Modelling the absorption of binary mixtures in plate heat exchangers
Predictions for Ocean Thermal Energy Conversion application

M.P. Nuijten
MODELLING THE ABSORPTION OF BINARY MIXTURES IN PLATE HEAT EXCHANGERS

PREDICTIONS FOR OCEAN THERMAL ENERGY CONVERSION APPLICATION

by

Menno Nuijten

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ABSTRACT

Ocean thermal energy conversion (OTEC) is currently developed as a clean and sustainable energy technology for offshore application in tropical areas. The technology shows promising possibilities in combination with the Kalina cycle, a thermodynamic cycle for the generation of energy from low temperature difference heat sources using a working fluid mixture. Due to the differences in boiling point between the components in a mixture and the resulting temperature glide, this allows more heat to be exchanged and thus a higher thermal effectiveness of the overall cycle.

Ammonia-water is identified to be a suitable candidate for application in the cycle, but still relatively little is understood about the performance of different components in the cycle when this mixture is used. The condenser in the cycle exchanges heat between the working fluid and cold deep ocean water. In case a mixture is used, the process in this heat exchanger is described as absorption. In this study, an attempt is made to model the relevant hydrodynamics and heat and mass transfer processes in the heat exchanger, given typical operating conditions and a specified heat exchanger geometry. A plate heat exchanger is used for the absorber, due to specific advantages of this type of heat exchanger.

A review of the used thermodynamic cycle, working fluids and their properties, and the heat exchanger type is included in this study. Also, literature on the hydrodynamics, pressure drop, heat transfer and mass transfer is discussed. A review of modelling approaches is provided before an extensive description of the implemented model is given.

The model is stable, the predicted parameters converge to constant values during the iterative procedure, and the predictions show reasonable agreement with other work. The heat transfer may be overestimated as a result of the assumed hydrodynamics in the model. Besides, jumps in parameter profiles result from full condensation and the corresponding change in flow pattern. The modelling approach shows promise to be extended to make predictions for different operating conditions, working fluids and heat exchanger geometries. In the future, the model could be developed into a tool for optimization of the absorber performance.

A preliminary experiment was performed for validation of the model. Comparison of the numerical results with the experiment and previously developed numerical models points out that the prediction of the model for the heat exchanger performance is promising. More experiments of sufficient accuracy and reliability are required for a better validation of the model, and to allow predictions of the heat exchanger performance for a wider range of conditions.
First of all, I would like to thank Carlos Infante Ferreira for support help as daily supervisor of the project. Our weekly meetings were always helpful, and he never minded them to take a bit more time than originally planned. His technical expertise and focus on achievable results allowed me to perform interesting and relevant work, whilst also finishing my work within the original time schedule.

My gratitude goes out to Thijs Vlugt for his interest in the progress of my work during the update meetings. Also, for giving me the confidence to start the project at the beginning, and working towards the finish at the end. I want to thank Freek Kapteijn for volunteering to jump on board of my thesis committee, and his contribution to the mid-term meeting.

I would also like to express my appreciation to the people at Bluerise, mainly Joost Kirkenier and Berend Jan Kleute, for inspiring me to work on this topic and helping out with many practical aspects of the work. Especially during the final weeks, where the first experiment was finally conducted after months of running into obstacles. My gratitude goes out to Bram and Alex for their contribution during this period, and to Vilborg and Bob for the useful collaboration in the earlier stages of my project.

Finally, I would like to thank the other students at the Process and Energy department for providing welcome distraction from the work during office hours, and my friends and family for the many activities I enjoyed outside of working times.
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**NOMENCLATURE**

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### Greek Symbols

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Two-phase multiplier

Chevron or corrugation angle

Lockhart-Martinelli parameter

Prandtl number function

Acentric factor

Corrugation parameter/wave number

**Dimensionless numbers**

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film  Film
flow  (To) flow direction
FC  Full condensation
g  Gap
G  Gas
h  Hydraulic
hor  Horizontal
hw  Hot water
HX  Heat exchanger
i  Interface
i  Number within a set
in  Inlet
instr  Instrumentation
it  Iteration
lam  Laminar
L  Liquid
m  Mean
max  Maximum
meas  Measured
min  Minimum
mix  Mixture
out  Outlet
o  Outer wall
p  Plate
pf  Pure fluid
port  Port
rad  Radians
red  Reduced
rep  Repeatability
sat  Saturated
sep  Separator
subc  Subcooled
sup  Superficial
top  Top
tot  Total
tur  Turbulent
TP  Two-phase
vap  Vaporization
ver  Vertical
V  Vapor
W  Water
wf  Working fluid
w  Wall

**Superscripts**

*  Relative ratio
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<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>BEM</td>
<td>Boundary element method</td>
</tr>
<tr>
<td>BPHE</td>
<td>Brazed plate heat exchanger</td>
</tr>
<tr>
<td>CFD</td>
<td>Computational Fluid Dynamics</td>
</tr>
<tr>
<td>COP</td>
<td>Coefficient of performance</td>
</tr>
<tr>
<td>EOS</td>
<td>Equation of state</td>
</tr>
<tr>
<td>GMTD</td>
<td>Generalized mean temperature difference</td>
</tr>
<tr>
<td>HC</td>
<td>Hydrocarbon</td>
</tr>
<tr>
<td>HCFC</td>
<td>Hydrochlorofluorocarbon</td>
</tr>
<tr>
<td>HFC</td>
<td>Hydrofluorocarbon</td>
</tr>
<tr>
<td>HFO</td>
<td>Hydrofluoroolefin</td>
</tr>
<tr>
<td>LMTD</td>
<td>Logarithmic mean temperature difference</td>
</tr>
<tr>
<td>NIST</td>
<td>National Institute of Standards in Technology</td>
</tr>
<tr>
<td>NTU</td>
<td>Number of transfer units</td>
</tr>
<tr>
<td>ORC</td>
<td>Organic Rankine Cycle</td>
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<tr>
<td>OTEC</td>
<td>Ocean Thermal Energy Conversion</td>
</tr>
<tr>
<td>PC-SAFT</td>
<td>Perturbed chain statistical associating fluid theory</td>
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<tr>
<td>PFD</td>
<td>Process Flow Diagram</td>
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<td>PHE</td>
<td>Plate heat exchanger</td>
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<tr>
<td>P&amp;ID</td>
<td>Process and Instrumentation Diagram</td>
</tr>
<tr>
<td>SEWTLE</td>
<td>Semiexplicit method for wall temperature linked equations</td>
</tr>
<tr>
<td>VDI</td>
<td>Verein Deutscher Ingenieure</td>
</tr>
<tr>
<td>WHR</td>
<td>Waste heat recovery</td>
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INTRODUCTION

1.1. OCEAN THERMAL ENERGY CONVERSION

Sustainability is an important issue in the modern world, with a shared interest wide across the society. The wish to gradually move towards a more sustainable world has become a global challenge, in which changing to a more sustainable energy world is a primary goal. In this large challenge for the future, many stakeholders play a role, such as universities, research institutes, companies, governments and all consumers of energy. They contribute to the challenge in different ways, such as by inventing, developing and regulating the energy alternatives.

When it comes to research in developing alternatives for conventional sources of energy, such as nuclear, coals, oil and gas, a great amount of work has been done on countless different technologies. Some of these have found considerable application already, such as biofuels, hydropower, wind energy, solar energy and geothermal energy. All technologies are continuously under development, and the same is true for new technologies that have yet to find commercial application. Some of these technologies may still be in infancy, where others are on the verge of commercial breakthrough. The International Panel for Climate Change (IPCC) published a report in which developments in renewable energy are reviewed and projections are made for the future (Edenhofer et al., 2012 [1]). One of the technologies in a further stage of development is Ocean Thermal Energy Conversion (OTEC), of which the concept was first developed in the late 19th century. In the years since the first ideas arose to develop the technology, interest in its use has fluctuated. In recent years, interest has been at a high level, and different companies, universities and research institutes are actively working on the development and implementation of the technology. In the Netherlands, the company Bluerise is working together with Delft University of Technology to improve the technology through scientific research.

OTEC uses the temperature difference between warm surface seawater and cold deep seawater to supply electrical power. Bluerise, active in developing OTEC technology for a potential worldwide market, published a report (Kleute and Vroom (2014) [2]) in which the most important factors for successful OTEC application are pointed out. The main requirement is a sufficiently large temperature difference, of more than 20°C, between water at the surface and the deep ocean water. The most promising locations for application of OTEC are tropical regions, ideally with a steep coastline and a demand for the produced energy near the coast. This encompasses a large area of the globe, and at the moment projects
are set up in locations such as Sri Lanka and the Caribbean. A representation of a 10 MW offshore OTEC installation is found in Figure 1.1. The typical temperatures of 25 – 28°C at the surface and 5 – 7°C at the bottom of the ocean are indicated, as well as the ocean depth of roughly 1 km at the location of the installation.

Figure 1.1: Representation of a 10 MW offshore OTEC installation near the coast. The offshore OTEC system is located on a platform. A large pipe is present to collect cold deep ocean water from a depth of around 1 km at a temperature of 5 – 7°C. The warm surface water is typically at a temperature of 25 – 28°C.

1.2. THE OTEC CYCLE

The OTEC technology is based upon the principle of a thermodynamic cycle containing a working fluid, exchanging heat with the environment. The environment in this case consists of the seawater, which is at much lower temperatures at the bottom of the ocean than near the surface. The working fluid in the cycle is pumped around through two heat exchangers. In the first heat exchanger, an evaporator, the working fluid exchanges heat with the surface seawater of higher temperature, allowing the working fluid to evaporate. As a result of the evaporation the volume of the working fluid increases, which drives a turbine that generates electricity through a generator. After the working fluid passes through the turbine, it is condensed in the second heat exchanger, the condenser. Condensation is made possible by exchanging heat with the cooler seawater. A schematic representation of the OTEC cycle is found in Figure 1.2.

More about the applied thermodynamic cycle is explained in Chapter 2. Discussing more about OTEC technology is outside the scope of this thesis. Information related to the topic is found in a wide selection of scientific literature, of which a fair amount is covered in Avery and Wu (1995) [4]. More recent studies include Wang and Hung (2010) [5], Sun et al. (2012) [6] and Morisaki and Ikegami (2013) [7].
Figure 1.2: Working principle of the OTEC cycle, including the most important components and typical inlet and outlet temperatures of the warm surface water and cold deep ocean water (Kirkenier, 2014 [3]).
1.3. OTEC HEAT EXCHANGERS

In the systems that are applied for OTEC, the most challenging custom-made components are the heat exchangers. For OTEC applications, at least two heat exchangers are required: an evaporator and a condenser. Little is still understood of the working principles behind the heat exchangers, and thus little is known about the predictions of the performance of these components. High performance of the heat exchangers is crucial for high effectiveness and thus energy output of OTEC systems, which makes a detailed study of the working principles of these components interesting. Of the two heat exchangers, the condenser is considered to be the most crucial and is therefore the focus of this study.

The heat exchanger type that is found to be most promising for both the evaporator and condenser is the plate heat exchanger. This type of heat exchanger has been used commercially for a longer time, but research is still ongoing into these heat exchangers. No general model for prediction of performance of the plate heat exchanger is available at present, which is valid for different fluids under varying conditions and typical geometries. More investigations are required into the use of working fluids, including binary and ternary mixtures, along with a proper prediction of their properties and the translation into heat exchanger performance. Another current topic of research is the investigation of the most suitable plate material, but this is outside the scope of this work.

A good overview of modelling, design and performance of different types of heat exchangers is found in the VDI Heat Atlas [8]. In Chapter 2 more is explained about the structure and use of plate heat exchangers, the type of heat exchangers considered in this thesis.

1.4. RESEARCH OBJECTIVES

The objective of this research is to analyze the performance of binary working fluid mixtures in plate heat exchangers functioning as absorbers. This involves numerical modelling of the relevant mass, heat and momentum transfer inside the heat exchangers. The model is validated using preliminary experimental results to assess the possibility of using the model as a tool to predict the performance of plate heat exchanger absorbers in thermodynamic cycles of OTEC and other low-grade heat source applications. In this thesis, an attempt is made to answer the following research question: ‘Can the performance of a vertical countercurrent flow plate heat exchanger functioning as absorber or condenser be predicted using a flow pattern based model, for an ammonia-water mixture under varying operating conditions and given geometry?’ The study aims to improve the process of working fluid selection by developing an improved model for the plate heat exchanger absorber or condenser. By incorporating good predictions of thermodynamics and transport properties into the model, the model should be able to predict the performance of different working fluids.

1.5. RESEARCH APPROACH

To answer the proposed research question, the following approach is followed in this study:

- Review of existing literature on two-phase flow in plate heat exchangers. This includes the use of different fluids and their properties, including mixtures. Also, phenomena occurring during phase change and flow characteristics are studied, such as the observed paths, regimes and patterns. Finally, relevant physical phenomena
related to the mass, heat and momentum transfer are reviewed.

- Review of existing literature on mathematical steady-state modelling of plate heat exchangers, including modelling approaches, assumptions and the used pressure drop, heat transfer and mass transfer correlations.

- Development and implementation of a numerical model, taking into account the findings from the literature study and the specified heat exchanger geometry, focusing on operating conditions currently used for OTEC applications.

- Collection of experimental data relevant for validation of the model, using an OTEC setup with an ammonia-water mixture as working fluid.

- Validation of the numerical model with the experiments by comparing the results, to identify points for improvement that may lead to more accurate predictions of the model.

- Discussion of the model results, drawing conclusions and proposing recommendations for future research.

1.6. THESIS OUTLINE
In Chapter 2 the theoretical background of the study is discussed, including the thermodynamic cycles used in OTEC technology, thermodynamics and transport properties of working fluids and a basic description of plate heat exchangers. Also, important studies related to heat and mass transfer and pressure drop for two-phase flow in heat exchangers are discussed, before moving into the hydrodynamics: flow regimes, paths and patterns. The final two sections in the chapter are on additional observations related to relevant physical phenomena reported in literature and modelling approaches for heat exchangers respectively.

In Chapter 3 the mathematical model is discussed, including the modelling approach, assumptions, relations used for pressure drop, heat and mass transfer. The solution procedure used for the implementation of the model is presented at the end of this chapter.

Chapter 4 is on the numerical results of the model, including the discussion of a base case and a comparison empirical models. Also, a sensitivity analysis is presented and a future outlook for the modelling and experimental work is provided. In Chapter 5 the results of the experimental work are discussed, including the setup used, the measurement devices and operating conditions, the experimental plan and an uncertainty analysis of the results. In the last section, a preliminary experimental validation of the numerical model is presented. Finally, in Chapter 6 conclusions are drawn and recommendations are made for future research.
In this chapter the thermodynamic cycles used for OTEC are first discussed in more detail in section 2.1. Next, in section 2.2 a review is given of the relevant literature related to the thermodynamic and transport properties of the fluids used in the system. In section 2.3, the plate heat exchangers are discussed and the motivation for their use regarding the application is given. A literature review of the relevant heat transfer, mass transfer and pressure drop for two-phase flow, condensation and absorption in plate heat exchangers is provided in section 2.4. In section 2.5, the flow characteristics and hydrodynamics of two-phase flow in plate heat exchangers are discussed extensively. Finally, in section 2.6 relevant physical phenomena related to phase change, momentum, heat and mass transfer are reviewed and section 2.7 covers relevant modelling approaches for plate heat exchangers in literature. The findings of the literature study are used in Chapter 3 to construct the numerical model and in Chapter 4 to allow a critical reflection on the results of the model.

2.1. THERMODYNAMIC CYCLE

A short introduction to the two thermodynamic cycle used in OTEC is given in Chapter 1. For this work, the focus will be on the Kalina cycle. This cycle is relevant for the use of partially evaporating working fluids, such as the ammonia-water mixture investigated in this study. The selection of a suitable mixed working fluid and more specifically ammonia-water is motivated further ahead in the text. The focus of this study is not on the Kalina cycle as a whole, but on a single component which in this case is the heat exchanger functioning as an absorber. Some understanding of the cycle is however required to also understand the use of the modelled component in the system. Besides, when measurements are performed using the experimental setup to obtain data relevant for the validation of the model of a single component, the setup needs to be operated as a whole.

The most relevant steps and components in the OTEC system are discussed in Chapter 1 and presented in Figure 1.2. These steps and components correspond to the Rankine Cycle or Organic Rankine Cycle (ORC). When working fluid mixtures are used, boiling and condensation occurs over temperature trajectories as opposed to a single temperature for pure fluids. This is caused by the difference in boiling point of the two components present in the mixture. The Kalina cycle thus allows the temperature of the working fluid to change during evaporation and condensation, allowing greater heat transfer than the traditional ORC [9, 10]. This heat transfer enhancement is the result of the so called ‘temperature glide’
effect, which allows the working fluid to exchange heat with the other fluid more effectively over the temperature range from inlet to outlet. The loss of exergy is reduced to increase the heat exchange efficiency. Exergy is the energy that is available to be used from the source, or transferred to the source. In general, for the Kalina cycle the heat transfer coefficients are lower than for traditional pure fluid cycles, but the heat transfer capacity is larger leading to a larger effectiveness for certain cases. The effectiveness of heat exchangers is explained in subsection 2.3.1.

The effect of the temperature glide is demonstrated in Figure 2.1, in the form of an entropy-temperature diagram. As is observed, when using a mixture the match between the working fluid and the source and sink profiles is better than when using a pure fluid. For a mixture, the slope of the temperature profile is equal to the slope of both the source and sink temperature profiles, where for a pure fluid the temperature is constant during the heat exchange process. The four major steps in the cycle are explained next:

- Isentropic pressure increase of the working fluid through the use of a pump.
- Evaporation of the working fluid under constant pressure, by heat transfer from the warm water source, represented by the upper diagonal line in Figure 2.1 for the mixture.
- Isentropic expansion of the vapor through a turbine, in order to produce power.
- Condensation of the working fluid under constant pressure, by heat transfer from the cold water source, represented by the lower diagonal line in Figure 2.1 for the mixture.

![Figure 2.1](image.png)

**Figure 2.1:** Representation of the effect of a temperature glide for the use of mixed working fluids in the OTEC cycle versus the use of pure working fluids, in a temperature-entropy diagram. The slopes of the working fluid curves match with those of the cold and hot sea water, which allows a higher effectiveness of the heat exchange process.

Two main additional components are needed for the Kalina cycle in comparison to the ORC: a recuperator and a separator. The separator separates the liquid and vapor phases.
after leaving the evaporator. The pure vapor phase is moved through the turbine, and the liquid phase is moved to the recuperator where it exchanges heat with the outlet stream of the condenser in order to slightly preheat the working fluid stream to the inlet of the evaporator. More about the specific setup used is found in Chapter 5, where the experimental work for this thesis is discussed. A schematic representation of the setup is included there. An additional advantage of the Kalina cycle is the option to adjust the mixture composition and consequently the thermophysical and transport properties of the working fluid. More about these properties is discussed in the next section, but first a review of general literature on working fluid selection is presented.

