of $T_{in}=383.7$ K case, inside the flashing chamber operating under finite flashing condition ($p_{op} = 1.4$ bar) are presented in this section. Gauge pressure, flow patterns, thermal performance, vapour volume fraction and mass transfer inside the flashing chamber are presented in Figures 9.6 through to 9.10.

9.3.1. Gauge pressure distribution

*Figure 9.6* shows the predicted gauge pressure distribution inside the flashing chamber. Below the free surface, the pressure increases with depth of brine, while gauge pressure decreases along the $x$-direction. The calculated pressure drop along the flashing chamber is 2190 Pa. The brine level can be estimated from *Figure 9.6b*. Also, *Figure 9.6b* shows that vertically, the pressure decreases dramatically crossing the free surface.

*Figure 9.6a*  Predicted gauge pressure field (Case: $v_x = 1.5$ m/s, $p_{op} = 1.4$ bar, $T_{in} = 383.7$ K)
Figure 9.6b  Gauge pressure distribution along vertical traverses at $x = 0.31\text{m}$,  
$0.62\text{m}$,  $0.93\text{m}$ (Case: $v_x = 1.5 \text{ m/s}$, $p_{op} = 1.4 \text{ bar}$, $T_{in} = 383.7 \text{ K}$)

Figure 9.6b  Gauge pressure distribution along horizontal traverses at $y = 0.03\text{m}$,  
$0.17\text{m}$, free surface line, $0.3\text{m}$ (Case: $v_x = 1.5 \text{ m/s}$, $p_{op} = 1.4 \text{ bar}$, $T_{in} = 383.7 \text{ K}$)
9.3.2. Particle paths and velocity distribution

The particle paths and velocity distribution are shown in Figure 9.7. Figure 9.7a shows streamlines with a magnitude of mixture velocity inside the flashing chamber; a recirculation zone is created below the free surface. The size and length of the recirculation zone can be seen.

Figure 9.7b shows the mixture speed distribution in the brine layer and the vapour layer. In the y-direction, the speed value decreases until it reaches a minimum value, at the free-surface line and subsequently, increases again. As shown in Figure 9.7c, in the x-direction, the mixture speed decreases along the flashing chamber length, however at the end of the flashing chamber exit, it increases to 1.94 m/s and this is the maximum speed found inside the flashing chamber. It is shown that the mixture speed inside the bubble nucleation region is almost constant.

\[ v_x = 1.5 \text{ m/s} \]
\[ p_{op} = 1.4 \text{ bar} \]
\[ T_{in} = 383.7 \text{ K} \]

*Figure 9.7a*  Predicted particle paths (streamlines) (Case: \( v_x = 1.5 \text{ m/s}, p_{op} = 1.4 \text{ bar}, T_{in} = 383.7 \text{ K} \)
**Figure 9.7b**  Mixture speed along vertical traverses at $x = 0.31\text{m}, 0.62\text{m}, 0.93\text{m}$

(Case: $v_x = 1.5 \text{ m/s}, p_{op} = 1.4 \text{ bar}, T_{in} = 383.7 \text{ K}$)

**Figure 9.7c**  Mixture speed along horizontal traverses at $y = 0.03\text{m}, 0.17\text{m}$, free surface line, 0.3m (Case: $v_x = 1.5 \text{ m/s}, p_{op} = 1.4 \text{ bar}, T_{in} = 383.7 \text{ K}$)
9.3.3. Thermal performance

The predicted thermal performance of the flashing process across the flashing chamber is shown in Figure 9.8. It is observed that the brine temperature reduces in the $x$ and $y$-direction.

Figures 9.8a shows there is no discontinuity is created due to the free surface between the brine and vapour layers. Vertically, temperature decreases as shown in Figure 9.8b. Horizontally, in the brine layer, the brine temperature is constant inside the bubble nucleation region, but it decreases outside the region along the flashing chamber. The bubble nucleation zone has the highest vapourisation enthalpy due to the highest brine temperature.

Figure 9.8a  Predicted temperature field (Case: $v_x = 1.5$ m/s, $p_{op} = 1.4$ bar, $T_{in} = 383.7$ K)
**Figure 9.8b** Temperature distribution along vertical traverses at $x = 0.31$ m, 0.62 m, 0.93 m (Case: $v_x = 1.5$ m/s, $p_{op} = 1.4$ bar, $T_{in} = 383.7$ K)

**Figure 9.8c** Temperature distribution along horizontal traverses at $y = 0.03$ m, 0.17 m, free surface line, 0.3 m (Case: $v_x = 1.5$ m/s, $p_{op} = 1.4$ bar, $T_{in} = 383.7$ K)
9.3.4. Vapour volume fraction and mass transfer rate

The vapour volume fraction and the mass transfer rate distribution in the flashing chamber are shown in Figures 9.9 & 9.10.

*Figure 9.9a* shows the vapour volume fraction in the evaporation zone inside the flashing chamber. The upper image (a) indicates the full range of vapour volume fraction scale from 0 to 1, and the lower set of images (b) focuses on a specific range, between 0 and 0.00004. The peak values of vapour volume fraction occur at the free surface and in the bubble nucleation region.

In the *y*-direction, (*Figure 9.9b*) the values of vapour volume fraction increase, while in the *x*-direction, (*Figure 9.9c*) the values of vapour volume fraction decrease due to the brine temperature reductions.

*Figure 9.9d* shows the distribution of vapour volume fraction along *y*=0.03 m, which passes through the bubble formation region. In the bubble nucleation region two different areas are numbered; the maximum bubble formation takes place in area (1), in the inlet jet just below the top edge of the inlet orifice. *Figures 9.9e* shows vapour volume fraction along the saturation temperature line.
Figure 9.9a Vapour volume fraction ($\alpha_v$) fields with two different ranges of scale; (i) upper field range [0 – 1], and (ii) lower field range [0 – 4x10^{-5}] (Case: $v_x = 1.5$ m/s, $p_{op} = 1.4$ bar, $T_{in} = 383.7$ K)

$v_x = 1.5$ m/s
$p_{op} = 1.4$ bar
$T_{in} = 383.7$ K
Figure 9.9b  Vapour volume fraction distribution along vertical traverses at $x = 0.31 \text{m}, 0.62 \text{m}, \text{and } 0.93 \text{m}$ (Case: $v_x = 1.5 \text{ m/s}, p_{op} = 1.4 \text{ bar}, T_{in} = 383.7 \text{ K}$)

Figure 9.9c  Vapour volume fraction distribution along horizontal traverses at $y = 0.03 \text{m}, 0.17 \text{m} \text{ and } 0.3 \text{m}$ (vapour region), and along free surface ($y$ varies) (Case: $v_x = 1.5 \text{ m/s}, p_{op} = 1.4 \text{ bar}, T_{in} = 383.7 \text{ K}$)
Figure 9.9d  Vapour volume fraction distribution along horizontal traverse $y$ =0.03m (Case: $v_x = 1.5$ m/s, $p_{op} = 1.4$ bar, $T_{in} = 383.7$ K)

Figure 9.9e  Average vapour volume fraction along the saturation temperature line ($y$ varies)
Figure 9.10 Mass transfer rate field with three different ranges of scale; (a) [-130, +16] kg/m³·sec, (b) [-26, +8] kg/m³·sec, and (c) [-0.0037, +0.34] kg/m³·sec

(Case: \( v_s = 1.5 \text{ m/s} \), \( p_{op} = 1.4 \text{ bar} \), \( T_{in} = 383.7 \text{ K} \))
9.4. Discussion of Effects of Inlet Brine Temperature under Finite Flashing

Thermo-fluid behaviour inside the flashing chamber is influenced by changing the inlet brine temperature. The effects are presented in the following sections.

9.4.1 Gauge pressure distribution

For all of the cases studied, the gauge pressure distributions do not change significantly due to the brine temperatures change, as evident in Figures 9.1, 5.7, & 9.6.

In the brine layer, the pressure increases with depth of brine, while it decreases along the length of the flashing chamber. The calculated pressure drop along the horizontal traverse at \( y = 0.3 \text{m} \) inside the flashing chamber is found to be increased by increasing the brine temperature, thus the \( T_{in} = 383.7 \text{K} \) case has the highest pressure drop. In the vapour layer, the gauge pressure is effectively zero Pa for all of the studied cases.

Over the free surface, the gauge pressure decreases dramatically, and the brine level decreases slightly by increasing the brine temperature as evident in Figures 9.1b, 5.7b & 9.6b. The brine level at \( x = 0.93 \text{ m} \) for \( T_{in} = 382.6 \text{K} \) case, is found to be 0.26 m, while for \( T_{in} = 383.7 \text{ K} \) case, it is found to be 0.25 m.

9.4.2. Particle paths (streamlines) and velocity distribution

The inlet brine temperature clearly affects the flow (Figure 9.11.). The size and the shape of the recirculation zone is different for each case, e.g. \( T_{in} = 382.6 \text{K} \) has the smallest size. The higher inlet brine temperature leads to more stretching in the elliptical shape of the circulation zone. However, the
centre of the recirculation zone remains at the same location along the flashing chamber.

Increasing the brine temperature affects the mixture speed as evident from Figures 9.2b, 5.7b, & 9.7b. For instance, in the brine layer, the maximum speed along vertical traverse \( x = 0.62 \text{m} \) is found to be 0.85 m/s when the \( T_{in} = 382.6 \text{ K} \), while it is found to be 0.96 m/s when \( T_{in} = 383.7 \text{ K} \). This is due to the increase of vapour phase in the mixture, due to the increases in phase change inside the evaporation zone. Thus the mixture contains more vapour than liquid phase, so velocity magnitude is increased.

Horizontally, in the vapour layers, the mixture speed distribution shows that speed decreases until reaching the minimum point somewhere along the flashing chamber and then it increases again (Figures 9.2c, 5.7c, & 9.7c). The increasing brine temperature affects the location of the minimum points as follows: for \( T_{in} = 382.6 \text{ K} \), it is found to be at 0.7 m, for \( T_{in} = 383.15 \text{ K} \), it is found to be at 0.75m, and for \( T_{in} = 383.7 \text{ K} \), it is found to be at 0.8 m.
Figure 9.11 Effect of inlet brine temperature on predicted particle paths – finite flashing
9.4.3. Thermal performance

The thermal performance inside the flashing chamber is affected by the change in brine temperature. The change of the brine temperature distribution for the three investigated cases can be observed in Figure 9.12.

A higher inlet brine temperature leads to a higher vapourisation enthalpy and a higher thermal (heat) wave as evident from Figure 9.12. Thus, the flashing range increases with the phase change driving force increased, thus the mass transfer rate, heat transfer rate, and bubble nucleation rate increase also.

The inlet brine temperature affects the flashing rate along the flashing chamber length as shown in Figure 9.13. The higher inlet brine temperature and the higher flashing rate (or phase change rate) occurs. This happens due to the higher flashing range and superheated temperature. It is obvious that the brine temperature affects the magnitude of the flashing rate but not the length (Figure 9.13). The length of the peak flashing rate is 0.3m for the three studied cases, while the magnitude of the flashing rate is as following for the three cases: 0.0012 kW/m$^2$, 0.0314 kW/m$^2$, and 0.1445 kW/m$^2$ for $T_{in}$=382.6 K, $T_{in}$=383.15 K, & $T_{in}$=383.7 K, respectively.

Due to the changes that occurred in the flashing rate, the decreasing rate of brine temperature is affected. The decreasing rate of the brine temperature in both $x$ and $y$ directions can be observed in Figures 9.3b,c, 5.3b,c & 9.8b,c. Thus, the outlet brine temperature at the outlet orifice is changed based on the studied case. It is found that the $T_{in}$=383.7 K case has the highest outlet temperature in comparison to the two other cases, and this means its thermal state is closer to the thermal equilibrium, corresponding to the flashing chamber operating conditions.
Figure 9.12  Effect of inlet brine temperature on predicted temperature field paths – finite flashing
Figure 9.13  Effect of inlet brine temperature on flashing rate along horizontal traverses at $y = 0.03$m; i) field range $[0 – 1.4]$ m, (ii) field range $[0 – 0.45]$ m, (iii) $T_{in}=382.6$ K field range $[0 – 0.35]$ paths – finite flashing
9.4.4. Vapour volume fraction and mass transfer

Inlet brine temperature has effects on both the vapour volume fraction distribution and mass transfer rate inside the flashing chamber. These effects can be seen in Figure 9.14. The higher inlet brine temperature leads to a higher flashing range and superheated temperature in the evaporation zone. This improves mass transfer, heat transfer and phase change rate.

In the bubble nucleation region, increasing the inlet brine temperature increases the maximum magnitude of the vapour void fraction inside the bubble nucleation region as shown in Figures 9.4a, 5.10d, & 9.9d. It is found that the maximum vapour void fraction for the three cases is as follows: $2 \times 10^{-6}$, $6 \times 10^{-6}$, $1 \times 10^{-5}$ for $T_{in} = 382.6$ K, $T_{in} = 383.15$ K, & $T_{in} = 383.7$ K, respectively. Thus, the maximum produced vapourisation amount occurs when $T_{in} = 383.7$ K.

The maximum magnitude of bubble nucleation frequency along the flashing chamber length is increased by increasing the inlet brine temperature. A higher inlet brine temperature leads to a higher nucleation frequency and phase change rate (Figure 9.14). It is found that the maximum nucleation frequency for the three cases is as follows: $2.2 \times 10^4$, $5.72 \times 10^5$, and $2.63 \times 10^8$ for $T_{in} = 382.6$ K, $T_{in} = 383.15$ K, & $T_{in} = 383.7$ K, respectively. Thus, the maximum produced vapourisation amount occurred when $T_{in} = 383.7$ K. Table 9.1 summarises the effects of the inlet brine temperature on the flashing process.

<table>
<thead>
<tr>
<th>$T_{in}$ [K]</th>
<th>Maximum vapour volume fraction at $y = 0.03$m</th>
<th>Maximum flashing rate [kW/m$^2$]</th>
<th>Maximum bubble nucleation frequency [sec$^{-1}$]</th>
</tr>
</thead>
<tbody>
<tr>
<td>382.6</td>
<td>$2 \times 10^{-6}$</td>
<td>0.0012</td>
<td>$2.20 \times 10^4$</td>
</tr>
<tr>
<td>383.15</td>
<td>$6 \times 10^{-6}$</td>
<td>0.0314</td>
<td>$5.72 \times 10^5$</td>
</tr>
<tr>
<td>383.7</td>
<td>$10 \times 10^{-6}$</td>
<td>0.1445</td>
<td>$2.63 \times 10^8$</td>
</tr>
</tbody>
</table>
Figure 9.14  Effect of inlet brine temperature on maximum bubble nucleation frequency along horizontal traverses at $y = 0.03 \text{m}$; (i) field range [0 – 1.4] m, (ii) field range [0 – 0.45] m, (iii) $T_{in}=382.6 \text{K}$ field range [0 – 0.35] – finite flashing
Figure 9.15a  Effect of inlet brine temperature on predicted vapour volume fraction field—finite flashing
Figure 9.15b  Effect of inlet brine temperature on predicted mass transfer—finite flashing

$v_x = 1.5 \text{ m/s}$
$p_{op} = 14 \text{ bar}$
$T_{in} = 382.6 \text{ K}$
9.4.5. Non-equilibrium losses and thermal flashing efficiency

As we discussed above, increasing the inlet temperature leads to greater vapour generation, and reduces the rate of the brine temperature reduction. Thus, the outlet brine temperature at the outlet orifice moves closer to the thermal equilibrium condition. Furthermore, increasing the inlet temperature leads to the NETD tending to zero.