2.1.1. OTEC WORKING FLUIDS

A large number of studies is identified in the literature on the selection of working fluids for OTEC, of which a few are highlighted here. Iqbal and Starling (1976) [11] report on advantages of mixtures over pure compounds, heat transfer properties in heat exchangers. The desirable characteristics of working fluids, related to e.g. thermodynamic and transport properties, are discussed in Ganic and Wu (1980) [12]. Angelino and Colonna di Paliano (1998) [13] report on the use of organic fluid mixtures, the thermodynamic cycles and their efficiencies. It also states that multicomponent fluids may be desirable, but complex thermodynamic tools are needed to predict their properties.

Liu et al. (2004) [14] discuss the effect of working fluids on ORC performance for waste heat recovery, dividing working fluids into three categories: dry, isentropic and wet depending on the slope of the temperature-entropy curve \((dT/dS)\) to be positive, infinite or negative, respectively. Working fluids used in the field of absorption refrigeration, which may also be relevant for OTEC or other low-grade heat sources, are discussed in Cerezo et al. (2011) [15]. Besides ammonia-water (\(NH_3-H_2O\)) and water-lithium bromide (\(H_2O-LiBr\)), \(NH_3-LiNO_3\) and \(NH_3-NaSCN\) are also discussed as alternative working fluids. It is believed that replacement of conventional working fluids by mixtures of better suitable thermodynamic and thermophysical properties will lead to higher efficiency of heat and mass transfer processes. Quoilin et al. (2011) [16] report on optimization of waste heat recovery (WHR) in ORCs. Desired properties of working fluids and the screening process of these are discussed.

Morisaki and Ikegami (2013) [7] report on the performance of heat exchangers in OTEC using ammonia-water as working fluid. The physical properties of the working fluid mixture and the LMTD and GMTD method are discussed, which are used to evaluate the performance of the heat exchangers. Non-azeotropic mixtures of refrigerants are considered promising. Oronel et al. (2013) [17] discuss the use of ammonia-lithium nitrate as working fluid mixture for absorption applications, as an alternative to ammonia-water. Certain advantages and disadvantages of both working fluid mixtures are pointed out. Astolfi et al. (2014) [18] report on the selection of working fluids for ORCs, drawing conclusions about use of supercritical cycles and certain preferred properties of the fluids. Ziviani et al. (2014) [19] report on the challenges in ORC systems modelling for low grade thermal energy recovery. The two main challenges that are stated are:

1. Estimating the working fluid thermodynamic properties.
2. Determining the heat transfer rate of these fluids in evaporators and condensers for two-phase flow, considering additional factors such as geometry.
Kleute and Vroom (2014) [2] state that one of the main improvements for the use of working fluids for OTEC applications is the use of mixed working fluids instead of single component working fluids. Relevant fluid properties here are the saturation conditions, heat capacity and the thermal conductivity. Other work on the subject of working fluid selection for ORCs used in OTEC or other thermal applications has also appeared in literature [3, 20–27].

Ammonia-water ($NH_3-H_2O$) is used in the experimental OTEC setup used for experimental validation of the model developed in this work. More about this working fluid mixture is discussed in subsection 2.2.2. The results of the model using ammonia-water are presented in Chapter 4 and those of the experiments in Chapter 5.

### 2.2. Thermodynamic and Transport Properties of Fluids

Different methods exist to derive thermodynamic and transport properties of pure fluids and their mixtures, which will be discussed in the following subsections. The focus in this work lies on methods with good reliability and convenient implementation into numerical models. More about the developed numerical model, where certain methods are used to compute the fluid properties, is discussed in Chapter 3.

#### 2.2.1. Pure Fluids

To derive the properties of a pure fluid, the most commonly used approaches use either a database or an equation of state (EOS). A fluid property database may be based on experimental measurements of fluid properties, or on selected equations of state. An equation of state describes the state of a fluid or other matter under certain physical conditions, through a thermodynamic relationship. Commonly used equations of state include:

- The Redlich-Kwong EOS (Redlich and Kwong (1949) [28])
- The Soave modifications of the Redlich-Kwong EOS (Soave (1972, 1980) [29, 30])
- The Peng-Robinson EOS (Peng and Robinson (1976) [31])

A more recently developed equation of state is the Perturbed Chain Associating Fluid Theory (PC-SAFT) EOS (Gross and Sadowski (2001) [32]), which is suitable for use with pure fluids but also fluid mixtures. This makes it a promising method to be used for property computation of working fluid mixtures. More about fluid properties of mixtures is discussed in the next section. A computer program developed to determine fluid properties is FluidProp (Colonna and Van der Stelt (2004) [33]). The program is connected to several thermodynamic databases, of which the RefProp database developed by the National Institute of Standards and Technology (NIST) is considered to be the most complete. For pure fluids, FluidProp together with the RefProp database is able to calculate many different fluid thermodynamic and transport properties, such as the enthalpy, internal energy, density, dynamic viscosity, specific heat and thermal conductivity. The surface tension has to be computed using a fluid specific method. For water, data from Vargaftik et al. (1983) [34] is used. More about the general relations is presented in Appendix A, subsection A.2.1.

#### 2.2.2. Binary Mixtures

Winkelmann (2010) [35] states that “properties of mixtures are generally less well understood than those of pure fluids”, which can make it challenging to compute reasonable
values of fluid properties for mixtures. Even though the FluidProp software together with the RefProp database is developed to be able to derive thermodynamic and transport properties of different fluid mixtures, not all properties are derived correctly for all mixtures. It is found that the enthalpy and density are derived correctly, but for other properties a fluid mixture specific method is required. For the case of ammonia-water, possible methods are discussed further ahead in this section. Also, computation of seawater properties is discussed, where seawater is modelled as water with a certain concentration of dissolved salts. Some general relations for binary mixtures are presented in Appendix A, subsection A.2.2.

Other fluid property values are derived using a mixture specific method. For the mixture of ammonia and water this is the method by Conde (2006) [36]. Values can alternatively be computed using a general method described in Perry’s Chemical Engineers’ Handbook (Green and Perry (2007) [37]). Other alternatives are the method described in Ruiter (1990) [38], or using the simplification of a linear interpolation as a function of the molar fraction, between the property values of the pure fluids.

**Ammonia-water**

A binary mixture which is at the moment widely used in condensers or absorbers for OTEC and generation of power from other low-grade heat sources is ammonia-water (Mejbri and Bellagi (2006) [39]). In this mixture, ammonia functions as refrigerant and the ammonia-water mixture as an absorbent (Le Lostec et al. (2013) [40]). The mixture is used for validation of the proposed numerical model in this study. Different studies have appeared on the physical properties of the ammonia-water mixture. A selection of these studies is discussed briefly here, several other studies are reviewed in the review paper by Thorin (2001) [41]. Ziegler and Trepp (1984) [42] report on an equation of state for ammonia-water mixtures, to derive the equilibrium properties. Results are presented in vapor pressure and enthalpy-concentration diagrams, and can be used in the design and testing of absorption units.

Pátek and Klomfar (1995) [43] describe five equations related to the vapor-liquid equilibrium properties of ammonia-water system, with the purpose of application in absorption systems. Tillner-Roth and Friend (1998) [44] discuss the properties of the ammonia-water mixture through a thermodynamic model incorporating the Helmholtz free energy formulation, representing vapor-liquid equilibrium properties, also valid in the pure vapor and liquid phases. Rukes and Dooley (2001) [45] present the calculation methods and property values based on the work by Tillner-Roth and Friend (1998) [44]. In Mejbri and Bellagi (2006) [39], the modelling of thermodynamic properties of the ammonia-water mixture by three different approaches is discussed: empirical, semi-empirical and theoretical. After comparison, the theoretical approach is found to be the most promising, which uses the PC-SAFT (perturbed chain statistical associating fluid theory) equation of state. Conde (2006) [36] describes a large amount of relations for many different thermodynamic and transport properties of the ammonia-water mixture, with the purpose of use in the industrial design of absorption refrigeration equipment.

In Cerezo et al. (2011) [15] correlations to calculate the viscosity for binary mixtures are discussed. Cuenca et al. (2013) [46] report on the thermal conductivity of ammonia-water mixtures measured over a wider temperature range. The values are compared to literature values and fitted with well-known models. Shamsetdinov et al. (2013) [47] report on the ammonia-water binary mixture thermal conductivity, over a wider range of temperatures, pressures and compositions. Correlation models are also developed.

Disadvantages exist for the use of specifically the ammonia-water mixture for energy
generation in thermodynamic cycles. It requires higher temperature differences than certain other known mixtures (Oronel et al. (2013) [17]). Also, due to the fact that the absorbent (water) is relatively volatile, water may end up in streams that should only contain ammonia. This may require an additional distillation column in the cycle. For OTEC applications, this specific disadvantage is not an issue, and the working fluid is considered as one of the most promising candidates for the OTEC technology (Kleute and Vroom (2014) [2]). The focus in this work is therefore on the ammonia-water mixture, but at the same time a step is taken towards introducing a model which is more generally applicable to different binary mixtures. Used thermodynamic relations are found in Appendix A.

In this work, the ammonia-water properties that cannot be computed using FluidProp, are calculated using the methods described in Conde (2006) [36]. This work is chosen since it contains relations to calculate a large selection of parameters, and is developed specifically for design of industrial absorption systems. Values that are computed using FluidProp can also be calculated using this method, such as the liquid and vapor phase composition. Functions to derive these values are based on the same inputs.

**Seawater**
Sharqawy et al. (2011) [48] gives a good overview of existing correlations and data to be used for the computation of thermophysical properties of seawater. The properties of seawater in this work are assumed to be dependent upon the temperature ($T$) and salinity ($S$) of the water. The pressure is not taken into account, since the seawater is assumed to be a pure liquid under all conditions under which the relations described in the paper are valid. The effects of the absolute pressure are thus minimal, assuming the pressure stays within certain bounds ($P < 10$ bar). Relations are found in Appendix A, subsection A.2.4.

### 2.3. Plate Heat Exchangers

When it comes to the heat exchanger type, many options have been investigated in past literature. For this study the focus will be on plate heat exchangers, considered as promising equipment for heat exchange in both single and two-phase flows. First, the general performance of heat exchangers is reviewed before moving into the geometry, structure, advantages and design of plate heat exchangers specifically.

#### 2.3.1. Heat Exchanger Effectiveness

The efficiency of heat exchangers is usually referred to as the ‘effectiveness’. A definition of effectiveness is given by Moran et al. (2003) [49], and is described as the ratio of the actual enthalpy change of the streams to the maximum theoretical enthalpy change. The effectiveness ($\eta$) is thus determined from the inlet specific enthalpy ($h_{in}$), measured outlet specific enthalpy ($h_{out, meas}$), and either the theoretical minimum of the outlet specific enthalpy ($h_{out, min}$) or maximum outlet specific enthalpy ($h_{out, max}$) depending on whether the stream is respectively being cooled or heated. The following relation is used for a heated stream:

$$\eta = \frac{h_{out, meas} - h_{in}}{h_{out, max} - h_{in}}$$  \hspace{1cm} (2.1)

For a cooled stream, $h_{out, max}$ should be replaced by $h_{out, min}$. In OTEC practice, both the evaporator and condenser are operated vertically positioned with a single-pass of the
working fluid. For the condenser, vapor enters from above, condenses and runs down in the heat exchanger (under influence of gravity). Liquid may also be present in the entering flow, depending on the operating conditions. Usually the other fluid (seawater) is pumped upwards through the heat exchangers, resulting in a countercurrent flow of both streams. Heat exchanger effectiveness is determined by the available total surface area, mean temperature difference between the two streams, the required pinch temperature for heat transfer and the heat transfer performance. As stated in Kleute and Vroom (2014) [2], the low effectiveness of the heat exchangers limits the design flexibility of the thermodynamic cycle, the desired operating conditions now push the limits of what is technically feasible.

2.3.2. GEOMETRY AND STRUCTURE

Good descriptions of plate heat exchanger geometry are given in Stephan et al. (2010) [8] and Tovazhnyanski and Kapustenko (1984) [50]. Most commonly used is the chevron-type plate heat exchanger, which has a close to sinusoidal corrugation pattern with a certain amplitude and wavelength. The crests and troughs of this pattern are oriented in a certain inclination angle to the main flow direction, in one or several strips. For chevron plates, this angle to either the main flow direction or horizontal is referred to as the chevron angle (\(\phi\)). Other types of corrugated plates exist as well, such as the herringbone type. All plates are placed together to form the heat exchanger. The plate spacing can be adjusted, but generally the plates are spaced as close as possible to have adjacent plates in contact. The plates can be kept in place using different methods. For example, welding is used for the welded type and plates are gasketed and clamped together for the gasket type.

Brazed plate heat exchangers, which were introduced in the 1970s, can be used for condensation and vaporization of high pressure refrigerant fluids (Longo et al. (2015) [51]). Tribbe and Müller-Steinhagen (2001) [52] provide a good description of the single plate geometry, consisting of inlet and outlet distribution triangles and a corrugated section. The pattern consists of repeating units cells that remain uniform along the length of the duct. A representation of a general chevron plate heat exchanger is presented in Figure 2.2. The casing or structure around the heat exchanger is not presented in this figure, but the main lines and distribution sections are visible. To present the structure of the plates more clearly, they have been spaced by large distances in the figure, where in practice they will be stacked directly on top of one another or with a limited amount of spacing. The average distance between the centers of the stacked plates in the direction of stacking is described as the compressed plate pitch (\(p_{cp}\)). When the heat exchanger plates are stacked as close as to have adjacent plates in direct contact with one another, structures consisting of units cells are formed in between the plates as a consequence of the corrugation pattern. A schematic representation of cross-corrugated plates and a single unit cell is found in Figure 2.3.

Using several basic geometrical parameters of the plate heat exchanger, all other geometrical parameters can be calculated. It is assumed the following geometrical parameters are known or can be measured for ordinary plate heat exchangers:

- Width of plate available for flow and transfer (\(B_p\)).
- Plate port-to-port length available for flow and transfer (\(L_p\)).
- Compressed plate pitch (\(p_{cp}\)).
Figure 2.2: Structure of a commercial plate heat exchanger (Asano *et al.* (2004) [53]) with corrugated chevron plates under counterflow operation.

Figure 2.3: (a) Cross-corrugated plates and (b) Typical unit cell of a commercial plate heat exchanger (Zhang and Che (2011) [54]).
2.3. PLATE HEAT EXCHANGERS

- Plate thickness \(d_p\).
- Outer walls thickness \(d_o\).
- Diameter of the inlet and outlet ports \(d_{port}\).
- Number of plates \(N_p\).
- Corrugation chevron angle to horizontal \(\phi\).

Other geometrical parameters need to be calculated based on the known values, and the relations used are discussed in Appendix A, section A.3:

- Channel gap \(d_g\).
- Corrugation amplitude \(\hat{a}\).
- Width of the stacked plates \(W_p\).
- Number of channels \(N_c\).
- Number of walls \(N_w\).
- Number of primary fluid channels \(N_{c,wf}\).
- Number of secondary fluid channels \(N_{c,cw}\).
- Single plate plane projected area \(A_0\).
- Total area of the plates \(A_{0,tot}\).
- Single channel cross-sectional area \(A_c\).
- Total channels cross-sectional area \(A_{c,tot}\).
- Primary channels cross-sectional area \(A_{c,wf}\).
- Secondary channels cross-sectional area \(A_{c,cw}\).
- Corrugation pitch \(p\).
- Chevron angle to the main flow direction \(\phi_{flow}\).
- Corrugation wavelength \(\Lambda\).
- Wave number or corrugation parameter \(\Omega\).
- Area enlargement factor \(\Gamma\).
- Hydraulic diameter \(d_h\).
- Enlarged plate surface area \(A_p\).
- Wetted perimeter of the flow \(\Pi\).
2.3.3. ADVANTAGES
Several advantages are reported in literature for the use of plate heat exchangers as compared to other traditional types of heat exchangers, such as the shell and tube type. Certain advantages are listed here:

- High turbulence achieved without great effort ([50, 55–58])
- Single pass of working fluid across the plate is sufficient ([50])
- Small liquid holdup ([8])
- Relative close temperature approach and uniform residence time ([59])
- High heat transfer coefficient ([50, 56, 60])
- High heat transfer area ([58])
- Good thermal performance, high thermal effectiveness ([57, 58, 61])
- Improved mass transfer process ([60])
- Excellent fluid distribution ([58])
- Compactness ([8, 58, 60, 61])
- Low cost ([57, 61])
- Simple adjustment to changed operating conditions by replacement, rearrangement or addition of plates ([8, 55, 56])

2.3.4. DESIGN
Sizing of plate heat exchangers is discussed, among others studies, in Quoilin et al. (2011) and Kirkenier (2014) [3]. For use of the plate heat exchangers in absorption or condensation applications, it is important that the pressure drop across the plates is maintained below a reasonable level (Tovazhnyanski and Kapustenko (1984) [50]). Asano et al. (2004) [62] report on the important issue between maximizing heat transfer whilst keeping pressure drop to a minimum. These and other design issues are not further discussed in this work, where this study focuses on the modelling of heat exchangers of given geometry rather than on their design.

2.4. HEAT TRANSFER, MASS TRANSFER AND PRESSURE DROP
In a large selection of scientific literature, condensation and absorption are discussed respectively for pure fluids and mixtures inside plate heat exchangers or other geometries. The term absorption is used for phase change processes of mixtures, where also mass transfer of species plays a role, including diffusion and convection. In literature the term condensation is sometimes used for flowing cooled mixtures undergoing a phase change. In this section, also literature is discussed on general two-phase flow in plate heat exchangers, without phase change. This may give insight into the flow patterns occurring in the heat exchangers, which is an important issue in understanding the behavior for flows undergoing phase change inside these geometries. The focus in most research in the field is on
investigating pressure drop and heat transfer performance, or at least one of the two. These together are the two most important parameters for determination of the heat exchanger performance, and can be directly related to their use in thermodynamic cycles. Pressure drop and heat transfer directly influence each other in heat exchangers where the working fluid undergoes a phase change, since the pressure inside the heat exchanger is related to the saturation temperature of the fluid through the vapor-liquid equilibrium (Kreissig and Müller-Steinhagen (1992) [56]).

2.4.1. General two-phase flow
Certain literature reports on general two-phase flow, without relevant phase-change processes, for flow in plate heat exchangers or other relevant geometries. Both are discussed in the next paragraphs.

Plate heat exchangers
Kreissig and Müller-Steinhagen (1992) [56] report on frictional pressure drop for cocurrent gas-liquid two-phase flow in plate heat exchangers. Measurements of air-water flow are presented and correlations developed for tubular flow are compared with the experimental results. Tribbe and Müller-Steinhagen (2001) [52, 63] report on experiments to measure the pressure drop in steady adiabatic air-water two-phase flows in plate heat exchangers with different channel geometries. Also, a mixture of air and carboxymethylcellulose is used. In various experimental and analytical studies, relations for the pressure gradient are derived either through empirical correlations or phenomenological models. Visualization experiments are performed, which are discussed in section 2.5. Vlasogiannis et al. (2002) [59] report on cocurrent downward air-water two-phase flow and heat transfer in the vertical channels of a plate heat exchanger. Visualization is performed for different flow regimes and measurements of the heat transfer are performed. The study aims to delineate the influence of flow regime on the heat transfer coefficient. Nilpueng and Wongwises (2010) [64] discuss experimental work on the two-phase gas-liquid flow characteristics inside a plate heat exchanger, focusing on measurement of pressure drop and observation of flow patterns. Experiments are performed for vertical, single pass, counter-current upflow and downflow in stainless steel commercial plate heat exchangers with corrugated sinusoidal shapes of unsymmetrical chevron angles.