The NETD distribution for the three studied cases is shown in Figure 7.16a, where the NETD values range of $T_{in}=382.6$ K, $T_{in}=383.15$ K, and $T_{in}=383.7$ K from $-4.15$ to $-3.74$ °C, $-4.01$ to $-3.48$ °C, and $-3.89$ to $-3.32$ °C respectively. It is found that $T_{in}=382.6$ K has the lowest NETD (furthest from thermal equilibrium). However, it is found that $T_{in}=383.7$ K has the highest NETD (closest to thermal equilibrium). Thus the increase of the inlet brine temperature leads to conveying the flashing chamber closer to thermal equilibrium.

Figure 9.16b shows the effect of inlet brine temperature on the relationship between the NETD and flashing down. When flashing down increases, the NETD increase (further from zero value) in all cases. The range of flashing down of $T_{in}=382.6$ K, $T_{in}=383.15$ K, and $T_{in}=383.7$ K is from 4.02 to 4.42 °C, 4.16 to 4.69 °C, and 4.70 to 5.27 °C respectively.

The thermal flashing efficiency is increased when the inlet brine temperature increases. Figure 9.16c shows the effect of inlet brine temperature on the relationship between the NETD and flashing thermal efficiency. The flashing thermal efficiency increases when the NETD increase (tends to zero). Thermal flashing efficiency ranges of $T_{in}=382.6$ K, $T_{in}=383.15$ K, and $T_{in}=383.7$ K from 0.06 to 0.07, 0.17 to 0.19, and 0.26 to 0.29 respectively. It is found that $T_{in}=382.6$ K has the lowest flashing thermal efficiency. However, it is found that $T_{in}=383.7$ K has the highest flashing thermal efficiency.
Figure 9.16a  Effect of inlet brine temperature on NETD distribution over outlet orifice – finite flashing

Figure 9.16b  Effect of inlet brine temperature on NETD distribution over flashing down– finite flashing
A summary of the effects of the inlet brine temperature on the MSF design factors i.e. $NETD$, flashing down and thermal flashing efficiency can be found in Table 9.2.

Table 9.2  Finite flashing – inlet brine temperature effects on MSF design factors

<table>
<thead>
<tr>
<th>$T_{in}$ [K]</th>
<th>$NETD$ range [$^\circ$C]</th>
<th>Flashing down range [$^\circ$C]</th>
<th>Thermal flashing efficiency range</th>
</tr>
</thead>
<tbody>
<tr>
<td>382.6</td>
<td>-4.15 – -3.74</td>
<td>4.02 – 4.42</td>
<td>0.06  – 0.07</td>
</tr>
<tr>
<td>383.15</td>
<td>-4.01 – -3.48</td>
<td>4.16 – 4.69</td>
<td>0.17  – 0.19</td>
</tr>
<tr>
<td>383.7</td>
<td>-3.89 – -3.32</td>
<td>4.70 – 5.27</td>
<td>0.26  – 0.29</td>
</tr>
</tbody>
</table>

9.5. Finite Flashing Summary and Conclusions

Increasing the inlet brine temperature leads to an increased driving force (superheated temperature) of phase change from liquid to vapour. Thus the mass transfer rate, flashing rate, heat transfer rate vapour volume fraction distribution and bubble nucleation rate and growth are improved and increased.
Varying the inlet brine temperature affects the thermo-fluid behaviour as follows:

- The calculated pressure drop along the flashing chamber length is found to be increased by increasing the brine temperature. While the brine level slightly decreased.
- The higher inlet brine temperature leads to bigger size of the circulation zone with higher mixture speed in the brine layer.
- Increasing the brine temperature leads to increase the magnitude of the maximum flashing rate but it does affect the maximum flashing rate length (size).
- The increase in the inlet brine temperature reduces the rate of brine temperature reduction.
- Increasing the brine temperature improves the mass transfer and phase change rate along the flashing chamber.
- A higher inlet brine temperature leads to a higher nucleation frequency and phase change rate.
- The flashing chamber can be closer to thermal equilibrium conditions, corresponding to the flashing chamber operation conditions.
- The thermal flashing efficiency is increased when the inlet brine temperature increased.

Flashing thermal efficiency in the flashing chamber is improved by increasing the inlet brine temperature; it brings the flashing chamber closer to thermal equilibrium conditions, corresponding to the flashing chamber operation conditions. Thermal losses (NETD) tend to zero value inside the flashing chamber.

Next, in Chapter 10, an overall summary and discussion are presented.
10. OVERALL SUMMARY & DISCUSSION

10.1 Introduction

The dissertation overall summary and discussion are presented in this Chapter.

10.2. Overall Summary

Flash chambers have the highest exergy in the MSF system and the evaporation zone has the highest exergy in each flashing chamber. Thus the evaporation zone is considered as the most important part of the flashing chamber.

The flashing chamber has been treated as an open system, mainly because the boundaries of the system are permeable both to energy and mass. According to the phase type and fluid behaviour, the chamber space may be divided into a number of layers and regions. In the horizontal sense, there are two main regions: A – the bubble nucleation region, near the inlet, which includes recirculating flow; and B - the channel flow, further downstream, which is predominantly unidirectional. In the vertical direction, it may be considered as comprising of three layers: I – the brine flow layer, near the base, II – the free surface and III - the vapour layer.

Vapourisation inside a desalination flashing chamber arises from the surface evaporation with flashing, or self-boiling, of liquid brine due to a reduction of pressure, and it is a complex, multiphase, thermo-fluid phenomenon. In
multistage flash (MSF) desalination systems in particular, flashing flow is the key to desalination of feed brine at every stage.

The flashing process has been classified into two types: finite flashing (completed) and infinite flashing (incomplete). In infinite flashing, the phase change continues to occur until the end of the flashing chamber, while, for finite flashing, the phase change stops when the flashing driving force, the superheat temperature, reaches zero. Each of the two flashing processes can be occurred based on the flashing chamber operation conditions. Operating parameters’ variables can affect the flashing chamber performance differently, based on the type of flashing.

A computational model for the flashing process inside a flashing chamber was developed around a two-phase VOF formulation. Two different phase-change mechanisms are allowed for, based on the saturation temperature and on the vapour pressure, respectively. These enable the model to compute the phase change regions, also the shape of the free surface. The model was applied to solve for steady multiphase flow inside a flashing chamber without a baffle, and thus to map the chamber flow pattern and behaviour.

Predicted results have been compared with regard to the thermo-fluid performance; gauge pressure, flow patterns, thermal performance, vapour volume fraction and the mass transfer inside the flashing chamber.

Our simulation results have been compared with the available measured values. The simulation results give average outlet temperature and average vapour temperature that closely agreed well with the values of a real MSF plant. Two main mechanisms of phase change produced by the flashing process are captured in the simulation. The vapour bubbles are formed at the entrance to the flashing chamber, and bubble production reduces along its
length. At the free surface of the liquid there is also a phase change and mass transfer. Results can be used to estimate MSF design factors such as non-equilibrium temperature difference, NETD, and flashing efficiency.

The effects of inlet flow rate and inlet brine temperatures on the thermos-fluid behaviour under both of infinite and finite flashing conditions were investigated. The model was applied to solve for steady multiphase flow inside a flashing chamber without a baffle, and thus to map the chamber flow pattern and behaviour. The main observations of these CFD investigations are as follows:

10.2.1. Flashing chamber operating under infinite flashing condition

- The inlet flow rate affects thermal efficiency (Figure 6.16c) in opposing directions by increasing and decreasing it. Increasing the flow rate leads to no change in the level of brine, and a reduced residence time. These lead to a reduction in the thermal efficiency of the flashing chamber system.
- Increasing inlet flow rate leads to an increase in both turbulence and mixing process and these in turn lead to an improvement in the thermal efficiency (Figure 10.1).
- Increasing the inlet brine temperature leads to an increased driving force (superheat temperature) of phase change from liquid to vapour, thus the mass transfer rate, flashing rate, heat transfer rate, vapour volume fraction distribution and bubble nucleation rate and growth are increased.
Figure 10.1 Sketch of the effects of inlet flow rate on the flashing process inside a flashing chamber, operating under infinite flashing conditions.

\[ T_{in} \]

\[ V_{in} = 2 \, \text{m/s} \]

\[ V_{in} = 1.5 \, \text{m/s} \]

\[ V_{in} = 1 \, \text{m/s} \]

\[ T_{out} \]

\[ T_{out} \]

\[ T_{sat} \]

\[ T_{in} \]

\[ T_{in} \]

\[ T_{in} = 382.7 \, \text{K} \]

\[ T_{in} = 383.15 \, \text{K} \]

\[ T_{in} = 383.6 \, \text{K} \]

Figure 10.2 Sketch of the effect of inlet brine temperature on the flashing process inside a flashing chamber, operating under infinite flashing conditions.

\[ T_{in} \]

\[ T_{in} \]

\[ T_{in} \]

\[ T_{sat} \]

\[ T_{in} \]

\[ T_{out} \]

\[ T_{out} \]

\[ T_{out} \]

\[ T_{out} \]
• Thermal flashing efficiency in the flashing chamber can be improved by increasing the inlet brine temperature. However, the flashing chamber system can be further away from thermal equilibrium conditions, corresponding to the flashing chamber operating conditions. Thermal losses ($NETD$) may increase inside the flashing chamber (Figure 10.2).