Other geometries
Hewitt and Roberts (1969) [65] discuss the interface configurations or "flow patterns" in vertical two-phase gas-liquid flow inside tubes, using visualization techniques involving high speed flash photography and simultaneous X-radiography. A detailed evaluation of flow structure and pattern is possible, even in the high liquid mass flow regimes, where this is usually considered difficult. Oshinowo and Charles (1974) [66] discuss two-phase gas-liquid flow in vertical tubes, for both upflow and downflow. Six different flow patterns are identified, and correlations are given for their transitions. The used gas-liquid combinations include air-water, steam-water, nitrogen-mercury and air-heptane. Data from flowing gas-oil wells is used as well. It is stated that a clear understanding of the momentum, heat and mass transfer rates involved in two-phase flow is essential for the design and operation of process equipment. The effect of system variables on flow patterns needs to be investigated, to get a basic understanding of the mechanics of the various flow configurations and the development of sound transport equations. The study aims to provide a
more complete understanding of the flow phenomena occurring under two-phase flow in vertical tubes. Oshinowo and Charles (1974) [67] continues on the other work, as the second publication in a series of two. Here, the liquid holdup and pressure drop are studied for the system under consideration.

2.4.2. Condensation

Various literature reports on condensation, for different pure fluids or mixtures in plate heat exchangers. Tovazhnyanski and Kapustenko (1984) [50] report on mass and heat transfer and pressure drop during single-phase flow and condensation in plate heat exchangers. The analogy between heat and momentum transfer is used to obtain heat transfer data from the pressure drop, related to the wall shear stress. Also, fouling inside the heat exchanger is studied. Factors that govern process intensity are listed:

1. Heat transfer in the condensate film.
2. Mass transfer of vapor to the interface border (if a noncondensable gas is present in the vapor).
3. Thermal resistance of the plate wall.
4. Thermal resistance of the fouling on the surface of the plate wall.
5. Convective heat transfer to the other fluid.

The paper states that data about the above factors and the mechanism of transport process intensification in the channels of the plate heat exchanger is needed for prediction of their performance.

Yan et al. (1999) [68] report on an experimental study on condensation heat transfer and pressure drop of refrigerant R-134a in a vertical plate heat exchanger of common geometry, under countercurrent flow, taking into account factors such as the vapor quality, mass flux, heat flux and inlet pressure. Correlations for heat transfer coefficients and pressure drop are given in terms of the Nusselt number and friction factor. Thonon and Bontemps (2002) [69] report on condensation of hydrocarbons (pure or mixtures) in a compact plate heat exchanger. Several factors will improve the heat transfer during condensation for mixtures:

1. Reduce the thermal resistance on the vapor side.
2. Increase the vapor velocity (for vapor mixtures containing noncondensables).
3. Use of finely undulated (wavelike) surfaces: corrugation, since it promotes turbulent equilibrium between the phases.

Würfel and Ostrowski (2004) [70] report on experimental investigations of heat transfer and pressure drop during the condensation process within plate heat exchangers of the herringbone type. Essential heat transfer resistances are found to exist in the condensate film. Several factors intensify heat transfer during the film condensation process:

1. An increase of interfacial shear stress
2. Roughness of the heat transfer area
3. Ribbed or corrugated surfaces

The correlations for heat transfer and pressure drop can be improved by performing more experiments and obtaining more data, using systems of different geometries and by local investigations of conditions. The work of García-Cascales et al. (2007) [57] contains an overview of heat transfer correlations for condensation (along with boiling and single-phase flows) in plate heat exchangers, and states that relations are limited to certain fluids and proposed models. Winkelmann (2010) [35] discusses heat transfer and pressure drop in corrugated channels using experiments for condensation. Also experiments for mixture condensation, adiabatic two-phase flow and single-phase flow are performed, all in channels of the same geometry. The aim is to contribute to the design of plate heat exchangers used as condensers. Quoilin et al. (2011) [16] report on general correlations for heat transfer in a plate heat exchanger, including condensation. Arsenyeva et al. (2011) [71] report on equations to calculate heat transfer and pressure drop for both single phase and condensing flows in plate heat exchangers, using the analogy between heat and momentum transfer. A one dimensional mathematical model is proposed, discussed in more detail in section 2.7. The study has the objective to derive proper correlations for condensation processes in channels of plate heat exchangers with corrugations. Both experimental and theoretical studies are used, taking into account change in local parameters across the channel length. All the different encountered resistances for heat and mass transfer are incorporated:

1. Heat and mass transfer from flow core of condensing stream to the condensate film surface.
2. The thermal resistance of condensate film.
3. Heat transfer across the plate wall (and its fouling).

Grabenstein and Kabelac (2012) [61] discuss experimental and theoretical analyses of the local condensation heat transfer in plate heat exchangers. In the paper it is stated that flow and heat interactions as well as their dependence on the geometrical parameters of the corrugated plates and operating conditions should be studied in more detail. The paper also states that no reliable correlations have yet been published in literature regarding heat transfer and the friction pressure drop in plate heat exchangers during phase change. Correlations that are presented are usually of empirical nature, based on equations fitted to a specific experimental setup. The study aims to work towards a reliable model based on physical approaches for the prediction of heat transfer and pressure drop during condensation inside a plate heat exchanger. A stepwise approach towards the construction of such a phenomenological model is proposed involving experiments to visualize flow patterns and measure heat transfer rates. By means of a flow pattern map, the heat transfer coefficient and the friction factor are calculated depending on vapor quality, mass flow and geometrical parameters. Experimental data is used to validate the predictive models. More about this study is discussed in subsection 2.5.3.

Longo et al. (2015) [51] report on a new computational procedure for the heat transfer coefficient during pure and near azeotropic refrigerant vapor condensation inside a brazed plate heat exchanger (BPHE) of the herringbone-type. The model is validated using a wide set of experimental data previously obtained by the authors, for condensation of both pure
and near azeotropic refrigerants. Refrigerants include hydrofluorocarbons (HFCs), hydrocarbons (HCs) and low global warming potential hydrofluoroolefins (HFOs).

### 2.4.3. Absorption

#### Plate heat exchangers

de Vega et al. (2006) [72] discuss the performance of a lithium bromide-water absorption chiller working with plate heat exchangers. Heat transfer coefficients are reported for the different heat exchangers, including the condenser. Driving temperatures and circulated mass flow rates are varied to analyze the heat transfer, and performance in terms of the efficiency or coefficient of performance (COP). In Jeong et al. (2009) [73] a computational fluid dynamics (CFD) technique is used to examine the heat transfer and pressure drop characteristics of welded type plate heat exchangers for absorption application. Besides the numerical analysis, described in more detail in section 2.7, also experiments are carried out. Different corrugated plates are investigated, of the chevron, elliptic and round corrugation type. Lithium bromide-water ($\text{LiBr-H}_2\text{O}$) is used as the mixture in this study.

Cerezo et al. (2009) [60] report on an experimental study of an ammonia-water bubble absorber using a plate heat exchanger, with a focus on the use as absorption chillers for refrigeration purposes. Typical conditions for absorption chillers are used, such as the low temperature heat source. A commercial corrugated plate heat exchanger is used consisting of three channels, ammonia vapor being injected in the central one in bubble mode. The absorption flux, heat transfer coefficient and thermal load are measured, and effects of parameters such as pressure, temperature, concentration and flow rate are investigated. Cerezo et al. (2010) [74] describe the comparison of a developed mathematical model for ammonia-water bubble absorbers with experiments performed using a plate heat exchanger. The effect of certain operating parameters are studied, such as the thermal load and mass absorption flux. Cerezo et al. (2011) [15] discuss different working fluid mixtures for use in plate heat exchangers used as bubble absorbers, with the focus on commercial absorption refrigeration. A mathematical model is developed, which is discussed in more detail in section 2.7. Effects of solution viscosity and other parameters corresponding to commercial refrigeration conditions in bubble mode on the performance in terms of thermal load and COP are studied.

Le Lostec et al. (2013) [40] discuss simulation of a single-stage absorption chiller using an ammonia-water mixture as working fluid, under steady-state conditions. The focus lies on absorption chillers, which use renewable energy in the form of waste heat energy for refrigeration purposes, but the findings may apply to power productions systems as well. The model is described in more detail in Chapter 3. The model is validated using experimental data, and used to predict temperature and concentration profiles in the heat exchangers. The effect of the temperature and other operating parameters on the performance in terms of COP is discussed. Oronel et al. (2013) [17] report on an experimental study using ammonia-lithium nitrate and ammonia-(lithium nitrate + water) mixtures to characterize the performance of bubble absorption in a corrugated plate heat exchanger of three channels. Operating conditions are similar to those of absorption systems driven by low temperature heat sources, and the effects of several of these on the absorber efficiency are studied. Jung et al. (2014) [58] attempt to analyze the heat transfer characteristics of ammonia-water absorption in plate heat exchangers, with the focus on application in hybrid heat pumps. An analysis is performed on the effect of internal pressure, ammonia
weak solution concentration and absorber geometric dimensions on the absorber capacity and COP.

**OTHER GEOMETRIES**

Absorption of ammonia-water is discussed in a mini-channel annulus \([75, 76]\) and in mini-channel multi-tube configurations in literature \([77]\). In these studies, the heat transfer and pressure drop relations are discussed. The derivation of temperature and velocity profiles through a mathematical model, using certain assumptions, is also discussed.

Heat and mass transfer modelling in ammonia-water absorption systems is discussed in Sieres and Fernández-Seara (2007) \([78]\). Relations are considered that take into account the heat and mass transfer resistances in the liquid and vapor phases. A model is developed, described in more detail in section 2.7, and compared to experimental data. A discussion is presented on the coupled interaction of heat and mass transfer, the contributions to both and the influence on flow properties. Other studies on ammonia-water absorption have appeared in literature \([79–84]\), but are not discussed here.

### 2.5. TWO-PHASE FLOW CHARACTERISTICS

Different studies have been performed on the flow characteristics of fluids inside plate heat exchangers. Studies are divided into three types of investigations, though several studies incorporate more than one type:

1. **Flow paths**: describes the path of the main fluid flow along the furrows and trenches of the heat exchanger corrugated plates, either for single or two-phase flows.

2. **Flow regimes**: describes the degree of turbulence in the flow, related to the Reynolds number. The complex flow paths caused by the plate corrugations induce turbulence.

3. **Flow patterns**: describes the distribution of the gas and liquid phases for two-phase flows, on a larger scale (the entire plate) or on a smaller scale (within a single flow channel between the plates).

#### 2.5.1. FLOW PATHS

Depending on the heat exchanger geometry, the main flow path of the fluid inside the heat exchanger is either cross flow, also referred to as crossing flow, or wavy longitudinal flow. In the first case the main flow path of the fluid is from one side of the plate to the other, before changing direction and moving along the other plate again to the other side. In the second case, the fluid changes direction at each crossing point of two troughs of the opposing plates between which the liquid flows. A representation of a single plate geometry and possible flow paths are presented in Figure 2.4, taken from Stephan *et al.* (2010) \([8]\). Crossing flow is observed more for heat exchanger plates with corrugations of low chevron angles to the main flow direction \((\phi_{flow})\), where a combination of crossing and wavy longitudinal flow is observed for higher chevron angles. In many cases, both paths are assumed to exist simultaneously. The central herringbone spine of the plate acts as a physical boundary to flow, directing approaching flows along the furrows of the opposite plate.
2.5.2. Flow regimes

Several studies report that turbulent flow occurs at low Reynolds numbers ($Re > 200$) for flow in plate heat exchangers, leading to the fact that in many practical applications of plate heat exchangers the flow will be turbulent ([55–57, 69]).

Kreissig and Müller-Steinhagen (1992) [56] report on cocurrent two-phase flow in plate heat exchangers. Due to the complicated geometry of the heat exchangers, with sudden contractions, also sudden changes in flow direction occur under two-phase flow in these heat exchangers. This keeps the two phases well mixed, and results in the fact that heterogeneous flow patterns are unlikely to occur in a plate heat exchanger. A high degree of turbulence results from the corrugated surface, even for low Reynolds numbers. Interaction between the two flowing phases influences the resulting flow patterns, and thus also the pressure drop and heat transfer properties. Vlasogiannis et al. (2002) [59] state that satisfactory understanding is lacking of the fundamental problem of gas-liquid film flow over a corrugated surface, and the effect of corrugation parameters. This understanding is needed to comprehend gas-liquid two-phase flow in the flow passages of the plate heat exchanger and of the effect of the adopted flow regimes on the heat transfer characteristics.

Grabenstein and Kabelac (2012) [61] give an explanation of heat transfer for different flow regimes. At low Reynolds numbers high heat transfer rates are achieved by the enlarged surface in the plate heat exchanger. At high Reynolds numbers, the heat transfer is enhanced due to the high degree of turbulence resulting from the small flow passages and complex flow structure. Longo et al. (2015) [51] report that at an equivalent two-phase Reynolds number of around 1600, the condensating refrigerant flow reaches the transition point between the gravity-dominated and forced convection condensation regime. The definition of the equivalent Reynolds number is presented in equation 3.11.
2.5.3. FLOW PATTERNS
Relevant literature on flow patterns for two-phase flow in corrugated channels of plate heat exchangers is discussed in Appendix B. Different studies for downward flow are compared, and an attempt is made to construct a universal flow pattern map for downward two-phase flow in corrugated channels. To allow this, all relevant data for comparison of the studies is collected from the specific sources [35, 59, 61, 63, 64]. An overview of the most important data for each study is presented in Table 2.1. As is observed in the table, important data necessary for proper comparison between studies is missing in certain cases. Even though, an attempt has been made to compare studies in terms of their flow pattern predictions, given the available data.
Table 2.1: Overview of studies on visualisation of flow patterns for two-phase flow in corrugated channels of plate heat exchangers

<table>
<thead>
<tr>
<th>Study</th>
<th>Heat exchanger model</th>
<th>Phase change</th>
<th>Fluids used</th>
<th>Plate materials</th>
<th>Visualization</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tribbe and Müller-Steinhagen (2001) [63]</td>
<td>Alfa-Laval</td>
<td>No</td>
<td>Air</td>
<td>Water</td>
<td>Polyester</td>
</tr>
<tr>
<td>Vlasogiannis et al. (2002) [59]</td>
<td>Alfa Laval P-01</td>
<td>No</td>
<td>Air</td>
<td>Water</td>
<td>Stainless steel, Plexiglass</td>
</tr>
<tr>
<td>Nipueng and Wongwises (2010) [64]</td>
<td>Commercial</td>
<td>No</td>
<td>Air</td>
<td>Stainless steel, Polyurethane</td>
<td></td>
</tr>
<tr>
<td>Grabenstein and Kabelac (2012) [61] (1)</td>
<td>Industrial</td>
<td>Yes</td>
<td>Air</td>
<td>R-134a</td>
<td>Stainless steel, Polyurethane</td>
</tr>
<tr>
<td>Grabenstein and Kabelac (2012) [61] (2)</td>
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<td>Yes</td>
<td>Air</td>
<td>R-365mfc</td>
<td>Stainless steel, Polyurethane</td>
</tr>
<tr>
<td>Grabenstein and Kabelac (2012) [61] (3)</td>
<td>Industrial</td>
<td>Yes</td>
<td>Air</td>
<td>Air</td>
<td>Polyurethane</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Basic data</th>
<th>Heat exchanger model</th>
<th>Phase change</th>
<th>Fluids used</th>
<th>Plate materials</th>
<th>Visualization</th>
</tr>
</thead>
<tbody>
<tr>
<td>Study</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Tribbe and Müller-Steinhagen (2001) [63]</td>
<td>Alfa-Laval</td>
<td>No</td>
<td>Air</td>
<td>Water</td>
<td>Polyester</td>
</tr>
<tr>
<td>Vlasogiannis et al. (2002) [59]</td>
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<td>No</td>
<td>Air</td>
<td>Water</td>
<td>Stainless steel, Plexiglass</td>
</tr>
<tr>
<td>Nipueng and Wongwises (2010) [64]</td>
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<td>No</td>
<td>Air</td>
<td>Stainless steel, Polyurethane</td>
<td></td>
</tr>
<tr>
<td>Grabenstein and Kabelac (2012) [61] (1)</td>
<td>Industrial</td>
<td>Yes</td>
<td>Air</td>
<td>R-134a</td>
<td>Stainless steel, Polyurethane</td>
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<tr>
<td>Grabenstein and Kabelac (2012) [61] (2)</td>
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<td>Yes</td>
<td>Air</td>
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<td>Yes</td>
<td>Air</td>
<td>Air</td>
<td>Polyurethane</td>
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<table>
<thead>
<tr>
<th>Plate length (mm)</th>
<th>Plate width (mm)</th>
<th>Corrugation structure</th>
</tr>
</thead>
<tbody>
<tr>
<td>Study</td>
<td></td>
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</tr>
<tr>
<td>Tribbe and Müller-Steinhagen (2001) [63]</td>
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<tr>
<td>Winkelmann (2010) [35]</td>
<td>641</td>
<td>550</td>
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<tr>
<td>Nipueng and Wongwises (2010) [64]</td>
<td>715</td>
<td>635</td>
</tr>
<tr>
<td>Grabenstein and Kabelac (2012) [61] (1)</td>
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<td>124</td>
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<td>Grabenstein and Kabelac (2012) [61] (2)</td>
<td>388</td>
<td>100</td>
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<td>388</td>
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</tbody>
</table>

<table>
<thead>
<tr>
<th>Corrugation dimensions (mm)</th>
<th>Chevron angle 1 (°)</th>
<th>Chevron angle 2 (°)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Study</td>
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<td></td>
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<td>30</td>
</tr>
<tr>
<td>Vlasogiannis et al. (2002) [59]</td>
<td>1.2</td>
<td>10</td>
</tr>
<tr>
<td>Winkelmann (2010) [35]</td>
<td>1.5</td>
<td>30</td>
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<tr>
<td>Nipueng and Wongwises (2010) [64]</td>
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<td>35</td>
</tr>
<tr>
<td>Grabenstein and Kabelac (2012) [61]</td>
<td>1.5</td>
<td>63</td>
</tr>
</tbody>
</table>

1 Assumed profile
2.5. TWO-PHASE FLOW CHARACTERISTICS

ORIGINAL FLOW PATTERN MAPS

The original flow pattern maps are presented in Appendix B. These are reproduced as accurately as possible by taking a certain amount of points on each line dividing two different flow patterns, and estimating the values of the quantities on the x- and y-axis of the graph. For the study by Winkelmann (2010) [35], no dividing lines between flow patterns are presented in the figure by the author. Only descriptions of the flow pattern are given for each data point. Dividing lines are constructed based on the data points presented in the figure of the study. Each dividing line for each study is then constructed by taking the data points estimated from the figures and producing several additional data points in between these points through an assumed linear relationship. In this way, straight lines are reproduced closer to the original than curved lines but in both cases a reasonable representation of the original curve is obtained.