10.2.2. Flashing chamber operating under finite flashing condition

• Inlet flow rate affects flashing rate, bubble formation rate, and thermal efficiency. Increasing the flow rate leads to an increase in phase change, flashing rate, bubble formation rate and thermal efficiency. This takes place due to improve both turbulence and the mixing process.

• Increasing the inlet flow rate may improve the mass transfer, heat transfer and phase change rate. As a result, the flashing process inside the flashing chamber is tend to reaching the thermal equilibrium conditions (and complete flashing), corresponding to the flashing chamber operation conditions (Figure 10.3).

• Increasing the inlet brine temperature leads to an increased driving force (superheat temperature) of phase change from liquid to vapour thus the mass transfer rate, flashing rate, heat transfer rate vapour volume fraction distribution and bubbles nucleation rate and growth are increased.

Thermal flashing efficiency in the flashing chamber is improved by increasing the inlet brine temperature. Furthermore, the flashing chamber system tends to be closer to thermal equilibrium conditions, corresponding to the flashing chamber operation conditions. Thermal losses ($NETD$) tend to zero value inside the flashing chamber (Figure 10.4).
**Figure 10.3** Sketch of the effects of inlet flowrate on the flashing process inside a flashing chamber, operating under finite flashing conditions.

**Figure 10.4** Sketch of the effects of inlet brine temperature on the flashing process inside a flashing chamber, operating under finite flashing conditions.
11. CONCLUSIONS

- The flashing process should be classified as a finite or infinite (flashing) process. Depending on which of the two flashing process variations occurs — finite or infinite — may determine the thermo-fluid behaviour inside the evaporation zone inside the flashing chamber.

- A VOF two-phase computational model of the evaporation zone inside a flashing chamber has been developed. The model is included heat transfer and mass transfer, as well as turbulent flow.

- Two different phase-change mechanisms are allowed for, based on the saturation temperature and the vapour pressure, respectively. These enable the model to compute the phase change regions, also the shape of the free surface.

- The model has been applied to a typical horizontal-flow flashing chamber without a flow baffle. In a preliminary steady-state solution the flow field features, including the recirculation zone and the free surface level and shape are predicted. The simulation results give average outlet temperature and average vapour temperature values that agree closely with those of a real MSF plant.

- The simulation results have been analysed to estimate MSF design factors including the non-equilibrium temperature difference, \textit{NETD}, flashing down and flashing efficiency.

- The model was applied for further studies such as investigations of the effects of the brine temperature, velocity and flashing chamber geometry on the thermo-fluid behaviour for both finite and infinite flashing process classifications.
• Increasing brine flow rate and its temperature improve the flashing performance and vapour production.

• Checking the \textit{NETD} is not enough to estimate the flashing chamber performance. The flashing rate and the bubble nucleation frequency are needed to explore the flashing performance inside the flashing chamber.

• Computational field modelling is valuable for a better understanding of the combined heat transfer, mass transfer and the fluid dynamics during the flashing flow evaporation processes inside the evaporation zone in the flashing chamber.

• Our computational method is applicable to assisting in the design procedure for MSF systems.
12. RECOMMENDATIONS FOR FUTURE WORK

12.1. Flashing Chambers with Different Configurations

The computational modelling can be applied to flashing chambers with different configurations, e.g. with baffles or with different shapes. This can be used to investigate the effects of flashing chambers’ configurations, e.g. baffle shape, location and height on the thermo-fluid behaviour in the flashing chambers.

12.2. Transient Solution

Unsteady solutions are useful to investigate the bubble growing stages, rising and transport to the free surface. They can be applied for predicting the shape of the free surface as well. Furthermore, it can help to investigate the effects of the brine residence time in the flashing chamber.

12.3. Series of Flashing Chambers

The computational modelling can be applied to study the flashing process simultaneously inside more than one flashing chamber, e.g. three flashing chambers at a time. In this case the second flashing chamber can be the test section. Studying three flashing chambers simultaneously allows the computational model to predict a realistic inlet velocity profile in the second flashing chamber. Furthermore, using three flashing chambers allows one to obtain a more realistic and accurate distributions inside the flashing chamber.
12.4. Effects of Orifice Configurations

A 3D model can be used to investigate the effect of orifice shape on the flashing process inside the flashing chamber. The orifice shape affects the velocity distribution in the flashing chamber, thus the phase change rate, mass transfer and the heat transfer inside the flashing chamber will be affected also. Furthermore, the velocity profile can change the dynamics and behaviour of the bubbles inside the bubble nucleation region specifically, and thus the flashing performance in general.

12.5. Effects of Bubble Size and Shape

The computational model can be used to study the effects of the bubble size and shape on the thermo-fluid behaviour inside the evaporation zone in the flashing chamber.

12.6. Other Different Applications

The flashing process exists in many different applications e.g. multi-effect desolation (MED), nuclear application. Thus, the computational model can be applied to these applications for investigation of the thermo-fluid behaviour.
APPENDICES A – I
APPENDIX A  MSF PLANT OF AL KHOBAR II, SAUDI ARABIA

In this appendix, a set of typical pictures of a MSF plant Al Khobar II are presented with flashing chambers. I am grateful to Dr. Philippe Bandelier, CEA Grenoble, for making these images available.

*Table A.1* Operating conditions of a MSF plant of Al Khobar II (1983)

<table>
<thead>
<tr>
<th>Variables</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Type of MSF</td>
<td>Brine recirculation</td>
</tr>
<tr>
<td>Capacity</td>
<td>223,000 m$^3$/day</td>
</tr>
<tr>
<td>Number of trains</td>
<td>10 parallel trains</td>
</tr>
<tr>
<td>Number of flashing chambers</td>
<td>16</td>
</tr>
</tbody>
</table>
Figure A.1  Aerial view of the plant during construction (10 trains)

Figure A.2  Inner view of a train of flashing chambers
Figure A.3  Deck along a train showing the doors of the flashing chambers

Figure A.4  Brine heater at the head of a train
Figure A.5  Vacuum deaerator (spraying tower under vacuum to decrease oxygen content down)

Figure A.6  Steam-jet ejectors for vacuum
Figure A.7  Vertical pumps after the last effect for brine recirculation

Figure A.8  Inside the first flashing chamber: demister, walls, feed brine orifices and bottom ground
**Figure A.9** Intermediate flashing chamber (detail of the guillotine set to adjust the pressure drop between the chambers)

**Figure A.10** Inlet orifice gates of an intermediate flashing chamber (the pen gives the scale – circled in red)
Figure A.11  Last flashing chamber (two vertical pipes for make-up flow)

Figure A.12  Last flashing chamber (the well for brine exhaust/recycling)
APPENDIX B  FLASHING CHAMBER IN AL KHOBAR MSF PLANT (JEDDAH PHASE II)

In this appendix, an example of a highly corroded flashing chamber in Al Khobar MSF plant is presented. The objective of this appendix is to give the reader an actual view of the effect of the high brine temperature on the flashing chamber walls.
APPENDIX B  

Flashing chamber in Al Khobar MSF plant (Jeddah Phase II) [126]

Table B.1  Al Khobar MSF Plant (Jeddah Phase-II)

<table>
<thead>
<tr>
<th>Variables</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Capacity</td>
<td>2.5MIGD</td>
</tr>
<tr>
<td>Number of stages</td>
<td>34</td>
</tr>
<tr>
<td>Top Brine Temperature (design)</td>
<td>121°C</td>
</tr>
<tr>
<td>Top Brine Temperature (actual)</td>
<td>115°C</td>
</tr>
</tbody>
</table>

Highly corroded flash chambers due to high brine temperature

Figure B.1  Flash chambers upper  

Figure B.2  Flash chambers walls
APPENDIX C   FLASHING OF WATER INSIDE A FLASHING CHAMBER

In this appendix, some images are taken from an experimental work by Lior [186] investigating the flashing process inside a flashing chamber without any flow baffles. The flashing chamber is scaled down into (7.8 cm width and 113 cm length) with glass windows for visual observation. The images perform a visual description of the brine flow during the flashing process (Source data: Lior, 1973 [186]).
Figure C.1  Lior’s flashing chamber and instruments (Fig.2, [186])
Figure C.2 Flasing chamber assembly drawing – front view (Fig.3, [186])
Figure C.3 Flow patterns inside Lior's flashing chamber (Fig. 11, [186]).
Figure C.4  Flashing flow with temperature distribution inside Lior's flashing chamber, Runs A, B, & C (Fig. 12, [186])
Figure C.5: Flashing flow with temperature distribution inside Lior's flashing chamber, Runs A, B, & C (Fig.13, [186])
Figure C.6 Flashing flow with temperature distribution inside Lior’s flashing chamber, Runs A, B, & C (Fig. 14, [186])
APPENDIX D  OVERVIEW OF SOME FLUENT BASICS

In this Appendix, we highlight some of FLUENT’s basics related to our computational model. An overview of the solution methodology with both of phase change mechanisms’ implementations, and the pressure definition employed in FLUENT are presented.
FLUENT 14.5 [263] uses a finite volume method to solve a series of conservation equations, which are required to solve transport equations of the flow over the predicted field.

Implicit solver [263, 291, 309] based within the pressure velocity coupling is used to solve equations in the predicted field. The following flowcharts describe solution steps and methodology. The pressure-implicit with splitting of operators, PISO, is applied to the VOF model spatial discretisation. Both the neighbour and skewness correction are used to improve the efficiency of the PISO algorithm calculation [309]. FLUENT uses internal data structures to define domains of the mesh, assign an order to cells, cell faces, and nodes in a mesh, and establish connectivity between adjacent cells (see Figure D.1). Figures D.2, & D.3 show the model solution methodology.

The solution methodology for phase change mechanisms (thermal or mechanical effect) is as follows. Both mechanisms apply throughout the field (together). Based on the local conditions, it considers whether (thermal or mechanical effect) one or both will be activated. The local conditions for the phase change are checked for every iteration and for every cell throughout the field (see Figure D.4).

All the pressures that are specified in FLUENT are gauge pressures, \( p'_g \), and they are called as static pressures in FLUENT (see Figure D.5). The operation pressure is the reference pressure for each cell (cell centre). When the gravitational acceleration is active in FLUENT, the pressure field is included the hydrostatic head. The predicted maps show the total pressure.

\[
p_{\text{absolute}} = p_{\text{op}} + p'_g \\
p_{\text{total}} = p_{\text{absolute}} + p_{\text{dynamic}}
\]
Figure D.1 The scalar control volume used for the discretisation of the continuity equation (adapted from [263, 309, 276]).

Figure D.2 Flowchart for the steps of the implicit pressure-based method (adapted from [263, 309]).
Figure D.3 Flowchart for the steps of the pressure velocity coupling method
(adapted from [263, 309]).

Figure D.4 Illustration of absolute, gauge, and vacuum pressure readings in FLUENT (adapted from [263])
APPENDIX E  MESH REFINEMENT TEST

**Figure E.3** Temperature distribution along horizontal traverses at $y = 0.03\text{m}, 0.17\text{m}, \text{free surface}, 0.3\text{m}$ for various mesh size; (a) coarse,(b) medium,(c) fine.
APPENDIX F  VALIDATION OF THE ZWART ET AL VAPOURISATION-CONDENSATION MODEL

In this Appendix, the Zwart et al vapourisation-condensation model within the Fluent VOF code is validated by published experimental data [310, 312-315] under non-equilibrium (flashing) conditions inside a converging-diverging nozzle.

Our model is applied to Abuaf et al.’s experiment [310] under two different conditions: under single phase flow and non-equilibrium (flashing) flow conditions.

The validation is evaluated by comparing the pressure distribution inside the converging-diverging nozzle under both the single phase flow and the flashing flow conditions. For the flashing flow condition, the void fraction distribution is also compared.
F.1. Abuaf *et al* Experiment – Overview

A steady isothermal flashing water flow in an axisymmetric converging-diverging nozzle was run by Abuaf *et al.* (1981) [310]. The main purpose of the experiment was to measure the net vapour generation rate under non-equilibrium (flashing) conditions. *Figure F.1* shows a schematic drawing of the experimental system arrangement.

![Schematic of the experimental system including the test section (Converging-Diverging Nozzle) [314].](image)

*Figure F.1* Schematic of the experimental system including the test section (Converging-Diverging Nozzle) [314].
The overall length of the test section is 787 mm, an axisymmetric converging-diverging nozzle forming part of that section. Its length is 559 mm, with 51 mm diameter at the inlet and 25 mm diameter at the throat, with a total open angle of 5.2°. High purity water is circulated throughout the system.

Abuaf et al.’s experiment was divided into two sections. The first section dealt with a single phase flow. Several flowrates were tested, and the pressure distribution along the converging-diverging nozzle (testing section) was recorded for each flowrate. The second section dealt with multiphase flow under flashing conditions (water changes from liquid to vapour, due to a reduction in pressure).

F.1.1. Flow loop inside the experimental system [310]

The water was pumped vertically upwards to the test section using a centrifugal pump (see Figure F.1). The flow rate was controlled using a 4” ball valve and two Omnillectric Dahl control valves. The flow rate was measured by a Cox industrial turbine flow meter. The water was heated by a heater system located just before the test section. The maximum power of the heater system was 520 kW. A pressurizer is used to fix the inlet pressure to the test section. The pressurizer can be isolated from the system and then the pressure in the system can be controlled by the pump flow rate. Thus, two operating modes can be achieved: flow controlled or pressure controlled. After the fluid passed the test section, it entered a condensation tank to condense the vapour. The pressure in this tank is the same as the inlet pressure in the exit of the test section.

During the single phase test, the experiment was operated under a flow controlled mode, while during the flashing condition it was operated under a pressure controlled mode.
F.1.2. Experimental measurements, measurement instruments and methods

Several flowrates were tested, and both pressure distribution and vapour void fraction were investigated and recorded for each flowrate. The range of operating conditions used for single-phase and multi-phase flow are summarised in Table F.1.

Table F.1. Operating conditions of the Abuaf nozzle experiment

<table>
<thead>
<tr>
<th></th>
<th>Range value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Test section inlet pressure</td>
<td>100 – 1000 kPa</td>
</tr>
<tr>
<td>Test section inlet temperature</td>
<td>20 – 150 °C</td>
</tr>
<tr>
<td>Mass flux</td>
<td>1.0 – 7.8 Mg/m² sec</td>
</tr>
<tr>
<td>Test section inlet Reynolds number</td>
<td>10⁵ - 10⁶</td>
</tr>
</tbody>
</table>

Measuring the pressure difference required wall pressure taps to be installed in the test section, with 0.4 mm in diameter. In total, 49 pressure taps were installed; the first three (taps 1, 2, and 3) were located at the constant area entrance upstream of the converging section. The last three (taps 47, 48, and 49) were located at the constant area exit downstream of the diverging section. At the throat point, tap 25 was located. The recorded measurements in each tap were an average of the four measurements around the wall pressure tap (summation of four reading divided by four).