The lines of different studies are plotted in the same figure, with the superficial velocities of the gas or vapor and liquid phases (\(u_{act,G}\) and \(u_{act,L}\)) as quantities on the x- and y-axis respectively. In the study by Vlasogiannis et al. (2002) [59] the original flow pattern map has the quantities on the opposite axes, so the newly constructed figure is a transformation of the original one. Three studies originally plotted the flow pattern maps using logarithmic scales [35, 63, 64] of which two are plotted together in Figure 2.5. Flow pattern maps for the three other studies are plotted together in Figure 2.6. In both figures, the descriptions of the flow patterns given by the authors of the studies are added to the corresponding locations in the figure. Figure 2.7 is constructed with logarithmic scales for both axes, which contains all the flow pattern maps.

![Figure 2.5: Flow pattern map of two-phase gas-liquid flow in a corrugated channel, using a logarithmic scale for selected studies [35, 63].](image)

COMPARISON OF DIFFERENT STUDIES

As is observed in Table 2.1, the geometries of the corrugated channels used in the different studies are not the same, besides the fact that certain details are not clear for specific stud-
Figure 2.6: Flow pattern map of two-phase gas-liquid flow in a corrugated channel, using a linear scale for selected studies [59, 61, 64].

Figure 2.7: Flow pattern map of two-phase gas-liquid flow in a corrugated channel, for all relevant studies using a logarithmic scale [35, 59, 61, 63, 64].
2.5. TWO-PHASE FLOW CHARACTERISTICS

All studies use two-phase flow of air and water under conditions where phase changes during flow are negligible. In the study by Grabenstein and Kabelac (2012) [61], also refrigerants R-134a and R-365mfc are used, which may undergo phase change under the experimental conditions. A flow pattern map is presented only for R-365mfc in the study. Even though the fluids in most of the studies are the same (air and water), the operating conditions in terms of pressure and temperature are different, or in some cases unclear. Fluid properties will vary between studies as a consequence of these different conditions. This fact, and the fact that the geometries are different, makes the flow pattern maps in terms of superficial velocities of the phases not directly quantitatively comparable between studies. For this reason, other quantities should be selected to be used on the axes, which also take into account the specific geometry and operating conditions affecting the fluid properties. Dimensionless quantities are preferred in this case, since these can express the relative importance of the physical phenomena that determine the flow field and flow patterns.

RELEVANT FORCES IN TWO-PHASE FLOWS

Generally speaking, four main forces play a role in flow development of two-phase flows within a system, and the relative magnitude of these forces at different locations in the system together determine the details of the flow. These four forces are the contributions of inertia, shear stress, gravity and surface tension. Two forces play a role mostly in the bulk flow of fluids, gravity and inertia, where surface tension is only relevant at the gas-liquid interface. Viscous forces are relevant both within the bulk fluid, at phase interfaces and at the regions of fluid flow near the walls. Dimensionless numbers associated with these forces and with describing details of fluid flow are the Reynolds, Froude and Weber numbers. These numbers describe the inertial forces relative to respectively the viscous, gravitational and surface tension forces. Equations for the general form of these dimensionless numbers are found in Appendix A, Equations A.8, A.3 and A.11. The characteristic length \((d)\) in the case of plate heat exchangers is the hydraulic diameter \((d_h)\). The equation of the Reynolds number for two-phase flow \((Re_{TP})\) in corrugated channels is presented next, where the subscript \(L\) is for the liquid phase and \(G\) for the gas phase:

\[
Re_{TP} = \frac{m}{\rho_L \left( \rho_G - \rho_L \right) d_h g^{1/2}} \left( 1 - X \right) + X \left( \frac{\rho_L}{\rho_G} \right)^{1/2} d_h / \mu_L \tag{2.2}
\]

This equation is more relevant to use when the two-phase flow is relatively homogeneous, such as for flow of small dispersed gas bubbles in a continuous liquid phase. Equations of the Froude number for two-phase flow in respectively horizontal and vertical tubes are given by equations:

\[
Fr_{TP, hor} = \frac{m}{\rho_L \left( \rho_G - \rho_L \right) d_h g^{1/2}} \tag{2.3}
\]

\[
Fr_{TP, ver} = \frac{\left( u_{sup,G} + u_{sup,L} \right)^2}{g d_h} \tag{2.4}
\]

The parameters present in these equations are the superficial flow velocities of the gas and liquid phases, the hydraulic diameter, the gravitational constant \((g)\) and different fluid specific properties: density \((\rho)\) and viscosity \((\mu)\).
OTHER RELEVANT PARAMETERS

For two-phase flow in corrugated channels, additional dimensionless numbers may be defined based on geometrical parameters. The geometrical parameters that are expected to play a role are those related to the detailed geometry of the plates: the corrugation pitch ($p$), the corrugation amplitude ($\hat{a}$) and the chevron angle ($\phi$). These parameters and other related parameters are discussed in subsection 2.3.2, and the equations in Appendix A, section A.3.

For two-phase flow in tubes, different studies present selected dimensionless quantities to predict the two-phase flow pattern under specified conditions. For two-phase flow in horizontal tubes, the study by Hewitt and Roberts (1969) [65] is well-known for the prediction of flow patterns and the presentation of a flow pattern map. However, the quantities on the axes of the flow pattern maps for this study are not dimensionless quantities. The study by Oshinowo and Charles (1974) [66] presents flow pattern maps for vertical flow in tubes, both upward and downward. Here, dimensionless quantities are used on the axes. On the x-axis the Froude number is present, which for two-phase flow is given through Equation 2.4, and is a function of both gas and liquid superficial velocities. Also a property modifying group ($\Lambda$) is defined. This parameter is not to be confused with the wave number discussed previously, and is a function of the liquid phase properties relative to those of water under the same conditions. This relation is given by the equation:

$$\Lambda = \frac{\mu_L/\mu_W}{\left[\frac{\rho_L}{\rho_W} \left(\frac{\sigma_L}{\sigma_W}\right)^3\right]^{1/4}}$$

(2.5)

The two dimensionless quantities are combined on the x-axis in the parameter $\frac{Fr TP}{\sqrt{\Lambda}}$. On the y-axis the square root of the delivered gas to liquid volume ratio ($R_v$) is used, which is given as the ratio of gas and liquid superficial velocities through the equation:

$$R_v = \frac{V_G}{V_L} = \frac{u_{sup,G}}{u_{sup,L}}$$

(2.6)

The original flow pattern maps by Oshinowo and Charles (1974) [66] are presented in Appendix B in Figures B.28 and B.29.

TRANSLATION INTO NEW FLOW PATTERNS MAPS

An effort has been made to translate the original flow patterns maps for two-phase downward flow in corrugated channels to new flow pattern maps with the dimensionless quantities on the axes as used by Oshinowo and Charles (1974) [66]. This is done by starting with the data points of superficial velocities from the original map, together with the reported channel geometries and conditions of pressure and temperature. Not all geometrical parameters and operating conditions are reported in each study, or they are not reported clearly. In certain cases assumptions or guesses had to be made to make further calculations possible. Fluid properties of both gas and liquid phases are computed using FluidProp, given the pressure and temperature. Isothermal and adiabatic conditions are assumed, as is also proposed in different studies. The surface tension can not be computed using FluidProp. For the studies with air-water, surface tension values are taken from Vargaftik et al. (1983) [34] and for R-365mfc from Fröba et al. (2004) [85].
From the obtained fluid properties, the property modifying group is computed. For calculation of the two-phase Froude number ($Fr_{TP}$) the hydraulic diameter is needed, which is given by Equation A.90, or through the simplified equation:

$$d_h = 2d_g \quad (2.7)$$

This equation is used when geometrical details of the corrugations are not known. The delivered gas to liquid volume ratio is computed as discussed before. All dimensionless quantities are then known and the new flow patterns maps are plotted. The plots of all flow pattern maps combined are presented in Figure 2.8. It must be noted that to avoid a wide spread of data points within the graph in both x- and y-directions, which makes it difficult to identify individual dividing lines between flow patterns, certain data points are left out of the combined flow pattern map. Data points that are left out for very low values of the x-axis parameter ($Fr_{TP} < 0.0005$) are for the studies of Nilpueng and Wongwises (2010) [64] and Vlasogiannis et al. (2002) [59]. Certain data points for the study of Winkelmann (2010) [35] are left out, corresponding to the dividing lines other than the one between the churn and wavy flow regimes. This last line is plotted in the figure. The other dividing lines in this study correspond to large values of $\sqrt{\frac{R_v}{A}}$, outside the ranges of lines for other studies ($\sqrt{\frac{R_v}{A}} > 20$). Descriptions of flow patterns are added to corresponding locations in the figure, except for the AB and BA flow regimes described in Vlasogiannis et al. (2002) [59], as this would make the figure unreadable due to the presence of other lines in the figure.

![Figure 2.8: Flow pattern map of two-phase gas-liquid flow in a corrugated channel, with dimensionless quantities on the x-axis ($Fr_{TP}$) and y-axis ($\sqrt{\frac{R_v}{A}}$). Definitions of the dimensionless quantities are explained in the text [35, 59, 61, 63, 64].](image)

**Errors in the Flow Pattern Map Construction**
The constructed figure gives a qualitative impression of in which areas within the flow pattern map the reported flow patterns are observed. However, it does not provide fully re-
liable quantitative data that clearly defines dividing lines for two-phase flow patterns for vertical downflow in corrugated channels. Several causes of error or bias are identified that explain why the plotted lines do not overlap, or do not reflect the actual line they represent well. Some of these causes were already pointed out in the text of this section, but to gain a complete overview the entire list is included and discussed here:

- **Different definitions of parameters**: certain quantities used in the construction of the new flow pattern map are defined differently between studies, or have different definitions in general literature. Different definitions for the Froude number exist, and it is not clear which one is the most representative to use for downward two-phase flow in corrugated channels. In this case, Equation 2.4 is used, taken from the study by Oshinowo and Charles (1974) [66]. Also, the definitions for corrugation pitch ($p$) and corrugation wavelength ($Λ$) may be confused with one another in certain studies. Finally, the channel spacing or channel gap ($d_g$) is used in different studies either with or without taking the plate thickness ($d_p$) into account.

- **Incomplete or unclear reporting of geometrical parameters**: certain parameters of the plate geometries are not defined or are not defined clearly for all studies, such as the plate thickness ($d_p$), plate height and width ($L_p$ and $B_p$) and the type of profile and pitch ($p$) of the corrugations. Not all parameters are used for computation of the presented flow pattern maps, but are important when comparing studies and taking all effects of geometry on the flow patterns into account.

- **Incomplete reporting of operating conditions**: The pressure ($P$) and temperature ($T$) are not reported in every study, so a common pressure in plate heat exchangers of around 5 bar and room temperature (25 °C) are adopted for certain studies. The values of pressure and temperature affect the values of different fluid properties, which are used in the property modifying group ($Λ$). This parameter is used to construct the flow patterns.

- **Simplified surface tension calculation method**: The studies by Vargaftik et al. (1983) and Fröba et al. (2004) are used to calculate the surface tension of respectively the air-water interface and the R-365mfc vapor-liquid interface. However, dependency of the surface tension on the surrounding pressure is not considered in these studies, only the dependency on temperature is taken into account.

- **Subjectivity in observations plays a large role in flow pattern determination**: The flow patterns that are identified for each study and the transitions between them are dependent on the person performing the experiments and their visual observations. This is due to three main causes. First of all, there are no clear definitions of which flow patterns may occur in two phase-flow in corrugated channels. The amount of identifiable flow patterns and the name or description of each individual one depends on the decisions made by the person reporting on the flow patterns. There may be two similar but slightly different flow patterns which one person considers as the same, but where another person identifies a clear difference. For example, certain authors only report on a single ‘bubbly’ flow regime, where others make a distinction between ‘regular bubbly’ and ‘irregular bubbly’ flow. Also, the ‘annular’ and ‘film’ flow regimes are different names for what is later described in a similar sense
for different studies. Secondly, the transitions or dividing lines between flow regimes introduce an additional role of subjectivity. Observations of flow patterns are only done for a discrete set of data points. When the flow patterns are different for two neighboring data points, a dividing line is usually constructed between these by the author but the exact location of this line is a choice rather than a given fact. As stated before, in the study by Winkelmann (2010) [35] the dividing lines are not provided by the author. It is left to the reader to construct these, either by imagination or physically adding these to the figure. Finally, the third possible cause for subjectivity is that the perception of the observer in determining the flow patterns may be influenced by the imaging conditions. This includes the quality of the lighting, quality of the produced images and the type of transparent covering plate used.

- **Experimental errors in the studies:** The experiments performed in the different studies also introduce errors. Reported values of the pressure and temperature or the superficial velocities of the gas and liquid phases are subject to measurement errors and random errors occurring during the experimental work. Not many authors report sufficiently on the expected magnitude of these errors in the articles of their studies.

- **Errors in the translation of data to newly constructed flow pattern map:** Errors are introduced in reproducing the original flow pattern maps. As described before, the values are not based on data sets, but on the published figures displaying the dividing lines between flow patterns. Values used to construct a line will contain errors. This is a result of the fact that these values are determined based on the corresponding values on axes for the chosen points on that line, and the accuracy of determining the values by this approach is limited. Since all lines are reproduced using several data points and assuming linear relationships between these, the errors introduced for curved lines are larger than those introduced for straight lines.

**Recommendations for future work**

Before quantitative conclusions are drawn from the combined flow pattern map of the different studies, it is important that at least all important geometrical quantities and operating conditions are known for each study, as well as the errors in reported values. Also, all definitions of parameters should be agreed upon clearly before reporting values. This does however not eliminate the important cause for error by subjectivity. This error can only be avoided by setting up a new controlled program of various experimental studies, focused on two-phase flow pattern determination in vertical downflow for the corrugated channels of plate heat exchangers. Ideally, the experimental work is performed by one research institute or group or even one single researcher. A clear agreement on certain points is then needed beforehand, including:

- What are the different flow patterns that have previously been defined and are considered relevant for representation of the two-phase flow in corrugated channels?

- What are the corresponding names for each flow pattern and in which way can each of the flow patterns be described to correspond with the previous observations in a clear and understandable way?
• What are relevant images to be used as reference material that properly represent each of the flow patterns?

• What are other factors that may influence the observers' perception? This includes lighting conditions, imaging quality and the thickness and material of the transparent cover plate.

• At what point does one flow pattern change into another? Is a transition regime observed or is the change between flow patterns abrupt? Are there observable characteristics of the flow that indicate a change of one flow pattern into another?

• Where should dividing lines be placed and what uncertainties should these incorporate, considering the fact that flow pattern determination is only done for a discrete set of measurement points?

To gain insight in the factors determining the two-phase flow patterns, and to construct a generally valid universal flow pattern map, a large amount of experimental work will need to be done. This includes experiments under varying operating temperatures and pressures. Phase composition of the fluids can also be varied, as long as the flow remains two-phase. Different fluids of varying thermophysical properties should be tested experimentally, with a focus on types that are used in practical applications. Mixtures may be used where the components separate into a pure vapor and pure liquid phase, as well as mixtures where either of the phases or both consist of multiple components. The influence of geometrical parameters on a small scale should be varied, such as the corrugation pitch, amplitude and chevron angle. Also large scale parameters such as plate width and height may be varied to study effects on the flow patterns which occur on a larger scale. Given the chosen geometry, fluids and operating conditions, the flow rates or velocities of the phases should be varied between reasonable bounds. Experiments should in the first case be performed under assumed adiabatic and isothermal conditions. However, also results for experiments involving temperature and pressure changes are interesting for comparison, including ones with phase change and resulting compositions changes.

SIMPLIFIED NEW FLOW PATTERN MAP
For now, a simple qualitative flow pattern map may be used as a guide for modelling plate heat exchangers under two-phase flow, taking into account the most important reoccurring observed flow patterns. From the reported studies, these are selected to be the film, bubbly, slug and churn regimes, since these are reported the most frequently. The film flow pattern also includes the 'annular' flow pattern, more unstable 'annular-liquid bridge' and 'wavy' flow patterns and partially wetting 'rivulet' and 'partial film' patterns. The bubbly flow patterns includes both 'regular' and 'irregular' bubbly flow. The flow pattern map, based on the constructed new flow pattern maps of combined studies, is presented in Figure 2.9

2.6. RELEVANT PHYSICAL PHENOMENA
To gain a better understanding of the performance of plate heat exchangers working as condensers or absorbers, the physical phenomena that play a role inside the heat exchangers should be sufficiently understood. In this section important physical phenomena reported in literature are discussed, related to phase change, momentum transfer, mass transfer and heat transfer.
2.6. RELEVANT PHYSICAL PHENOMENA

Figure 2.9: Flow pattern map of two-phase gas-liquid flow in a corrugated channel, using a simplified qualitative impression.

2.6.1. PHASE CHANGE: CONDENSATION MECHANISMS

In Thonon and Bontemps (2002) [69] it is stated that three main mechanisms of condensation exist, which are observed dependent on the Reynolds number:

1. Laminar
2. Wavy film
3. Turbulent film

For pure fluids, filmwise condensation is observed for low Reynolds numbers, and transition to the turbulent regime occurs between Reynolds numbers of 100 and 1000, depending on the operating conditions. Behavior for mixtures is different. García-Cascales et al. (2007) [57] state that two condensation mechanisms can exist in plate heat exchangers:

1. Filmwise
2. Turbulent

The filmwise mechanism corresponds to the mechanism described as laminar in Thonon and Bontemps (2002) [69]. Longo et al. (2015) [51] report on two condensation mechanisms under condensing flow in plate heat exchangers:

1. Gravity controlled
2. Forced convection dominated
As stated before, the transitions from the one mechanism to the other occur for two-phase equivalent Reynolds numbers larger than 1600. Laminar, filmwise or gravity controlled condensation may be considered as the same mechanism, and the turbulent and forced convection mechanisms are also considered as the same.

### 2.6.2. Momentum Transfer: Gravity, Surface Tension, Shear Stress and Inertia

Tovazhnyanski and Kapustenko (1984) [50] discuss that due to the surface tension forces, which are present in condensing flows in plate heat exchangers, a part of the condensate is drawn into the valleys between corrugations. This effect is described in Gregorig (1954) [86]. Gradeck and Lebouche (2000) [87] perform measurements of the wall shear stress under two-phase flow in corrugated channels, showing the flow structure modification from single phase to two-phase flow. The wall shear stress influences both pressure drop and heat transfer, and the mean wall shear stress is compared to that occurring in single phase flows for equal liquid velocities. Effects of the wall shear stress on the flow observed for the different flow patterns in different channels are discussed. Tribbe and Müller-Steinhagen (2001) conclude that shear stress plays an important role both at relative low gas flow rates (bubbly flow regime) and at very high gas flow rates (partial film flow) under two-phase flow in heat exchangers.

Thonon and Bontemps (2002) [69] state that for small channels, surface tension needs to be taken into account. Flow is controlled either by gravitational forces in the gravity dominated regime or by shear forces in the forced convection regime. Heat transfer is closely related to the flow regime or flow pattern. Vlasogiannis et al. (2002) [59] state that wetting characteristics of the plate material, thus the surface tension between the gas, liquid and solid phases, may play a role on the flow regimes for two-phase flow in heat exchangers. Würfel and Ostrowski (2004) [70] state that there are two controlling flow regimes for non-adiabatic condensation processes within plate heat exchangers of the herringbone type:

- Gravity: at low mass flux, the vapor phase has a small influence on the hydrodynamics. Calculation of heat transfer in the condensate can be done through the Nusselt laminar film theory.