For the second part of the experiment, a single beam gamma densitometer was added to the system, whose function was to measure the centreline diametrical averaged void fraction. Resistance temperature detectors (RTD) were applied to measure the inlet and condensing tank temperatures. The measurement uncertainties in the experiments are summarised in Table F.2.
**Table F.2.** Measurement uncertainties in the Abuaf nozzle experiments [310]

<table>
<thead>
<tr>
<th>Variable</th>
<th>Type of sensor</th>
<th>Range value</th>
<th>Uncertainties</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature</td>
<td>Resistance temperature detector (RTD)</td>
<td>-200 – 500 °C</td>
<td>1.2 %</td>
</tr>
<tr>
<td>Pressure difference</td>
<td>Strain gage transducer</td>
<td>4 – 500 kPa</td>
<td>1 %</td>
</tr>
<tr>
<td>Flow rate</td>
<td>Turbine meter</td>
<td>3 – 950 l/min</td>
<td>0.5 %</td>
</tr>
<tr>
<td>Void fraction</td>
<td>Gama densitometer</td>
<td>0 - 1</td>
<td>5 %</td>
</tr>
</tbody>
</table>

**F.2. Validation cases**

The geometry and dimensions of the test section in the experiment can be seen in Figure F.2. The operating conditions for both single phase and multiphase flow under flashing are:

**Case 1:** for a single phase flow (experiment number 9):
1. Inlet temperature 300.25 K (± 3.6 K)
2. Inlet pressure 709 kPa (± 7 kPa)
3. Condenser tank pressure (exit pressure) 674 kPa
4. Average inlet velocity 3.127 m/s (± 0.016)

**Case 2:** under flashing condition (experiment number 358):
1. Inlet temperature 373.25 K
2. Inlet pressure 370.2 kPa
3. Condenser tank pressure (exit pressure) 114.8 kPa
4. Average inlet velocity 5.97 m/s. (± 0.030)
5. Area-averaged outlet void fraction 0.54.

Uniform inlet velocity profile and temperature of the water are used to define inputs to the computational model, applied as boundary conditions, while for validation the vapour production and pressure field are computed and compared with measured values [310].
Figure F.2  Computational domain and boundary conditions used for simulating Abuaf experiments: (a) single-phase flow case, (b) flashing flow case (source data [310])
F.3. Computational Modelling Setup

In this section, computational domain and mesh, boundary conditions, numerical schemes and convergence criteria are presented for both a single phase flow and a multi-phase flow under flashing condition inside a converging-diverging nozzle.

F.3.1. Case 1: Converging-diverging nozzle single phase flow

The single phase flow simulation is used to determine a suitable mesh resolution by testing several meshes of varying initial resolution to ensure that results were independent of the mesh spacing. Five runs have been tested (experiment numbers [312, 310]: 6, 7, 8, 9, and 10). Here, experiment number 6 is used for mesh resolution, and experiment number 9 is used to compare the measurement data with our computational predictions.

For predicting the fluid flow in the converging-diverging nozzle, we treat the flow as incompressible, two-dimensional axisymmetric, steady, adiabatic, single-phase and turbulent.

F.3.1.1. Computational domain and mesh

The dimensions of the converging-diverging nozzle are shown in Figure F.2. The flow inside the converging-diverging nozzle is wall dominated, thus the mesh is extended into the viscous sublayer and buffer layer to give $y^+ < 60$ (see Figure F.3a) along the boundary wall mesh line, therefore enhanced wall functions are used with the $k$, $\epsilon$ model to better resolve the flows in these near-wall regions (by considering the effect of the buffer and sublayer regions).
Quadrilateral rectangular shape elements with 6 inflation layers are used for the mesh generation. The total number of elements is 2,608 elements. Figure F.3b shows a typical computational domain mesh representing in the flow domain inside the converging-diverging nozzle.

![Graph showing y+ distribution along the boundary wall of the converging-diverging nozzle](image)

*Figure F.3a*  $y^+$ distribution along the boundary wall of the converging-diverging nozzle

![Mesh inside the converging-diverging nozzle at the throat](image)

*Figure F.3b*  Typical near wall domain mesh inside the converging-diverging nozzle at the throat

### F.3.1.2. Boundary conditions

The boundary conditions applied are as shown in Figure F.2

The required values of water properties at temperature of $300.25 \text{ K}$ are as follows: $\rho_l = 996.59 \text{ kg/m}^3$ and $\mu = 0.852 \times 10^{-3} \text{ kg/m-s}$. 

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The convergence criteria (see Figure D.4) are as follows: for each of the $x$-velocity, $y$-velocity, and $k, \varepsilon$ it is $10^{-5}$, for the continuity equation it is $10^{-4}$ and for the energy equation it is $10^{-6}$.

![Example plot of residuals for the single-phase case](image)

**Figure F.4** Example plot of residuals for the single-phase case

F.3.1.3. Numerical schemes and convergence criteria

Implicit solver based within the pressure-velocity coupling is used for solving equations in the prediction field. First-order upwind differencing is applied for the spatial discretisation of the convective terms in the momentum, turbulence variables and energy equations.

The simulation was run on an Intel Core2 quad-CPU Q6600 of 2.4 GHz. Single cores of the available four are used to run this computational model. The simulation converges after 600 iterations.
F.3.2. Case 2: Converging-diverging nozzle for multi-phase flow under flashing condition.

No changes have made to the computational domain or to the mesh element numbers or shapes, which remain the same as case 1 (see § F.3.1.1.).

F.3.2.1. Boundary conditions

The boundary conditions are applied as shown in Figure F.2.b

The required values of water properties at temperature 373.25 K are as follow: $\rho_l = 958.05$ kg/m$^3$ and $\mu = 0.282 \times 10^{-3}$ kg/m-s.

F.3.2.3. Numerical schemes and convergence criteria

Implicit pressure based within the pressure-velocity coupling is used to solve equations in the prediction field. The pressure-implicit with splitting of operator algorithm, PISO, is applied to the VOF model spatial discretisation. Both the neighbour and skewness correction are used to improve the efficiency of the PISO algorithm calculation. Second-order upwind differencing is applied for the spatial discretisation of the convective terms in the momentum, volume fraction, turbulence variables and energy equations. During each iteration, for each cell the mass-transfer and energy interaction between the two phases are calculated, then returned as source terms to the particular transport equations.

The convergence criteria are as follows: for each of the $x$-velocity, $y$-velocity, $k$, $\varepsilon$ and volume-fraction-vapour equations it is $10^{-5}$, for the continuity equation it is $10^{-4}$ and for the energy equation it is $10^{-6}$.

The simulation was run on an Intel Core2 quad-CPU Q6600 of 2.4 GHz. Single cores of the available four are used to run this computational model. The simulation converges after 1900 iterations (3-4 hrs).
F.4. Validation

Our computational model prediction results are compared with the converging-diverging nozzle for both single phase flow and flashing flow conditions as follows.

F.4.1. Case 1: converging-diverging nozzle single phase flow

In this section, an evaluation of our computational model mesh resolution by comparing the pressure distribution inside the converging-diverging nozzle, is presented.

F.4.1.1. Mesh resolution

Several meshes of varying initial resolution were tested to ensure that results were independent of the mesh spacing. The cell sizes investigated were coarse, medium and fine size. Each mesh was tested using the flashing process model with the same boundary and initial conditions. Predicted values of pertinent variables were extracted along the nozzle axis, and plotted for comparison. Abuaf et al.’s experiment (run number 6) [310] is used for this purpose. **Figure F.5a.** shows the pressure distribution of different mesh sizes along the converging-diverging nozzle axis. The pressure distribution for all mesh size cases (fine, medium and coarse) seems identical, as shown in **Figure F.5a.** For showing the difference between the cases (see **Figure F.5b**) we defined the predicted pressure values of the fine mesh as reference value, and then we compared the predicted pressure values of the medium mesh with it, the same we did for the coarse mesh case. We found the maximum pressure difference value range is [− 0.048, + 0.032] Pa. Thus, the results are effectively independent of the mesh spacing. To produce the most accurate results, the fine mesh size has been used in all of the computational study cases.
Figure F.5a  Comparing predicted pressure for different mesh sizes along the nozzle axis ([310], experiment number 6)
Figure F.5b Predicted distributions along nozzle of differences in pressure between coarse, medium and fine meshes (exp. 6, [310])
F.4.1.2. Pressure distribution validation

Results of experimental data and CFD predictions are plotted in Figure F.6. The values of the pressure are taken along an axial traverse located 0.2 mm from the nozzle centreline axis.