- Shear stress: at high mass flux, significant interaction exists between vapor and liquid phases. Conditions here are considered turbulent.

Asano et al. (2005) [88] state that separated downward two-phase air-water flow in plate heat exchangers might be based on the least energy dissipation by friction and gravity. Also inertial forces are observed to play a role in the flows. García-Cascales et al. (2007) [57] state that the main phenomena affecting the flow pattern in condensation in plate heat exchangers are gravity and shear stress. Longo et al. (2015) [51] report that dependent on the condensation mechanism, either gravity or both gravity and shear play a role in condensing refrigerant flows.

### 2.6.3. Heat Transfer and Mass Transfer: Diffusion and Convection

Tovazhnyanski and Kapustenko (1984) [50] discuss the limitations of mass transfer during condensation processes in plate heat exchangers. If a noncondensable gas is present
in the vapor, the condensation process becomes complicated due to the diffusive resistance to vapor transport from the flow core to the condensate film surface. Panchal and Kuru (1997) [89] find that vapor-phase diffusion is a controlling mechanism for the condensation of binary ammonia-water mixtures on a vertical tube. Analytical predictions are given for three extreme cases of mass transfer conditions:

1. Vapor and liquid phases are in equilibrium.

2. Perfect-mixing condition for the liquid phase.

3. No-mixing condition for the liquid phase.

Mass transfer resistance in binary vapor mixtures can be represented by the Colburn-Drew analysis, based on the film theory. The results may however only be valid for certain condenser configurations and operating conditions.

Thonon and Bontemps (2002) [69] report that for mixtures mass transfer affects the heat transfer up to a factor 4 in the heat transfer coefficient, mainly for low Reynolds numbers. It is stated that a predictive model is needed for condensation, in which diffusion is taken into account. The role of mass transfer on heat transfer for mixtures is described, where the condensing vapor needs to diffuse through the gas to the interface. In the same study it is reported that the heat transfer performance is closely linked to the resulting flow patterns in the channels. In the laminar regime mass transfer in the vapor limits the heat transfer, since species need to diffuse to the interface first before moving across the interface during condensation, and in the turbulent regime heat transfer is controlled by the liquid film. For an increasing Reynolds number, the mass transfer resistance decreases and the heat transfer resistance increases, as presented in Figure 17 of Thonon and Bontemps (2002) [69]. The mass transfer resistance is presented in the same units as the thermal resistance \((m^2K/W)\), through the application of assumed proportionality between the mass transfer resistance and the heat transfer resistance in the vapor phase. More details of how this assumption is applied are found in the publication [69].

Jung et al. (2014) [58] state that for ammonia-water absorption in a plate heat exchanger, the heat transfer coefficient of the solution side increases for an increasing aspect ratio of the plate length to the plate spacing \((L_p/d_g)\), but the aspect ratio of the plate width to channel spacing \((B_p/d_g)\) has little effect. Longo et al. (2015) [51] state that for refrigerant condensation in plate heat exchangers, heat transfer coefficients are independent of mass flux in the gravity-dominated regime but a dependency arises in the forced convection condensation regime. For superheated vapor, corrections are made in the heat transfer coefficient relations taking into account the effect of mass transfer on sensible heat transfer between the superheated vapor and the condensate interface.

**2.7. MODELLING APPROACHES**

Numerical modelling approaches for heat exchangers are discussed, divided into steady-state and general approaches. Some modelling approaches are specific for plate heat exchangers, others for heat exchangers or fluid flow systems in general. For each approach, the focus is described along with the applied method, and in certain cases the important assumptions are listed. After the review of modelling approaches in literature, the modelling approach of this work is discussed in Chapter 3.
2.7.1. STEADY-STATE
Classical methods such as the LMTD and $\eta$-NTU method are based on steady, single phase flow with constant fluid properties and heat transfer. Longitudinal wall conduction is neglected, and a uniform distribution of flow is assumed. For most cases this approach is oversimplified, as stated by Corberán et al. (2001) [90].

Infante Ferreira et al. (1984) [79] describe a finite element model for the simultaneous heat and mass transfer processes in vertical tubular bubble absorbers. Only overall conditions are considered in the model, which is constructed by heat and mass balances based on conservation laws. The following assumptions are made:

- Heat and mass transfer coefficients are constant along the absorber tube.
- The gas and liquid flows are homogeneous, i.e. the concentrations of ammonia are uniform in each phase.
- Simultaneous heat and mass balances need to be solved.
- Only overall conditions are considered in the end, not the profiles along the tube.
- The wall friction term is negligible when compared to the hydrostatic pressure drop caused by the weight of the gas and liquid.
- For a differential element of height $\Delta z$ parameters are assumed constant.
- At each difference element both the gas and liquid phases are assumed to be at the same mean temperature.

The procedure aims to calculate the absorber height and the amount of heat to be transferred to the coolant. Predicted parameters are compared to experiments.

Corberán et al. (2000) [91] describe a model for compact evaporators and condensers, where the 2-D flow along the channel is split into separated 1-D paths. Every 1-D flow path is then discretized into as many elements as required. Conservation equations for single or two-phase flow are integrated, and calculations are performed using an explicit finite volume scheme. Temperature fields and wall temperatures are derived in consecutive iteration steps. Certain assumptions are made:

- Two-phase flow with phase change occurs.
- The flow is steady.
- The flow is described by the annular flow pattern.
- Thermodynamic equilibrium is assumed.

An empirical relationship is used to account for the void fraction and slip ratio. A balance for the heat exchanged with the surrounding fluids is set up, neglecting the effect of longitudinal conduction. Boundary conditions are given by the pressure and temperature at the inlet, and by the adiabatic condition for the outer plates. A semiexplicit method for wall temperature linked equations (SEWTLE) is described, which is a general finite volume technique for calculations of complex heat exchangers. SEWTLE is used as a solution method,
with a simple but effective discretization for the momentum equation. An iterative procedure is started which allows adequate evaluation of the thermophysical properties of the fluids, and of the friction factor and the heat transfer coefficient at every cell. Corberán et al. (2001) [90] use the SEWTE method, where the heat exchanger is discretized as well as the governing equations. A solution strategy is proposed which solves the equations iteratively. The method is flexible, it can be applied to different flow arrangements and geometries. The method can also be applied for the consideration of multiple streams or for the local evaluation of fluid properties, the friction factor and heat transfer coefficient. The method shows good accuracy, high robustness and fast computational time.

Tovazhnyansky et al. (2004) [92] describe a numerical simulation using semi-empirical equations of heat and mass transfer for condensation of multicomponent mixtures in plate heat exchangers, containing non-condensable components. The presence of several components in the mixture makes the simulation process more complicated, because of possible simultaneous permeability of the vapor-condensate interface with all components. The diffusion process is thus very significant for multicomponent mixtures. When it comes to heat transfer, several thermal resistances are identified:

1. Vapor phase
2. Condensate
3. Film fouling
4. Wall
5. Coolant

The rate of heat transfer significantly depends on diffusion interactions, since a high concentration of non-condensable components creates an additional thermal resistance. To simplify the description of the simultaneous heat and mass transfer in vapor (gaseous) and liquid phases, equalization of temperatures and concentrations in turbulent bulk due to intensive mixing is supposed. Certain important assumptions are made and are listed here:

1. The model is one-dimensional.
2. The flows are turbulent.
3. The presence of gradients of concentrations, temperatures, and velocities is assumed to occur in a thin boundary layer at the interface, and for the temperatures and velocities at the wall as well.
4. Data correlations are used for the condensation of pure vapor and binary vapor-noncondensable gas mixtures.

Heat and mass balances are set up, and the resulting system of equations is solved with a prediction-correction type method. Non-ideality of the mixture components is accounted for with coefficients of activity, and changes of local parameters are taken into account.

Fernández-Seara et al. (2005) [81] describe a differential mathematical model on the basis of mass, concentration and energy balances and heat and mass transfer equations for the absorption of ammonia into water in a co-current vertical tubular absorber. The model
takes into account the different predicted flow patterns separately, and solves the resulting set of nonlinear differential equations together using the finite difference method. An algorithm using four iterative loops is described, which solves the equations from the absorber bottom to top using Fortran 95. Simultaneous heat and mass transfer in both phases are taken into account, as well as heat transfer to the cooling medium. Proper boundary conditions are derived for a differential control volume and certain assumptions are made:

- State equations and equilibrium properties are taken from Ziegler and Trepp (1984).
- The absorption process is in steady-state and the absorber pressure is constant.
- Heat losses to the environment are negligible.
- The two-film and the Lewis and Whitman theory of non-interfacial resistance are applied, i.e., the interface concentrations of vapor and liquid are the equilibrium concentrations at the interface temperature.
- The heat and mass transfer areas between liquid and vapor phases are equal.
- Bubble coalescence and break-up are not considered.
- There is no direct heat transfer between the vapor and the coolant.
- For a transversal tube section, the coolant, liquid and vapor properties are assumed to be constant with the angular coordinate.

Mass transfer equations result from the combined contribution of molecular diffusion and bulk transport of material through the interface, taking into account the mass continuity requirement. Heat transfer equations are given for the two regions where the heat transfer takes place:

1. Between the vapor and liquid phase.
2. Between the liquid phase and the coolant through the tube wall.

Due to the combined heat and mass transfer in the liquid and vapor phases, there is a difference in sensible heat between the bulk and interface of the phases that is affected by the mass flux. The hydrodynamics within the absorber are characterized by a changing two-phase flow, since the mass fractions of the phases change along the length of the absorber. The heat and mass transfer coefficients, the specific interfacial area and the two-phase flow void fraction depend on the flow pattern. A detailed knowledge of the two-phase flow behavior is thus required.

Sieres and Fernández-Seara (2007) [78] describe a general differential mathematical model to analyze the simultaneous heat and mass transfer processes that occur in different components of an ammonia–water absorption system. A finite difference numerical method is used to solve the resulting set of nonlinear differential equations, proposing an iterative algorithm for its solution. Several assumptions are made in the model based on the application of mass and energy balances and heat and mass transfer equations to a differential control volume:

1. The processes are in steady-state and the flow is 1-D.
2. The pressure is constant.

3. The kinetic, potential and mechanical energy variations are neglected.

4. There is no heat transfer by radiation or heat losses to the environment.

5. No chemical reactions are considered, thus the total number of moles are conserved.

6. The heat and mass transfer areas are equal.

7. The Lewis and Whitman theory of non-interfacial resistance is applied; i.e., the interface concentrations of vapor and liquid are the equilibrium concentrations at the interface temperature.

8. The film theory is applied to the heat and mass transfer equations; i.e., the heat and mass transfer resistances are confined to a thin region close to the interface.

9. The heat and mass transfer occurs only in the transversal direction to vapor and liquid flows.

10. There is no direct heat transfer between the vapor and the external fluid.

The corresponding mass and heat transfer equations are given, along with the overall mass and heat balances. These are solved together with appropriate interface and boundary conditions. The model can be used for simulation, quantification of performance, or to perform parametric and optimization analyses.

Winkelmann (2010) [35] uses the boundary element method (BEM) to solve the inverse heat conduction problem required to obtain the temperature profile and wall heat flux profile of condensing flow in corrugated aluminum plates. A numerical method is used, since the complexity of the geometry prevents an analytical solution. Certain assumptions are made related to different topics:

- The Heat Conduction Problem

  1. The heat conduction problem is treated as 2-D because the change of temperature over the width of the plate is much smaller than the change of temperature over the thickness and over the length of the plate.

   2. The thermal conductivity of the aluminum plate is isotropic.

   3. The uninstrumented plate behaves in the same way as the instrumented plate. The total heat flow rate is twice the heat flow of the instrumented plate.

   4. The average between the two temperature and heat flux profiles is equal to the average over the whole width of the plate.

   5. The temperature profile along the path of the cooling water channel is neglected, meaning a certain profile is assumed based on the known inlet and outlet values.

   6. The heat transfer coefficient for the cold water channels is calculated from the Gnielinski correlation using a channel specific friction factor correlation.

   7. The influence of the bends on the heat transfer coefficient for the cold water channels is neglected.
8. Insulation is perfect, no heat losses to the environment.

- The Operating Conditions
  1. All experiments are conducted at steady-state.
  2. The vapor is slightly superheated at the inlet of the test section, the quality at the inlet is equal to one.
  3. After the desuperheating the temperature profile of the condensing fluid is linear.
  4. The pressure profile in the hot channel is linear.
  5. Incompressible flow.

- The fluid is a Newtonian fluid.

Arsenyeva et al. (2011) [71] describe 1-D modelling of condensing flow in plate heat exchangers, using the analogy between heat and momentum transfer, taking into account variations in properties along the length of the channel. The model aims to predict local film heat transfer coefficients and friction factors, and was validated using experimental data. Cerezo et al. (2011) [15] discuss a mathematical model of bubble absorbers, using constant absorber control volume increments. The model is based on several assumptions:

- The absorption process takes place at steady-state conditions.
- The vapor and liquid phases are in equilibrium at the interface zone.
- The bubble velocity is constant.
- The vapor bubble has a spherical shape.
- There is no break-up and no interaction or coalescence amongst bubbles.

Using these assumptions, mass and energy balances are given for the control volume considered and over the interface of the two phases.

Le Lostec et al. (2013) [40] describe a numerical model of an absorption chiller operating with ammonia-water as working fluid. It is claimed that a detailed numerical model is required to take into account mass transfer coefficients when dealing with mixtures. The steady-state model is based on heat and mass balances for each component, including heat transfer, mass transfer in the vapor and liquid phases and the condensation of a binary vapor. The mass, species and energy conservation equations were implemented into a Fortran code, using the following assumptions:

- Steady-state.
- Negligible pressure drops.
- Negligible heat losses to the environment.
- Negligible leaks to the environment.
- Saturated vapor at the outlet of the purification column.
Different studies are used to evaluate correlations for the thermodynamic and thermophysical properties of the ammonia-water and water-ethylene glycol mixtures. The condenser is modelled as a plate heat exchanger consisting of two different zones:

1. The condensation zone: liquid and vapor phases are assumed to coexist at thermodynamic equilibrium.

2. The subcooling zone: for an estimated condensation pressure values are calculated for certain parameters in both zones, and ultimately the area of the heat exchanger.

Longo et al. (2015) [51] discuss a model for condensing refrigerant flows in plate heat exchangers. The local heat transfer coefficient is integrated along the heat transfer area using a finite difference approach to compute the average condensation heat transfer coefficient referred to the projected area of the plates. To allow this, the heat exchanger was divided into 100 parts and a heat transfer correlation was applied starting from the vapor inlet port with a step by step procedure adjusting vapor quality one step to the successive.

2.7.2. General

Tovazhnyanski and Kapustenko (1984) [50] attempt to model the condensation process in a plate heat exchanger by dividing the heat exchanger into three zones, to allow taking into account the change of process parameters along the channel as the vapor is condensing. The following three zones are considered:

1. The inlet port zone which serves to admit the flow and to distribute it uniformly along the channel width.

2. The corrugated operating field, where the basic heat transfer process takes place.

3. The outlet port zone.

To take important effects into account, a stepwise calculation for small channel increments is required, for instance by using a modified version of the Colburn-Hougen approach. Heat and mass transfer analogy is used to predict heat and mass transfer coefficients, using these of a single-phase flow.

Kreissig and Müller-Steinhagen (1992) [56] attempt to estimate the pressure drop for two-phase flows in plate heat exchangers. Several assumptions are made to derive the acceleration and hydrostatic pressure losses inside the heat exchangers, of which some may also be relevant for modelling flows of fluids mixtures undergoing phase change:

1. The velocity distribution over the cross section of the channel is uniform, which is a rough simplification.

2. The frictional pressure losses in the connectors and the ports are negligible.

3. There is no deformation of the plates due to the pressure difference between the two sides of the heat exchanger and thus no change in channel gap.

4. Evaporation of water into air and dissolution of air into water are negligible (only valid for flow without relevant phase change).

5. The liquid phase is incompressible.
6. No frictional pressure losses occur in the connecting lines to the pressure transducer.

Grabenstein and Kabelac (2012) [61] base a model on different flow patterns for condensation or general two-phase flow in plate heat exchangers. Heat transfer and pressure drop relations are suggested for the different observed flow patterns, including the film flow pattern. An adopted model of the filmwise condensation of Nusselt is suggested for this flow pattern, which should be expanded for the effects of increasing turbulence in the condensate film and the wave formation of the film surface due to the fast streaming gas phase.
3

MATHEMATICAL MODEL

The actual model that is implemented into a Matlab code is discussed in this chapter. Findings from the literature study on modelling presented in Chapter 2.7 are taken into account, along with other literature relevant to the model reviewed in Chapter 2. In section 3.1 the general modelling approach is discussed, before moving into the assumptions made in section 3.2. In sections 3.3, 3.4 and 3.5 respectively the implemented pressure drop, heat transfer and mass transfer correlations are discussed. Finally, in section 3.6, a discussion of the solution procedure in the model is presented.

3.1. MODELLING APPROACH

In this subsection, first the focus of the model is made clear, before moving into the basic fundamentals, required input and produced output of the model.

3.1.1. FOCUS

The proposed model is developed for plate heat exchangers functioning as absorbers or condensers of specific geometry: corrugated chevron plates of symmetrical chevron angles and sinusoidal corrugations. The model assumes the plates are placed next to one another with their surfaces in contact, with the orientation of the corrugation pattern interchanged between each consecutive plate. The main orientation of the flow in the heat exchanger is vertical and the flows of working fluid and cooling water are modelled to be in opposite direction, where working fluid moves downward through the channels. Schematic representations of the plate heat exchanger geometry, including the main dimensions and flow structure are found in figures 3.1, 3.2 and 3.3.

At the working fluid side, the model works for two-phase vapor-liquid flow, but also for single phase flow of either a pure vapor or liquid. In the case of pure vapor flow, the model accounts for superheating, and in the case of pure liquid flow, the model accounts for subcooling. Phase change of the working fluid throughout the heat exchanger is accounted for, which in this case is either absorption or condensation. The focus is on ammonia-water as the working fluid, with high ammonia fractions up to nearly pure ammonia. The model is constructed in a way that it is easily adaptable to use with other binary mixtures than ammonia-water, but the actual implementation of this is outside the scope of this work. At the moment, only relations relevant for the ammonia-water mixture of varying composition are implemented in the model.
Figure 3.1: Representation of the working fluid and cooling water channels in the heat exchanger, and single plate geometry. Indicated dimensions are heat exchanger width of the stacked plates ($W_p$), plate width ($B_p$) and plate length ($L_p$). Figures are originally published in Le Lostec et al. (2013) [40] (left) and Tsai et al. (2009) [93] (right).

Figure 3.2: Representation of the single plate detailed geometry. Indicated parameters and dimensions are heat exchanger plate area and enlarged plate area ($A_0$ and $A_p$), channel gap ($d_g$), corrugation pitch ($p$) and amplitude ($\hat{a}$). Figures are originally published in Tsai et al. (2009) [93] (left and top right) and Winkelmann (2010) [35] (bottom right).
3.1. MODELLING APPROACH

Figure 3.3: Representation of the orientation of the working fluid and cooling water flows, where the red box encompasses the entire region that is modelled. The figure on the left is originally published in Le Lostec et al. (2013) [40].

3.1.2. BASICS

The model is based on conservation equations of energy (heat) and mass (general and species), and more about the balances is discussed further ahead in this section. Conservation of momentum is not directly implemented into the model, but taken into account through the assumption of a flow pattern, which is discussed in the next section. The conservation equations are complemented by transfer equations of heat, mass and momentum (pressure drop). These may be analytically derived or (semi-)empirical, including effects of phase change. The transfer equations are discussed in the sections further ahead in this chapter.

The model assumes steady-state operation of the heat exchanger. The finite volume method is used to discretize the heat exchanger into a set amount of control volumes along its length, which are not necessarily of equal size. Across the flow direction, a single control volume encompasses half the working fluid channel, half the cooling water channel and the wall separating the two flows. Only half the channel widths are taken into account, because the flows and transfer processes are assumed to be symmetrical on either side. The cooling water flow is considered to always be single-phase, where the working fluid can contain either or both of a vapor and liquid condensate. At the inlet, under typical conditions the flow will be two-phase for the ammonia-water mixture, and pure vapor flow when pure ammonia is used. At the outlet, the flow may either be a two-phase vapor-liquid flow or pure liquid, depending on whether or not the model predicts full condensation at some point along the heat exchanger. This type of discretization can be classified as 1½-D. The actual discretization only occurs in one direction, but different regions are defined within the control volumes itself, each with their own corresponding relations.
3.1.3. INPUT AND OUTPUT

The model predicts the values of a large set of parameters in each control volume, and depending on the parameter, in different regions within the control volume. These regions are:

- The cooling water bulk
- The cooling water near the wall
- The wall edges at the cooling water side and working fluid side
- The working fluid liquid phase near the wall and near the vapor-liquid interface
- The working fluid liquid bulk
- The working fluid phase interface
- The working fluid vapor phase near the interface
- The working fluid vapor bulk

In case that there is no liquid phase flow in a certain section of the heat exchanger channel, the vapor phase is in direct contact with the wall and the heat transfer between the wall and working fluid occurs directly to the vapor phase.

For experimental validation of the model, the current setup only allows values of certain parameters to be measured at the inlet and outlet of the heat exchanger, not within the heat exchanger itself. Measurements at the inlet and outlet of both streams are done for the temperature and pressure. The mass flow rate of either stream is measured at a certain point in the cycle for the specific fluid. The same is the case for the composition of the working fluid stream, but this can also be determined during filling of the system with the working fluid. The cooling water stream composition is assumed to be known. All other parameters at the inlet, within the heat exchanger itself, and at the outlet are calculated using the inlet values. The inlet and outlet value of the cooling water temperature are used to compute initial estimate of the temperature profile, which is used in the first step of the implemented iterative loop in the model. In case the model is not used for comparison with an experiment, but instead to predict the performance of the heat exchanger for a certain set of conditions, the initial cooling water outlet temperature can be based on an educated guess. A better guess should lead to faster convergence of the model. Validation of the model is performed by comparing the measured outlet values with the values predicted by the model, and certain overall values based on calculations for both the model and experiments discussed further ahead.

Besides the measured inlet and outlet conditions, geometrical details of the heat exchanger are given as input to the model. The working fluid and cooling water composition are specified, and together with the operating conditions of temperature and pressure the fluid properties are calculated through the methods discussed in Appendix A, section A.2. Using the conservation and transfer equations for each control volume, profiles of many parameters such as the vapor quality, temperatures, heat and mass transfer coefficients are computed along the length of the heat exchanger.
3.2. ASSUMPTIONS

Certain assumptions in the model have already been discussed in the previous section. This includes the assumption about the detailed geometry of the plates, symmetry of the flow in the channels, dimensionality of the model (1-D) and time-dependency (steady-state). Other assumptions need to be made to further specify the model, related to:

- Relevant physical effects (surface tension, gravity, inertia, shear stress)
- Flow regime (laminar or turbulent)
- Flow pattern (single phase or two-phase: film, bubbly, slug, churn)
- Thermodynamic condition at the interface (equilibrium or non-equilibrium)
- Pressure drop (correlations to be applied)
- Heat transfer (correlations to be applied)
- Mass transfer (correlations to be applied)
- Numerical approach

Related to the first three points, surface tension is not taken into account in the model, and the flow is assumed to be a gravity controlled laminar condensing film flow. This assumption is supported by different observations. First of all, the main flow direction is downwards and gravity plays an important role in driving the flow. Next to that, for typical operating conditions the relative effect of surface tension to the other forces is expected to be small enough to make it negligible. This assumption should be checked by calculating the Weber, Eötvös and Capillary numbers, for which the general relations are found in section A.1 and specific two-phase relations in subsection 2.5.3. Also, the Reynolds number of the vapor and liquid phases, or the two-phase Reynolds number, is expected to be small enough to assume the flow is within the laminar regime. This implies that the viscous forces are larger than the inertial forces.

Finally, for typical operating conditions the operating line of the condenser lies within the bounds where in literature mostly film flow or similar flow structures are reported. This corresponds to the region in the top left part of the graphs presented in Figures 2.8 and 2.9. Under certain conditions, also bubbly flow is expected to occur in the working fluid flow. This will however be limited to a very small section, where the two-phase flow is almost fully condensed. A transition between the two-phase vapor-liquid film flow and the single phase pure liquid flow will then occur. Since it is uncertain whether this transition will occur, and in case it does occur the expected impact is limited, the bubbly flow regime is neglected in the model. Churn flow is not expected to occur, but if it would it could be argued that it is possible to model the flow in a similar way to the film flow, as stated in Fernández-Seara et al. (2005) [81].

Related to the fourth point, the working fluid vapor-liquid interface is assumed to be at thermodynamic equilibrium. This allows computation of phase compositions and certain fluid properties at the interface, of which the first is essential to determine the driving force for mass transfer across the interface. This assumption is supported by a large amount of studies [15, 40, 78, 81, 89, 91].
Two other assumptions serve mostly as simplifications, and to justify the fact that only a single set of neighbouring channels for the working fluid and cooling water is modelled. Other sets of channels are assumed identical, by requiring that no heat losses occur to the environment and the distribution of flow is equal amongst all channels. The first assumption is more valid for channels more towards the center of the heat exchanger. Further away, near the edges of the heat exchanger, heat can be directly exchanged with the environment. The general validity of this assumption depends on how well the actual heat exchanger that is being modelled is insulated, but the absolute amount of heat transferred between the fluids should generally be larger than the heat transferred to the environment. The outer walls are thus considered to be adiabatic. The distribution of flows amongst the channels depends on the exact design of the flow distribution section, but generally this section is designed to divide the flow as evenly as possible between the fluid channels. Another assumption that is made is that there is no significant heat conduction in the direction of the flow, which accounts both for the flowing fluids and the dividing wall. Diffusive mass transfer in this direction is also neglected. Summarizing, the most important assumptions for the model are:

- Steady-state flow and absorption process
- Laminar condensing film flow, gravity controlled
- Phase interface at thermodynamic equilibrium
- No heat losses to environment, adiabatic outer plates
- Equal distribution of flow amongst channels
- No conduction of heat or diffusion of mass in the flow direction

The assumptions correspond to all those made in Corberán et al. (2000) [91], and to most assumptions made in Fernández-Seara et al. (2005) [81]. Differences with the last study are that the geometry is different, other fluid property calculation methods are used and bubble flow is not considered to occur in the model presented in this work. To elaborate further on the first assumption, for each control volume in the model a flow pattern is assumed. In case of single-phase flow only one flow pattern is possible, since there is only one phase to distribute over the channel. Single-phase vapor flow may occur under typical conditions when pure ammonia is used as working fluid, or when ammonia-water as the working fluid is heated above typical OTEC conditions at the inlet. Single-phase liquid flow occurs when the flow is fully condensed. Whether this will be the case depends on the predicted heat transfer across the wall, and from the liquid phase through the interface to the vapor phase.

Two-phase flow usually occurs at the inlet and in a larger part of the heat exchanger. When two-phase flow occurs, the assumed flow pattern is always film flow. Bubly flow or other flow patterns may occur but this is neglected as discussed earlier in this section. The main sections along the flow direction in the heat exchanger are represented in Figure 3.4. In this figure, the red boxes together encompass the entire region that is modelled, where the top one indicates the two-phase vapor-liquid film flow region and the bottom one the single phase liquid flow region. The last region does not occur under all circumstances,
3.3. Pressure Drop

Studying the most suitable pressure drop relations to be used for the different streams is not the main focus of this work. However, it is important to implement pressure drop relations hence the question mark in the figure. The second red box indicates the two-phase bubbly flow region. A schematic representation of the region in the two-phase film flow section and of certain important physical parameters is given in Figure 3.5. The methods for calculating all relevant parameters in the control volume are presented in the upcoming sections.

Figure 3.4: Representation of the different flow patterns that are modelled for the working fluid flow along the channel length. Two-phase film flow always occurs at the inlet, bubbly flow is not considered in the model and if single phase subcooled liquid flow occurs depends on the predictions of the model.

Figure 3.5: Representation of the regions within a single control volume of the heat exchanger. This encompasses the cooling water liquid phase, wall and the working fluid liquid and vapor phases. Certain parameters are indicated at different positions such as the temperature \( T \), heat flow \( Q \), interface and wall area \( A \), mass flow rate \( \dot{m} \), liquid and vapor phase fraction \( x \) and \( y \) and specific enthalpy \( h \).

3.3. Pressure Drop

Studying the most suitable pressure drop relations to be used for the different streams is not the main focus of this work. However, it is important to implement pressure drop relations...
into the model and study their magnitude. For significant pressure drops in the fluid flow, other parameters will be affected. Also, the assumed thermodynamic equilibrium at the vapor-liquid interface of the working fluid flow is affected by the pressure value. In total, four contributions are taken into account for the pressure drop:

- Losses in the inlet and outlet ports and distribution sections: $\Delta P_{\text{port}}$
- Losses due to friction in the main flow channels: $\Delta P_{f}$
- Losses due to deceleration of the flow as a consequence of phase change, relevant in the two-phase absorption/condensation section: $\Delta P_{\text{dec}}$
- Losses due to elevation change over the heat exchanger channels: $\Delta P_{\text{ele}}$

Since the flow pattern for the two-phase flow is based on an assumption rather than a derivation of the hydrodynamics from the basic Navier-Stokes or other conservation of momentum equations, the frictional pressure drop relations are not derived analytically but are based on empirical relations.

### 3.3.1. INLET AND OUTLET LOSSES

During the experiments values of the pressure are measured in the inlet and outlet lines of the working fluid and cooling water. There is a pressure change between the positions where the pressure is measured in the lines and the actual inlet and outlet of the fluid channels within the heat exchanger due to pressure losses in the inlet and outlet manifolds and ports. The losses are estimated as 1.5 times the head due to the flow expansion at the channel inlet, as suggested by Yan et al. (1999) [68]:

$$\Delta P_{\text{port}} \approx 1.5 \left(\frac{u_{\text{act},m}^2}{2v_m}\right)$$

Here, $u_{\text{act},m}$ is the mean actual velocity of the fluid single or two-phase flow, considering equal liquid and vapor velocities in the case of two-phase flow. $v_m$ is the mean specific volume of the fluid, which for the two-phase mixture is calculated from the individual specific volumes of the phases ($v_V$ and $v_L$) and the mean vapor quality of the mixture ($X_m$):

$$v_m = X_m v_V + (1 - X_m) v_L$$

The mean specific volumes of the phases are given as the reciprocal of the phase densities:

$$v_V = \frac{1}{\rho_V}$$

$$v_L = \frac{1}{\rho_L}$$

The actual mean velocity is calculated by multiplying the total mass flux through the ports ($j_{\text{port}}$) with the mean specific volume:

$$u_{\text{act},m} = j_{\text{port}} v_m$$

The mass flux is obtained from the known port diameter ($d_{\text{port}}$) and total mass flow ($\dot{m}$) of each stream:
3.3. PRESSURE DROP

\[ j_{\text{port}} = \frac{m}{(\pi/4)d_{\text{port}}^2} \]  

(3.6)

The term in the denominator represents the cross-sectional area of the port.

3.3.2. FRICTIONAL LOSSES

The losses due to friction inside the flow channels are calculated for each control volume. Friction losses occur due to shear forces within the phase, at the interface between phases and for the phases in contact with the walls. Generally speaking, the frictional pressure loss over the entire heat exchanger is a function of the Fanning friction factor \( f \) and several other parameters:

\[ \Delta P_f = \frac{2fL_p j^2}{\rho_m d_h} = \frac{2fL_p j^2 v_m}{d_h} \]  

(3.7)

Here, \( L_p \) is the effective plate length which in this case is the port-to-port length of the flow channels. \( \rho_m \) is the mean or effective density, which continuously changes for two-phase flow. The value is calculated from the mean specific volume:

\[ \rho_m = \frac{1}{v_m} \]  

(3.8)

Since the mean density continuously changes throughout the flow channel, the frictional pressure drop is calculated separately for each control volume of length \( \Delta z \):

\[ \Delta P_{f,\text{CV}} = \frac{2f\Delta z j^2 v_m}{d_h} \]  

(3.9)

TWO-PHASE FLOW

Even though the pressure drop for two-phase flow is assumed negligible in the model, the pressure drop relations are implemented to check if this assumption is valid. For two-phase condensing flow, the friction factor is calculated using the relations proposed in Yan et al. (1999) [68]. The relation for the friction factor is given as:

\[ f_{TP} Re^{0.4} Bo^{-0.5} \left( \frac{P}{P_{\text{crit}}} \right)^{-0.8} = 94.75 Re_{\text{eq}}^{-0.0467} \]  

(3.10)

Here, \( Re \) is the liquid phase Reynolds number, \( Re_{\text{eq}} \) is the equivalent Reynolds number for two-phase flow, and \( Bo \) is the Boiling number. The relation is valid for \( Re_{\text{eq}} < 6000 \). The absolute pressure \( (P) \) is known, and the critical pressure \( (P_{\text{crit}}) \) is calculated for the ammonia-water mixture based on the relations by Conde (2006) [36]. The equivalent Reynolds number is defined as follows:

\[ Re_{\text{eq}} = \frac{j_{\text{eq}} d_h}{\mu_L} \]  

(3.11)

The equivalent mass flux \( j_{\text{eq}} \) depends on the phase densities and mixture mean vapor quality:

\[ j_{\text{eq}} = j \left[ 1 - X_m + X_m \left( \frac{\rho_L}{\rho_V} \right)^{1/2} \right] \]  

(3.12)
The Boiling number is a function of the average imposed heat flux across the wall \( q_w \), the bulk specific enthalpy of vaporization \( \Delta h_{vap,b} \) and the mass flux:

\[
Bo = \frac{q_w}{j\Delta h_{vap,b}}
\]

(3.13)

The specific enthalpy of vaporization is given by the difference in bulk specific enthalpies of the two phases:

\[
\Delta h_{vap,b} = h_{V,b} - h_{L,b}
\]

(3.14)

The heat average imposed heat flux across the wall in a single control volume is given as the calculated heat flow across the wall \( Q_w \) divided by the wall area within the control volume \( A_w \):

\[
q_w = \frac{Q_w}{A_w} = \frac{Q_w}{\Gamma \Delta z B_p}
\]

(3.15)

**Single phase flow**

In case of single-phase flow, the friction factor is calculated using the relations proposed for plate heat exchangers in Stephan *et al.* (2010) [8]. A correlation for the Darcy friction factor \( \xi \) and the Fanning friction factor \( f \) for flow in corrugated channels is obtained by rearranging Equation 3.7, and by writing the mass flux as a product of the actual velocity \( u_{act} \) and single-phase density \( \rho \):

\[
\xi = \frac{2\Delta P_f d_h}{\rho u_{act}^2 L_p} = 4f
\]

(3.16)

\[
j = u_{act} \rho
\]

(3.17)

This can again be rewritten to the differential form of the single phase pressure drop, with \( \xi = f(Re) \) (Tribbe and Müller-Steinhagen (2001) [63]):

\[
\frac{dP_f}{dz} = \xi \frac{1}{2} \frac{\rho v^2}{d_h}
\]

(3.18)

The chevron angle \( \varphi \) is an important factor in determining the pressure drop inside plate heat exchangers. When two plates of corrugation angles \( \varphi_1 \) and \( \varphi_2 \) are used, the effective corrugation angle is expressed as:

\[
\varphi = \frac{1}{2}(\varphi_1 + \varphi_2)
\]

(3.19)

However, in this work only plates of symmetrical and equal chevron angle are considered. Two limiting cases can be identified for the value of the chevron angle. Limiting case 0 implies smooth longitudinal flow, and \( \varphi_{flow} = 0^\circ \).

For \( Re < 2000 \) (laminar flow):

\[
\xi_0 = \frac{C_0}{Re}
\]

(3.20)

For \( Re \geq 2000 \) (turbulent flow):
\[ \xi_0 = (1.8 \log (Re) - 1.5)^{-2} \] 

The constant \( C_0 \approx 64 \). Limiting case 1 corresponds to \( \phi_{flow} = 90^\circ \) and wavy longitudinal flow. When both plates are in phase, according to Focke et al. (1985) [94], for \( Re < 2000 \) (laminar flow):

\[ \xi_{1,0} = \frac{C_1}{Re} + C_2 \] (3.22)

For \( Re \geq 2000 \) (turbulent flow):

\[ \xi_{1,0} = \frac{C_3}{Re^{C_4}} \] (3.23)

For \( \Lambda/\hat{a} = 4 \) the constants are: \( C_1 = 597, C_2 = 3.85, C_3 = 39 \) and \( C_4 = 0.289 \). When the wave patterns are shifted by \( \pi = 180^\circ \), the plates are in contact over the whole width at certain points:

\[ \xi_{1,\pi} \to \infty \] (3.24)

When the plates are not exactly in phase, nor shifted by exactly \( 180^\circ \), an empirical factor is used for multiplication of the friction factor for the in-phase situation (\( 1 \leq C_5 < \infty \)):

\[ \xi_1(Re) = C_5 \cdot \xi_{1,0}(Re) \] (3.25)

For crossed corrugation patterns of varying chevron angle \( (0^\circ < \phi_{flow} < 90^\circ) \), two types of flow within the heat exchanger may be identified:

- Small angles \( \phi_{flow} \): crossing flow (low pressure drop)
- Larger angles \( \phi_{flow} \): wavy longitudinal flow (high pressure drop)

Different effects are taken into account to derive the new friction factor \( \xi = f(\phi, Re) \):

- Longer flow path: \( 1/\cos(\phi_{flow}) \)
- Flow is reversed at the edges of the corrugation pattern: \( C_6 \tan(\phi_{flow}) \)
- Crossing of the streamlets, meaning the small substreams caused by the corrugations: \( C_7 \sin(\phi_{flow}) \)
- Smooth change of crossing flow into longitudinal flow

The overall friction factor is then given by:

\[ \frac{1}{\sqrt{\xi}} = \frac{\cos(\phi_{flow})}{\sqrt{C_6 \tan(\phi_{flow}) + C_7 \sin(\phi_{flow}) + \xi_0(Re)/\cos(\phi_{flow})}} + \frac{1 - \cos(\phi_{flow})}{\sqrt{\xi_1(Re)}} \] (3.26)

The constants in the equation depend on corrugation geometry:

\[ C_5, C_6, C_7 = f(\hat{a}, \Lambda) \] (3.27)

For \( \Lambda/\hat{a} \approx 4 - 8 \), the constants are: \( C_5 = 3.8, C_6 = 0.18 \) and \( C_7 = 0.36 \).
3.3.3. ELEVATION LOSSES
Due to the weight of the vapor and liquid in the plate heat exchanger, pressure increases in
downward direction within the plate heat exchanger. This means that for upward flow the
elevation difference will cause a decrease in pressure, but for downward flow elevation dif-
fferences causes an increase in pressure. For two-phase flow, the mean phase density ($\rho_m$)
is required along with the gravitational acceleration ($g$) and height difference of a single
control volume ($\Delta z$), to calculate the elevation pressure drop ($\Delta P_{ele, CV}$):

$$\Delta P_{ele, CV} = \rho_m g \Delta z$$  \hspace{1cm} (3.28)

The mean phase density is obtained through Equation 3.8. For single phase flow, the
single phase density ($\rho$) is instead used in the same equation.