F.4.2. Case 2: converging-diverging nozzle for multi-phase flow under flashing condition

The prediction results of the multi-phase flow under a flashing condition are compared with Abuaf et al.’s experiment data (run number 358) [310]. The comparison validated by checking the pressure distribution and vapour void fraction inside the converging-diverging nozzle under flashing condition.

Figure F.7 shows a pressure distribution comparison between Abuaf et al.’s experimental data (run number 358) and our computational predictions. The values of the pressure for both cases are taken along the axial centre.

Figure F.8 shows a vapour void fraction comparison between Abuaf et al.’s experimental data (run number 358) and our computational predictions. The values of the vapour void fraction are area averaged.

As shown in Figures F.6, F7, and F8, the results show good agreement between predicted and measured results. The measured average vapour volume fraction at the outlet of the nozzle was 0.544, for which case our predicted value is 0.543.
Figure F.6  Converging-diverging nozzle, signal-phase flow ([310], experiment 9): comparing of predicted and measured pressure distributions.
Figure F.7 Comparing results axial pressure distribution of experimental and CFD under flashing condition ([310], experiment number 358)
Figure F.8  (a) CFD prediction of vapour void fraction inside the converging-diverging nozzle (b) comparing results of area-averaged void fraction of experimental and CFD under flashing condition ([310], experiment number 358)
F.5. Summary and Discussion

A validation of our developed numerical procedure for flashing flow was presented. Our model was multiphase, based on the FLUENT VOF code implementation, and encapsulates an interaction mechanism for phase change — based on vapour pressure — that arises during the flashing process inside a converging-diverging nozzle. The results show a good agreement between the predicted pressure distributions and vapour void fraction with the measured data.

The model was tested for three different mesh sizes to ensure mesh independence. The fine mesh size was used in all study cases to ensure sufficient resolution.

Two cases were used for validating our computational model. Some points should be discussed for each case.

The measurement data was only available in graphical form, and that is why we used WebPlotdigitizer software to obtain the numerical values of the pressure values and the vapour void fraction from the experimental work. The graphical form was plotted with big gradation (scales) (e.g. all the experimental graphs of the pressure were in kPa, with 100 kPa as a scale step). The range of uncertainty can be defined based on the thickness of WebPlotdigitizer marker, and it is ±10kPa. Thus, the use of WebPlotdigitizer software for obtaining the experimental data is considered as a source of error.

For multi-phase flow under flashing condition:

- The phase change mechanism and the vapour void fraction can be seen in Figure F.8. It is shown that flashing occurred at a location just upstream of the throat.
- The predicted pressure drop distribution after the throat (when the phase change occurs) is slightly larger than the experimental values (*Figure F.7*). This happens since this experimental case runs under critical flow* conditions [298], but the model was run just downstream of the critical point. Running just downstream of the critical point increased the phase change rate, thus increased the predicted vapour void fraction over the experimental measurements.

* Critical flashing flow is that flow which no longer depends on small changes in downstream conditions. The critical velocity in two-phase flow is difficult to predict, since the rapid expansion of the fluid may induce both mechanical and thermal non-equilibrium between the two phases [316, 317].
APPENDIX G  EFFECTS OF THE INLET VELOCITY PROFILE

In this Appendix, an adapted user-defined function for defining parabolic and skew-shear inlet velocity profiles to the flashing chamber are presented. The effects of these velocity profiles on the steady state thermo-fluid behaviour are presented in terms of flow patterns, free surface shape and level, thermal performance, vapour void fraction, and mass transfer rate.

The UDFs, which were written in the C language, are presented at the beginning of the following section. Then the thermo-fluid predicted results of uniform, parabolic and skew-shear inlet velocity profiles for finite flashing case; \( v_x = 1.5 \) m/s, \( p_{op} = 1.4 \) bar, \( T_{in} = 383.15 \) K. are provided. The profiles are all unidirectional along the x-axis, and scaled so as to give the same mass flow rate.

The available measurement data we have is the average outlet temperature of the flashing chamber. So, by checking the values of average outlet temperature at the flashing chamber exit for different velocity profile cases (as evidenced in Figures G2 – G4), it was found that the simulation result of the uniform inlet velocity profile (378.6 K) is the closest value to the real plant (379K).
APPENDIX G    Parabolic Inlet Velocity Profile

Mean Inlet Velocity = 1.5 m/s  
Saturation Temperature = 382.32 K  
Orifice Height = 0.06 m

\[ u_x(y) = u_{max} - u_{max} \left( \frac{y}{h} \right)^2 \]

Flashing chamber inlet velocity profile.c
UDF for specifying steady-state velocity profile boundary condition

************************************************************************/
 Flashed chamber inlet velocity profile.c
UDF for specifying steady-state velocity profile boundary condition
************************************************************************/
#include "udf.h"

DEFINE_PROFILE(inlet_x_velocity, thread, position)
{
    real x[ND_ND];    /* this will hold the position vector */
    real y, h;
    face_t f;
    h = 0.06; /* inlet height in m */
    begin_f_loop(f, thread)
    {
        F_CENTROID(x, f,thread);
        y = 2.*(x[1]-0.5*h)/h; /* non-dimensional y coordinate */
        F_PROFILE(f, thread, position) = 1.5*(1.0-y*y); /* this is the equation which describes the profile velocity at the orifice of the flashing chamber */
    }
    end_f_loop(f, thread)
}
Skew-shear Inlet Velocity Profile

Mean Inlet Velocity = 1.5 m/s  
Saturation Temperature = 382.32 K  
Orifice Height = 0.06 m

\[ u_x(y) = u_{avg} - \left(1 - \left(\frac{y}{h}\right)^2\right) \]

The velocity increases from the wall to a maximum in the main stream of the flow. Mean inlet velocity is 1.5 m/s

Figure G.1  Screenshot of the UDF written by C language
<table>
<thead>
<tr>
<th>Inlet velocity profile</th>
<th>Inlet velocity profile</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Uniform</strong></td>
<td>![Uniform Velocity Profile]</td>
</tr>
<tr>
<td><strong>Parabolic</strong></td>
<td>![Parabolic Velocity Profile]</td>
</tr>
<tr>
<td><strong>Skew-shear</strong></td>
<td>![Skew-shear Velocity Profile]</td>
</tr>
</tbody>
</table>

**Figure G.2** Inlet velocity profiles

\[
\overline{v_x} = 1.5 \text{ m/s}
\]
Figure G.3  Predicted particle paths results for different inlet velocity profile of uniform, parabolic and skew-shear ($\bar{v_x} = 1.5$ m/s, $p_{op} = 1.4$ bar, $T_{in} = 383.15$ K)
Figure G.4  Predicted temperature results for different inlet velocity profile of uniform, parabolic and skew-shear ($\bar{v}_x = 1.5$ m/s, $p_{op} = 1.4$ bar, $T_{in} = 383.15$ K)

<table>
<thead>
<tr>
<th>Inlet velocity profile</th>
<th>Predicted temperature field</th>
</tr>
</thead>
<tbody>
<tr>
<td>Uniform</td>
<td><img src="image1" alt="Uniform Temperature Field" /></td>
</tr>
<tr>
<td>Parabolic</td>
<td><img src="image2" alt="Parabolic Temperature Field" /></td>
</tr>
<tr>
<td>Skew-shear</td>
<td><img src="image3" alt="Skew-shear Temperature Field" /></td>
</tr>
</tbody>
</table>

- $\bar{v}_x = 1.5$ m/s
- $p_{op} = 1.4$ bar
- $T_{in} = 383.15$ K
Figure G.5  Predicted temperature distributions along horizontal traverses at $y = 0.03\text{m}, 0.17\text{m}, 0.3\text{m}$ for different inlet velocity profiles: (a) uniform, (b) parabolic and (c) skew-shear ($\bar{v}_x = 1.5 \text{ m/s}, p_{op} = 1.4 \text{ bar}, T_{in} = 383.15 \text{ K}$)
Figure G.6  Predicted vapour void fraction, free surface shape and level, and bubble nucleation region results for different inlet velocity profile of uniform, parabolic and skew-shear ($\bar{v}_x = 1.5$ m/s, $p_{op} = 1.4$ bar, $T_{in} = 383.15$ K)

<table>
<thead>
<tr>
<th>Uniform</th>
<th>Parabolic</th>
<th>Skew-shear</th>
</tr>
</thead>
</table>

- **Uniform**
- **Parabolic**
- **Skew-shear**

\[
\begin{align*}
\bar{v}_x &= 1.5 \text{ m/s} \\
p_{op} &= 1.4 \text{ bar} \\
T_{in} &= 383.15 \text{ K}
\end{align*}
\]
Figure G.7  Predicted mass transfer rate results for different inlet velocity profile of uniform, parabolic and skew-shear ($\bar{v}_x = 1.5$ m/s, $p_{op} = 1.4$ bar, $T_{in} = 383.15$ K)
In this Appendix, a time-dependent prediction is developed for the finite flashing case; \( v_x = 1.5 \text{ m/s}, p_{op} = 1.4 \text{ bar}, T_{in} = 383.15 \text{ K} \). The predicted results of the thermo-fluid performance (flow patterns, thermal performance, and vapour volume fraction) inside the flashing chamber are presented (see Table 1, 2, and 3). It should be noted that the boundary conditions are steady, but the flow is intrinsically unsteady. The computational model's time step is 0.001 sec, while the overall duration is 2.16 sec. The transient solution is applied to study the residence time effect on the thermo-fluid behaviour inside the flashing chamber.

A comparison between the finite flashing case (\( v_x = 1.5 \text{ m/s}, p_{op} = 1.4 \text{ bar}, T_{in} = 383.15 \text{ K} \)) steady state and the transient solution is shown in Figure H.1. It is evident that the outlet temperature increases with time.

Predicted temperature fields in the flashing chamber at time: 0.001, 0.5, 1, 1.5, 2, and 2.15 sec, are shown in Figure H.2.

Figure H.3 shows the NETD as a function of time. It is noticed that the NETD is increased with the time. We found the following mathematical function to describe the changes of the NETD with specific range of time:

\[
NETD = 0.32t + 1.4979 \quad \text{for} \quad 0.5 \text{ sec} < t < 2 \text{ sec}
\]
**APPENDIX H  Transient Solution**

*Table H.1* Predicted particle paths fields at discrete times

<table>
<thead>
<tr>
<th>Time (t)</th>
<th>Predicted particle paths (streamlines)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.001 sec</td>
<td><img src="image" alt="Streamline Image" /></td>
</tr>
<tr>
<td>0.5 sec</td>
<td><img src="image" alt="Streamline Image" /></td>
</tr>
<tr>
<td>1 sec</td>
<td><img src="image" alt="Streamline Image" /></td>
</tr>
<tr>
<td>1.5 sec</td>
<td><img src="image" alt="Streamline Image" /></td>
</tr>
<tr>
<td>2 sec</td>
<td><img src="image" alt="Streamline Image" /></td>
</tr>
<tr>
<td>2.16 sec</td>
<td><img src="image" alt="Streamline Image" /></td>
</tr>
</tbody>
</table>

\[ \frac{V_x}{x} = 1.5 \text{ m/s} \]

\[ p_{op} = 1.4 \text{ bar} \]

\[ T_{in} = 383.15 \text{ K} \]
Table H.2  Predicted temperature fields at discrete times

<table>
<thead>
<tr>
<th>Time (t)</th>
<th>Predicted temperature field</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.001 sec</td>
<td>![Image]</td>
</tr>
<tr>
<td>0.5 sec</td>
<td>![Image]</td>
</tr>
<tr>
<td>1 sec</td>
<td>![Image]</td>
</tr>
<tr>
<td>1.5 sec</td>
<td>![Image]</td>
</tr>
<tr>
<td>2 sec</td>
<td>![Image]</td>
</tr>
<tr>
<td>2.16 sec</td>
<td>![Image]</td>
</tr>
</tbody>
</table>

\[\overline{v_x} = 1.5 \text{ m/s} \]
\[p_{op} = 1.4 \text{ bar} \]
\[T_{in} = 383.15 \text{ K} \]
Table H.3  Predicted vapour void fraction fields at discrete times with two different ranges of scale

<table>
<thead>
<tr>
<th>Time (t)</th>
<th>Predicted Vapour volume fraction</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.001 sec</td>
<td><img src="image0.png" alt="Image" /></td>
</tr>
<tr>
<td>0.5 sec</td>
<td><img src="image1.png" alt="Image" /></td>
</tr>
<tr>
<td>1 sec</td>
<td><img src="image2.png" alt="Image" /></td>
</tr>
<tr>
<td>1.5 sec</td>
<td><img src="image3.png" alt="Image" /></td>
</tr>
<tr>
<td>2 sec</td>
<td><img src="image4.png" alt="Image" /></td>
</tr>
<tr>
<td>2.16 sec</td>
<td><img src="image5.png" alt="Image" /></td>
</tr>
</tbody>
</table>

\[
\bar{u_x} = 1.5 \text{ m/s} \\
p_{op} = 1.4 \text{ bar} \\
T_{in} = 383.15 \text{ K}
\]
Figure H.1. Predicted outlet temperature distributions along horizontal traverses at $y = 0.3$ m at time (0.001, 0.5, 1, 1.5, 2, and 2.15) sec and for finite flashing steady state; $v_x = 1.5$ m/s, $p_{op} = 1.4$ bar, $T_{in} = 383.15$ K

Figure H.2. Predicted temperature distributions along horizontal traverses at $y = 0.3$ m at different times
Figure H.3. \( \text{NETD} \) as a function of time
APPENDIX I 3D TRANSIENT SOLUTION

I.1 Geometry and Boundary Conditions

Figure I.1  Computational domain: evaporation zone dimensions and boundary conditions

Figure I.2  Predicted vapour void fraction fields at time = 0 sec
### I.2 Predicted Results

*Table I.1* Predicted particle paths fields at discrete times

<table>
<thead>
<tr>
<th>Time (t)</th>
<th>Predicted particle paths</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.1 sec</td>
<td><img src="image1.png" alt="Image" /></td>
</tr>
<tr>
<td>7 sec</td>
<td><img src="image2.png" alt="Image" /></td>
</tr>
<tr>
<td>20 sec</td>
<td><img src="image3.png" alt="Image" /></td>
</tr>
</tbody>
</table>
Journal Papers

Conference Papers

Poster Presentations

Papers in Progress
Nigim, T. H., Eaton, J. A. Finite and infinit flashing processes in a MSF desalination stage: CFD study. *Desalination*. (Chapter 5)
REFERENCES

55. German Climate Portal, (access: 31-03-2014), URL: http://www.deutschesklimaportal.de/EN/Topics/1_Klimawandel_Ueberblick/thema2_node.html.
95. State Energy Conservation Office (SECO), Austin, Texas USA (access: 30-06-2016) URL:http://www.seco.cpa.state.tx.us/publications/renewenergy/solarenergy.php


397


203. Miyatake, O., Hashimoto, T. & Lior, N. The liquid flow in multi-stage flash


280. Dupont, JB., & Legendre, D. Numerical simulation of static and sliding drop


299. Okawa, T., Yoneda, K., & Yoshioka, Y. New interfacial drag force model


314. Wu, B.J.C., Saha, P. Abuaf, N., & Jones, O.C. A one-dimensional model of