3.3.4. DECELERATION LOSSES
In two-phase flow where phase change occurs, pressure losses or gains arise due to respec-
tively decrease or increase in volume. For condensing flow, the relative mass of the liquid
phase increases compared to the vapor phase. Due to the higher density and thus lower
specific volume of the liquid, the effective volume decreases causing a loss in pressure. The
deceleration pressure drop for a control volume ($\Delta P_{dec, CV}$) is given as:

$$\Delta P_{dec, CV} = \frac{1}{2} \Delta \nu \Delta X_m$$  \hspace{1cm} (3.29)

The changes in specific volume between vapor and liquid phase ($\Delta \nu$) and in mean va-
por quality between the control volume inlet and outlet ($\Delta X_m$) are defined as follows:

$$\Delta \nu = \nu_V - \nu_L$$  \hspace{1cm} (3.30)

$$\Delta X_m = X_{b, out} - X_{b, in}$$  \hspace{1cm} (3.31)

3.3.5. TOTAL PRESSURE DROP
The total pressure drop in each control ($\Delta P_{CV}$) volume is obtained by adding all three con-
tributions:

$$\Delta P_{CV} = \Delta P_{f, CV} + \Delta P_{ele, CV} + \Delta P_{dec, CV}$$  \hspace{1cm} (3.32)

The cumulative pressure drop ($\Delta P_{cum}$) in the core of the heat exchanger is then ob-
tained by adding the pressure drops of each control volume $i$, starting from the first control
volume up to control volume $N$:

$$\Delta P_{cum} = \sum_{i=1}^{N} \Delta P_i$$  \hspace{1cm} (3.33)

The cumulative pressure drop of each contributing pressure drop term may be calcu-
lated in a similar way. Taking the total cumulative pressure drop for all control volumes
($\Delta P_{tot}$), so for $N = \gamma$, and including the pressure drop over the inlet and outlet sections
($\Delta P_{port, in}$ and $\Delta P_{port, out}$) gives the expected pressure difference over the heat exchanger
($\Delta P_{HX}$):
\[ \Delta P_{HX} = \Delta P_{tot} + \Delta P_{port, in} + \Delta P_{port, out} = \sum_{i=1}^{Y} \Delta P_i + \Delta P_{port, in} + \Delta P_{port, out} \quad (3.34) \]

### 3.4. HEAT TRANSFER

Heat transfer correlations are provided for the two-phase and single phase flows in this section. For the two-phase flow, the heat transfer correlations are based on the assumption of a laminar condensing film flow. In this section, the Reynolds and Prandtl numbers are based on the general definitions of these dimensionless numbers presented in section A.1. In section A.4, on alternative heat transfer correlations, the same definitions are used. The liquid and vapor phase superficial velocities \( u_{sup} \) and hydraulic diameter \( d_h \) are used as respectively the characteristic velocity and characteristic length in determining the phase Reynolds numbers:

\[
\begin{align*}
Re_L &= \frac{\rho_L u_{sup,L} d_h}{\mu_L} \quad (3.35) \\
Re_V &= \frac{\rho_V u_{sup,V} d_h}{\mu_V} \quad (3.36)
\end{align*}
\]

Since different definitions of the hydraulic diameter are used in literature, and often the used definition is not reported, Equation A.90 included in Appendix A is used throughout the entire model and report for the hydraulic diameter, unless stated otherwise.

### 3.4.1. TWO-PHASE FILM FLOW

In the two-phase film flow section, heat transfer relations are derived separately for the liquid phase and the vapor phase.

**LIQUID PHASE FILM**

Different approaches may be used to calculate the heat transfer coefficient of the working fluid liquid phase film. It is expected that the Nusselt film theory for condensation gives the most accurate results, which is an analytically derived relation to provide local values for flow over a vertical surface:

\[
\alpha_L = \frac{\lambda_L}{d_{film}} \quad (3.37)
\]

Here, it is assumed that the heat transfer is purely conductive, and occurs over the thickness of the liquid film \( d_{film} \). Film thickness relations are given further ahead in this section. This relation is also used in the work of Würfel and Ostrowski (2004) [70] and Thonon and Bontemps (2002) [69], as discussed in Chapter 2. A similar relation for the liquid film heat transfer coefficient for gravity-controlled laminar film condensation on the smooth vertical surface is given by Tovazhnyanski and Kapustenko (1984) [50], where the relation for the film thickness is incorporated in the relation for the heat transfer coefficient:

\[
\alpha_L = \frac{\rho_L^2 g \lambda_L^3 (\Pi/2)^{1/3}}{3 \mu_L m_{L,CV}} \quad (3.38)
\]
3. MATHEMATICAL MODEL

Here, \( \Pi \) is the wetted perimeter of the flow, and \( \dot{m}_{L,CV} \) is the liquid phase mass flow rate within a single control volume. Direct relations for the film thickness are discussed further ahead. The general relation between the Nusselt number \( (Nu_L) \) and heat transfer coefficient \( (\alpha_L) \) for the liquid phase is presented here:

\[
Nu_L = \frac{\alpha_L d_{film}}{\lambda_L} \tag{3.39}
\]

In case the Nusselt film theory for condensation is used, the value of the Nusselt number is equal to unity: \( Nu_L = 1 \). The heat transfer resistance within the liquid film is present at two different regions, near the wall and near the vapor-liquid interface of the working fluid. The heat transfer coefficients in these two regions are considered equal, since the heat transfer mechanism at either side is identical:

\[
\alpha_{L,i} = \alpha_{L,w} = \alpha_L \tag{3.40}
\]

VAPOR PHASE

The Nusselt relation for natural convection of laminar vapor flow over a plate (Mills (1998)) is used to predict the heat transfer within the vapor phase flow, which is assumed not to be in contact with the plates but with the liquid phase film. The Nusselt value is given as function of the Prandtl number function \( (\psi_V) \) and vapor phase Rayleigh number within a certain specified range \( (Ra_V \lesssim 10^9) \):

\[
Nu_V = 0.68 + 0.670(Ra_V \psi_V)^{1/3} \tag{3.41}
\]

The Prandtl number function is given by the equation:

\[
\psi_V = \left[ 1 + \left( \frac{0.942}{Pr_V} \right)^{9/16} \right]^{-16/9} \tag{3.42}
\]

The Prandtl number of the vapor phase \( (Pr_V) \) is calculated using the general relation for the Prandtl number, given in Equation A.7. The Rayleigh number is a function of position \( (z) \) inside the channel, with \( z = 0 \) at the bottom of the channel. The Rayleigh number is also a function of the volumetric expansion coefficient \( (\beta) \), temperature difference between interface and bulk, gravitational acceleration and certain fluid properties:

\[
Ra_V = \frac{\beta(T_{V,b} - T_i)z^3 \rho_V^2 c_p,V}{\lambda_V \mu_V} \tag{3.43}
\]

The general relation between the Nusselt number and heat transfer coefficient for the vapor phase is given by:

\[
Nu_V = \frac{\alpha_V d_V}{\lambda_V} \tag{3.44}
\]

Here, \( d_V \) is the characteristic length of the vapor phase flow. The heat transfer resistance within the vapor phase is present near the vapor-liquid interface of the working fluid, or near the wall in case the flow consists of a pure vapor phase:

\[
\alpha_{V,i} = \alpha_{V,w} = \alpha_V \tag{3.45}
\]
**Film Thickness**

The film thickness is calculated using the Nusselt film theory for condensation:

\[
d_{film} = \left(\frac{\lambda_L \mu_L (T_{sat,L} - T_{wf,w})}{\Delta h_{vap,i} \rho_L (\rho_L - \rho_V)}\right)^{1/4}
\]  

(3.46)

Here, the saturation temperature of the liquid phase \((T_{sat,L})\) and the temperature of the wall at the working fluid side \((T_{wf,w})\) are required and assumed to be known. \(\Delta h_{vap,i}\) is the specific enthalpy of vaporization corresponding to the liquid and vapor phase enthalpies at the interface:

\[
\Delta h_{vap,i} = h_{V,i} - h_{L,i}
\]  

(3.47)

Alternatively, the Nusselt film theory for condensation equation is given based on only fluid properties and the liquid phase mass flow rate within a control volume, based on the work by Holman (2010) [96]:

\[
d_{film} = \left(\frac{3 \dot{m}_{L, CV} \mu_L}{g \rho_L (\rho_L - \rho_V)}\right)^{1/3}
\]  

(3.48)

Both of the equations discussed should give identical or similar values of the film thickness. The second relation is implemented into the model, because of its simplicity. For any definition of the film thickness, the characteristic length scale of the vapor phase flow is described by the following equation:

\[
d_V = d_g - 2d_{film}
\]  

(3.49)

The factor 2 is present in this equation, since there are two separated liquid films on both plates within a single control volume, but the vapor phase flow is continuous within the central part of the channel.

**Surface Tension**

In case the effects of surface tension are significant, it is relevant to take these effects into account when calculating the film thickness of the liquid phase film. In this study, surface tension effects are assumed to be negligible. However, values of dimensionless numbers related to surface tension are calculated in the model, to check if the assumption is valid. The three most important numbers related to surface tension are the Weber number \((We)\), Eötvös number \((Eo)\) and Capillary number \((Ca)\). The phase Weber number relates the inertial effects to the viscous effects of the liquid and vapor phase flow, separately for each phase:

\[
We_L = \frac{\rho_L u_{sup,L}^2 d_h}{\sigma_L}
\]  

(3.50)

\[
We_V = \frac{\rho_V u_{sup,V}^2 d_h}{\sigma_V}
\]  

(3.51)

The Eötvös number relates the effects of gravity to the effects of surface tension, and the Capillary number relates the viscous forces in the flow to the surface tension. General
MATHEMATICAL MODEL

definitions are found in Equation A.1 and Equation A.2, specific two-phase relations are given here:

\[ Eo = \frac{d_h^2 g (\rho_L - \rho_V)}{\sigma_L} \] (3.52)

\[ Ca = \frac{u_{sup,V} \mu_L}{\sigma_L} \] (3.53)

Generally speaking, the smaller each of these dimensionless numbers is, the larger the relative effect of surface tension in the system. Typical values for which surface tension is dominant over the other force are: \( We < 1, Eo < 1 \) and \( Ca < 10^{-5} \).

3.4.2. SINGLE PHASE FLOW

The Nusselt number is based on the generalized Lévêque equation for developing flow in corrugated channels, reported in Stephan et al. (2010) [8]:

\[ Nu = 1.615 \left( \frac{\xi Re^2 Pr d_h}{64 L_{corr}} \right)^{1/3} \] (3.54)

The characteristic length \( (L_{corr}) \) is the length scale specific for the flow along the channel corrugations defined in Equation A.86. The friction factor \( (\xi) \) is calculated through the relations described in section 3.3. The heat transfer coefficient \( (\alpha) \) is calculated by rearrangement of the general Nusselt number equation.

3.5. MASS TRANSFER

Mass transfer occurs in the working fluid when a mixture is used. The mass transfer is a result of the combination of diffusion, convection and transport across the vapor-liquid interface. The Chilton-Colburn analogy between heat and mass transfer is used to obtain mass transfer coefficients of the phases, as is also done in the work of Nefs (2013) [77]. The diffusive molar fluxes \( (F) \) of the phases are given through the following relations:

\[ F_L = c_L \cdot k_L \] (3.55)

\[ F_V = c_V \cdot k_V \] (3.56)

Where \( c_L \) and \( c_V \) are the molar volumetric concentrations in the liquid and vapor phases respectively, based on the phase density, composition and species molar mass:

\[ c_L = \frac{\rho_L}{\bar{x}_b M_1 + (1 - \bar{x}_b) M_2} \] (3.57)

\[ c_V = \frac{\rho_V}{\bar{y}_b M_1 + (1 - \bar{y}_b) M_2} \] (3.58)

\( k_L \) and \( k_V \) are the mass transfer coefficients, relating heat and mass transfer through the Chilton-Colburn analogy:

\[ k_L = \frac{\alpha_{L,i}}{\rho_L c_p L} Pr_L^{2/3} Sc_L^{-2/3} \] (3.59)
3.6. Solution Procedure

The Schmidt numbers \((Sc)\) of the liquid and vapor phases are given by the general equation for the Schmidt number in Appendix A, using the fluid properties specific for each phase.

\[
k_V = \frac{\alpha_{V,i} \rho_V c_{p,V}}{Pr_V^{2/3} Sc_V^{-2/3}} \tag{3.60}
\]

The Schmidt numbers \((Sc)\) of the liquid and vapor phases are given by the general equation for the Schmidt number in Appendix A, using the fluid properties specific for each phase.

### 3.6. Solution Procedure

In the next section the solution method is described for implementation of the model. Through several additional relations, the different equations and parameters describing the system geometry, fluid properties, pressure drop, heat and mass transfer are coupled to obtain a solvable system of equations. The steps required to solve the system are described in logical order, in the same way as they are implemented in the Matlab code. A schematic representation of the solution algorithm is given in Figure 3.7. In this figure, the starting and end points of the model are indicated by the black boxes with a thick edge. Different colors correspond to specific types of calculations, for instance blue refers to the specific calculations for the cooling water side and red to the specific calculations for the working fluid side. Lines and arrows represent one-way or two-way relations of input and output between steps, and numbers next to lines represent the order in which steps are walked through in the model. A simplified version of the figure is presented in Figure 3.6, where the different substeps for the initial values, working fluid and cooling water loops are left out.

![Figure 3.6: The basic steps in the solution algorithm of the model. This includes the initialization, model calculations starting with the initial value computation before the working fluid and cooling water iteration loops, and the computation of the wall heat flux deviation.](image)

#### 3.6.1. Initial Settings and Main Code

Before execution of the model, certain parameters need to be specified. An overview of these parameters is listed in Table 3.1. The different approaches for fluid property calculation are discussed in Appendix A.

The main calculation steps in the model involve the computation of the initial values for both streams, and the iterative loop where the parameters are calculated for the working fluid.
fluid and cooling water streams respectively. The loop runs until the computed heat fluxes across the wall are close enough based on the calculations on either side. All values are saved and plots are made of a large selection of parameters as function of position inside the heat exchanger, based on the implemented division into control volumes. A schematic representation of the main code and the initial settings is given in Figure 3.6.

Figure 3.7: The basic steps in the solution algorithm of the model, including the substeps for the initial value computation, working fluid and cooling water iteration loops.
### Table 3.1: The initial settings of the model, including the type of setting, default and alternative options, and symbol in case of a parameter setting.

<table>
<thead>
<tr>
<th>Setting</th>
<th>Symbol</th>
<th>Default option</th>
<th>Alternative option</th>
</tr>
</thead>
<tbody>
<tr>
<td>Experimental dataset</td>
<td></td>
<td>1 – N</td>
<td></td>
</tr>
<tr>
<td>Working fluid species</td>
<td></td>
<td>Ammonia-water</td>
<td>Other binary mixtures (^1)</td>
</tr>
<tr>
<td>Plate material type</td>
<td></td>
<td>Stainless steel</td>
<td>Other materials</td>
</tr>
<tr>
<td>Constant fluid properties</td>
<td></td>
<td>NIST Refprop database</td>
<td></td>
</tr>
<tr>
<td>Vapor phase volumetric expansion coefficient</td>
<td>(\beta_V)</td>
<td>Ideal gas approximation (A.45)</td>
<td>FluidProp [33] (A.46)</td>
</tr>
<tr>
<td>Vapor quality</td>
<td>(X)</td>
<td>Main FluidProp [33] method (A.26)</td>
<td>Alternative methods (A.27, A.28)</td>
</tr>
<tr>
<td>Cooling water salinity</td>
<td>(S)</td>
<td>0 g kg(^{-1})</td>
<td>35 g kg(^{-1}), or other values</td>
</tr>
<tr>
<td>Gravitational acceleration</td>
<td>(g)</td>
<td>9.81 ms(^{-2})</td>
<td>–</td>
</tr>
<tr>
<td>Amount of control volumes</td>
<td>(\gamma)</td>
<td>20000</td>
<td>Other values &gt; 1000 (^4)</td>
</tr>
<tr>
<td>Ratio first control volume size to average size</td>
<td>(\delta)</td>
<td>1</td>
<td>Other values ≥ 0</td>
</tr>
<tr>
<td>Maximum number of iterations</td>
<td>(N_{it, max})</td>
<td>10 – 25 (^5)</td>
<td>Other values &gt; 0</td>
</tr>
<tr>
<td>Maximum allowable relative wall heat flux deviation (^6)</td>
<td>(\Delta q_{dev, w, max}^*)</td>
<td>&lt; 0.1%</td>
<td>Other values &gt; 0</td>
</tr>
<tr>
<td>Maximum vapor quality for two-phase flow</td>
<td>(X_{TP, max})</td>
<td>0</td>
<td>Other values ≥ 0 and ≤ 1</td>
</tr>
</tbody>
</table>

\(^1\)The model is only made compatible with the ammonia-water mixture.  
\(^2\)The correlations by Green and Perry (2007) [37] are not yet implemented in the model.  
\(^3\)The water used in the experiments in demineralized water with negligible salt contents.  
\(^4\)This value should be in the order of at least several thousands for reasonable stability and reliability of the model.  
\(^5\)There is a minimum amount of iterations needed to achieve significant convergence, and too many iterations will lead to large simulations times, which is unpractical.  
\(^6\)The calculation of the relative wall heat flux deviation is found further ahead, in Equation 3.150.
3.6.2. Heat Exchanger Geometry and Experimental Data

From the experimental dataset, certain measured parameters need to be converted to obtain their value in the desired units. This is the case for pressure, temperature and flow measurements:

\[ P_{Pa} = P_{bar} \cdot 10^5 \]  
(3.61)

\[ T_K = T_C + 273.15 \]  
(3.62)

\[ \dot{m}_{kg/s} = \frac{V_{Lh} \rho}{3.6 \cdot 10^6} \]  
(3.63)

\[ \dot{m}_{kg/s} = \frac{\dot{m}_{kg/min}}{60} \]  
(3.64)

All relevant constant geometrical parameters are either known or calculated using the approach discussed in Appendix A.

3.6.3. Control Volume Parameters

The heat exchanger is divided into a selected amount of control volumes with individual sizes that depend on the initial settings of the model. The control volume size depends on the total plate length \((L_p)\), the amount of control volumes \((\gamma)\) and the ratio of the first control volume size to the average size \((\delta)\). First, a control volume size distribution function \((\epsilon_i)\) is defined as function of the control volume number \((i)\):

\[ \epsilon_i = \frac{2i(1-\delta) + \delta(\gamma + 1) - 2}{\gamma - 1} \]  
(3.65)

Now, the size of control volume \(i\) \((\Delta z_i)\) is calculated:

\[ \Delta z_i = \frac{L_p}{\gamma \cdot \epsilon_i} \]  
(3.66)

The position inside the heat exchanger \((L_{HX})\) of control volume \(N\) is given as:

\[ L_{HX} = 0.5\Delta z_N + \sum_{i=1}^{N-1} \Delta z_i \]  
(3.67)

Here, \(z_N\) is the size of control volume \(N\). The calculations for fluid properties, phase composition and certain other parameters in the first control volume are done based on the inlet values of the channel control volume: temperature \((T_{in})\), pressure \((P_{in})\) and also composition \((w_{in})\) in case of the working fluid. In the control volumes after the first one, the values in the control volume bulk are assumed to be equal to the inlet values of all parameters in the control volume. This is done to avoid complexity when using average parameter values, which requires outlet values to be known and can only be implemented by introducing an additional iterative loop. This approach has been implemented in the model, but was found to lead to larger computation times and a larger degree of instability, after which it was rejected. All relations to derive thermophysical properties are discussed in Appendix A. In case of the phase and interface temperatures, pressure, mass flow rate, composition and phase enthalpies, the inlet values of each control volume after the first
one are taken as the outlet values of the previous control volume. The calculations required
to get these outlet values are discussed throughout this section.

The initial profiles of the temperature and pressure on the cooling water side are es-

timated based on the measured inlet and outlet values. The estimated temperature and

pressure profiles at the cooling water side are used to estimate the heat transfer coefficient

on this side. The heat transfer coefficient and temperature profile estimates are then used

in calculation of the heat transfer across the wall in the first iteration loop of the working

fluid side. The profiles are by default assumed to be linear profiles between the measured

inlet and outlet values as a function of the control volume number. Other approaches are

possible, such as assuming a constant value equal to the measured inlet value or the nu-

merical average of the inlet and outlet value. The assumed profiles should have no effect

on the final convergence of the model, but will affect how close the first iteration is to the

correct result, and thus on how quickly the final solution is reached.

When the mean vapor quality \(X\) within a control volume is known, all relevant mass

flows are calculated based on the known total mass flows \(\dot{m}_{\text{tot}}\) of both the working fluid

and cooling water streams, combined with the number of channels for each stream. The

mass flow rates within the cooling water channel, and within the control volume of the

cooling water channel (half the channel) are given as follows:

\[
\dot{m}_{cw} = \frac{\dot{m}_{cw,\text{tot}}}{N_{c,cw}} \tag{3.68}
\]

\[
\dot{m}_{cw,\text{CV}} = \frac{\dot{m}_{cw,\text{tot}}}{2 \cdot N_{c,cw}} \tag{3.69}
\]

The liquid and vapor mass flow rates in a single working fluid channel are given by the

following relations:

\[
\dot{m}_L = 2 \cdot \dot{m}_{wf,\text{CV}} \cdot (1 - X) \tag{3.70}
\]

\[
\dot{m}_V = 2 \cdot \dot{m}_{wf,\text{CV}} \cdot X \tag{3.71}
\]

The liquid and vapor mass flow rates in a control volume are half of these values:

\[
\dot{m}_{L,\text{CV}} = \dot{m}_{wf,\text{CV}} \cdot (1 - X) \tag{3.72}
\]

\[
\dot{m}_{V,\text{CV}} = \dot{m}_{wf,\text{CV}} \cdot X \tag{3.73}
\]

The species mass flow rates in a control volume of ammonia (species 1) and water

(species 2) are given by the following equations:

\[
\dot{m}_{1,\text{CV}} = x_b \dot{m}_{L,\text{CV}} + y_b \dot{m}_{V,\text{CV}} \tag{3.74}
\]

\[
\dot{m}_{2,\text{CV}} = (1 - x_b) \dot{m}_{L,\text{CV}} + (1 - y_b) \dot{m}_{V,\text{CV}} \tag{3.75}
\]

The mass flow rate within the control volume of a working fluid channel is calculated in

a similar way as for the cooling water:
\[ \dot{m}_{wf, CV} = \frac{\dot{m}_{wf, tot}}{2 \cdot N_{c, wf}} \] (3.76)

Mass fluxes \(j\) are calculated through any mass flow \(\dot{m}\) divided by the channel cross-sectional area.

\[ j = \frac{\dot{m}}{A_c} \] (3.77)

The liquid and vapor phase volumetric flow rates in the channel are given by the following relations:

\[ \dot{V}_L = \frac{\dot{m}_L}{\rho_L} \] (3.78)

\[ \dot{V}_V = \frac{\dot{m}_V}{\rho_V} \] (3.79)

The liquid and vapor phase superficial velocity in the channel are given as:

\[ u_{sup, L} = \frac{\dot{V}_L}{A_c} \] (3.80)

\[ u_{sup, V} = \frac{\dot{V}_V}{A_c} \] (3.81)

In these equations, \(A_c\) is the cross-sectional area of the channel. The actual velocity of both phases depends on their characteristic dimensions \((d_L\) and \(d_V)):

\[ u_{act, L} = \frac{\dot{V}_L}{B_p d_L} \] (3.82)

\[ u_{act, V} = \frac{\dot{V}_V}{B_p d_V} \] (3.83)

In the case of film flow, the characteristic dimension of the liquid phase is the film thickness \((d_{film})\):

\[ d_L = d_{film} \] (3.84)

The characteristic length for the vapor phase flow is given by Equation 3.49.

### 3.6.4. Working Fluid Main Calculations

In Figure 3.8, the steps in the working fluid loop are presented. In this loop, first the grid is specified and required parameters are extracted from the existing Matlab structure. Phase partial enthalpy differences are computed using relations A.34 to A.39, and heat capacities using relations A.55 and A.56. Next, the liquid phase saturation temperature is computed using relation A.44. The areas for heat and mass transfer at the phase interface and area for heat transfer at the wall are assumed equal to each other, and equal to the enlarged plate area of the control volume through:

\[ A_i = A_w = \Gamma \cdot \Delta z \cdot B_p \] (3.85)
Relevant heat transfer properties are calculated using the relations described in section 3.4, and mass transfer properties through the relations in section 3.5. These last calculations require the ratio of ammonia molar flux to the total molar flux ($z$) to be known, which is determined through the iterative loop discussed next.

**RATIO AMMONIA MOLAR FLUX TO TOTAL MOLAR FLUX**

A schematic representation of the solution procedure used to calculate the ratio of ammonia molar flux to the total molar flux ($z$) is given in Figure 3.9. In the following paragraph, more is explained about the steps and equations needed to obtain the final value of $z$.

Four different cases are distinguished, of which two with two different subcases. All cases are also discussed in Sieres and Fernández-Seara (2007) [78]. The four cases differ in the relative values of the molar fractions of both phases in the bulk and at the interface: $\bar{x}_b$, $\bar{x}_i$, $\bar{y}_b$, and $\bar{y}_i$. An overview of the different cases is presented in Figure 3.10. The vertical lines in this figure represent the vapor-liquid interface. Generally, for ammonia absorption the vapor phase bulk molar fraction of ammonia is larger than the liquid phase bulk molar fraction:

$$\bar{y}_b > \bar{x}_b$$  \hspace{1cm} (3.86)

Case 1 is typical for desorption, and case 3 for evaporation. These cases are not discussed here, since they are not expected to occur inside a condenser or absorber. Case 2, however, is typical for condensation or absorption. It occurs when the interface temperature ($T_i$) is smaller than the liquid phase bulk saturation temperature ($T_{sat,L,b}$) and smaller than the vapor phase bulk saturation temperature ($T_{sat,V,b}$). This results in the interface molar fraction of the vapor phase ($\bar{y}_i$) to be larger than the bulk molar fraction ($\bar{y}_b$), and the interface molar fraction of the liquid phase ($\bar{x}_i$) to be larger than the bulk molar fraction ($\bar{x}_b$). Also, the interface molar fraction of the liquid phase is expected to be smaller than the bulk molar fraction of the vapor phase. The following relations thus apply to this case:

$$T_i < T_{sat,L,b}, T_{sat,V,b}$$  \hspace{1cm} (3.87)
3. MATHEMATICAL MODEL

Figure 3.9: Steps taken in the solution procedure used to calculate the ratio of ammonia molar flux to total molar flux ($z$). Different cases can be identified, each with a different initial estimate for $z$. For each case, the value of $z$ is found iteratively by satisfying the equation at the bottom. The approach is similar to the approach used in Nefs (2013) [77].

Figure 3.10: Possible cases of molar fraction profiles in the liquid and vapor phase during heat and mass transfer processes in ammonia-water absorption systems, where the vertical lines represent the liquid-vapor interface. The figure is an altered version from Sieres and Fernández-Seara (2007) [78].
\[ \tilde{y}_i > \tilde{y}_b, \tilde{x}_i > \tilde{x}_b, \tilde{x}_i < \tilde{y}_b \quad (3.88) \]

The previous relations are related through the temperature-composition diagram of the saturated ammonia-water mixture, at a certain pressure. The relevant parameters are indicated in Figure 3.11, which was constructed using FluidProp with 20 data points at an absolute pressure of 6.7 bar.

![Temperature-composition diagram for the ammonia-water mixture at an absolute pressure of 6.7 bar](image)

Figure 3.11: Temperature-composition diagram for the ammonia-water mixture at an absolute pressure of 6.7 bar, created using FluidProp. The saturated liquid and vapor lines are represented, and the phase interface and bulk compositions are indicated for the case where \( T_i < T_{sat,L,b}, T_{sat,V,b} \) (case 2).

Case 4 is typical for absorption:

\[ T_{sat,V,b} < T_i < T_{sat,L,b} \quad (3.89) \]

\[ \tilde{y}_i < \tilde{y}_b, \tilde{x}_i > \tilde{x}_b \quad (3.90) \]

Two different subcases exist. The difference between the cases is a consequence of the value \( z \) being either higher or lower than the bulk molar fractions. When \( z \) is higher than the bulk molar fractions, case 4a occurs, which is typical of pure absorption:

\[ F_V \cdot (\tilde{y}_i - \tilde{y}_b) > F_L \cdot (\tilde{x}_b - \tilde{x}_i) \quad (3.91) \]

\[ \tilde{y}_b < z < \infty \quad (3.92) \]

Case 4b is typical of either absorption or desorption:

\[ F_V \cdot (\tilde{y}_i - \tilde{y}_b) < F_L \cdot (\tilde{x}_b - \tilde{x}_i) \quad (3.93) \]

\[ -\infty < z < \tilde{x}_b \quad (3.94) \]
3. Mathematical model

Mass transfer of species across the interface
After finding the value of \( z \), the species molar fluxes through the interface are calculated. If \( z = 0 \), all molar fluxes are zero, since there is no mass transfer across the interface:

\[
\tilde{j}_{V,1} = \tilde{j}_{V,2} = \tilde{j}_{L,1} = \tilde{j}_{L,2} = 0
\]  

(3.95)

For other values of \( z \), the molar fluxes for both species in both phases are calculated:

\[
\tilde{j}_{V,1} = z \cdot F_{V} \cdot \ln \left( \frac{z - \tilde{y}_{b}}{z - \tilde{y}_{i}} \right)
\]  

(3.96)

\[
\tilde{j}_{V,2} = \frac{\tilde{j}_{V,1}}{z - \tilde{j}_{V,1}}
\]  

(3.97)

\[
\tilde{j}_{L,1} = z \cdot F_{L} \cdot \ln \left( \frac{z - \tilde{x}_{b}}{z - \tilde{x}_{i}} \right)
\]  

(3.98)

\[
\tilde{j}_{L,2} = \frac{\tilde{j}_{L,1}}{z - \tilde{j}_{L,1}}
\]  

(3.99)

To satisfy the mass balances of species across the interface, the species fluxes in the vapor and liquid phases should equal:

\[
\tilde{j}_{V,1} = \tilde{j}_{L,1}
\]  

(3.100)

\[
\tilde{j}_{V,2} = \tilde{j}_{L,2}
\]  

(3.101)

Mass flows of the species through the interface are then calculated through:

\[
\dot{m}_{i,1} = A_{i} \cdot \tilde{j}_{V,1} \cdot M_{1}
\]  

(3.102)

\[
\dot{m}_{i,2} = A_{i} \cdot \tilde{j}_{V,2} \cdot M_{2}
\]  

(3.103)

A schematic representation of the mass transfer process across the interface is given in Figure 3.12. The interface between the phases is indicated by a dashed vertical line, as well as the location where the bulk of the phases begins. The species mass fraction is indicated at the interface \((x_{i}, y_{i})\) and the bulk \((x_{b}, y_{b})\) of the phases. Also the interface area \((A_{i})\), mass transfer coefficients of both phases \((k_{L}, k_{V})\) and molar fluxes \((\tilde{j})\) of both species in either phase are indicated. The horizontal black arrows represent the direction in which the mass flux is defined. The values should be negative for condensation or absorption, and rough profiles of the mass fractions are indicated by the thin dashed line.

Heat transfer through the interface
If the mass transfer across the interface is known, together with the associated heat effects with this transfer and the heat transfer in both phases on either side of the interface, the interface temperature \( (T_{i}) \) is determined in a few calculation steps. The calculations are based on an energy balance across the interface. The total heat flows across the interface for both the liquid and vapor side are set up in the same way as described in Sieres and Fernández-Seara (2007) [78]. For each phase, the total heat flow across the interface \((Q_{i})\)
is the sum of the heat flow due to convective and conductive transport in direction perpendicular to the flow within the phase \((Q_L\) or \(Q_V\)) and the heat that is lost or added due to the absorption of species at the interface. The heat flows on either side of the interface thus differ in the amount of heat change that is associated with the absorption process, and the total heat flow from the interface to the liquid phase and from the vapor phase to the interface are given as:

\[
Q_{L,i} = Q_L + Q_{L,i,1} + Q_{L,i,2} \quad (3.104)
\]
\[
Q_{V,i} = Q_V + Q_{V,i,1} + Q_{V,i,2} \quad (3.105)
\]

In the equations above, the terms \(Q_{L,i,1}\), \(Q_{L,i,2}\), \(Q_{V,i,1}\) and \(Q_{V,i,2}\) are the heat flow terms associated with the absorption of species at the interface:

\[
Q_{L,i,1} = m_{i,1} \tilde{h}_{L,i,1} \quad (3.106)
\]
\[
Q_{L,i,2} = m_{i,2} \tilde{h}_{L,i,2} \quad (3.107)
\]
\[
Q_{V,i,1} = m_{i,1} \tilde{h}_{V,i,1} \quad (3.108)
\]
\[
Q_{V,i,2} = m_{i,2} \tilde{h}_{V,i,2} \quad (3.109)
\]

In the equations above, the values of \(\tilde{h}\) give the specific partial enthalpy of a certain species within a certain phase. Specific partial enthalpy relations are found in subsection A.2.2. The convective heat flows in both phases are calculated using the obtained interface temperature, phase bulk temperatures, phase heat transfer coefficients and the interface area. The heat flow from the interface to the liquid phase and from the vapor phase to the interface are given respectively by:

\[
Q_L = A_i \cdot \alpha_{L,i} \cdot (T_{L,b} - T_i) \quad (3.110)
\]
\[
Q_V = A_i \cdot \alpha_{V,i} \cdot (T_i - T_{V,b}) \quad (3.111)
\]

Since energy continuity applies at the vapor-liquid interface, equations 3.104 and 3.105 are set equal to one another, which allows isolation and calculation of the interface temperature:
\[ T_i = \frac{T_L + T_V \cdot \alpha^*_i + \Delta T_{L,\text{cond}}}{1 + \alpha^*_i} \quad (3.112) \]

In the equation above, two new parameters are introduced for simplification of the relation. The temperature change associated with condensation or absorption from the vapor phase into the liquid phase \((\Delta T_{L,\text{cond}})\) is defined by the sum of the heat removal of both species, divided by the interface area of the control volume and the liquid phase heat transfer coefficient:

\[ \Delta T_{L,\text{cond}} = \frac{Q_{L,i,1} + Q_{L,i,2} - (Q_{V,i,1} + Q_{V,i,2})}{\alpha_{L,i} A_i} \quad (3.113) \]

The other parameter, the ratio of the heat transfer of the vapor phase to the liquid phase \((\alpha^*_i)\) is defined as:

\[ \alpha^*_i = \frac{\alpha_{V,i}}{\alpha_{L,i}} \quad (3.114) \]

As a check if the energy balance across the interface is satisfied, the difference in the balance on either side is computed by setting up the energy balance. The absolute heat flux difference \((q_{i,\text{diff}})\) in the energy balance across the interface is obtained by calculating:

\[ q_{i,\text{diff}} = \alpha_{L,i} (T_{L,b} - T_i) - \alpha_{V,i} (T_i - T_{V,b}) - \tilde{j}_{V,1} \cdot M_1 \cdot \Delta \tilde{h}_1 - \tilde{j}_{V,2} \cdot M_2 \cdot \Delta \tilde{h}_2 \quad (3.115) \]

The difference should be zero if all of the previous equations related to the heat transfer across the interface are implemented correctly. The difference relative to the liquid phase heat flow to the interface is given by:

\[ q_{i,\text{diff}}^* = \frac{q_{i,\text{diff}} \cdot A_i}{Q_{L,i}} \quad (3.116) \]

Similarly to the mass transfer process across the interface, a schematic representation of the heat transfer across the interface is found in Figure 3.13. In this figure, the directions of the liquid and vapor phase heat flows \((Q_L \text{ and } Q_V)\) are indicated by the horizontal arrows, and the thin dashed lines indicate the rough temperature profile from the liquid bulk \((T_{L,b})\) to the vapor bulk \((T_{V,b})\) and at the interface \((T_i)\). The interface area, heat flow change associated with the absorption process at the interface \((\Delta Q_{\text{abs}})\) and the heat transfer coefficients near the interface of either phase \((\alpha_{L,i} \text{ and } \alpha_{V,i})\) are also indicated.

**Heat Transfer Across the Wall**

To obtain the wall temperature of the wall at the working fluid side \((T_{wf,w})\), the heat transfer characteristics within the wall and on the working fluid side of the wall need to be known. The thermal conductivity of the plate material, in case of stainless steel, as function of the temperature in °C is given by (Bailey (2005) [97]):

\[ \lambda_w = 14.6 + 1.27 \cdot 10^{-2} \cdot \frac{1}{2} (T_{wf,w,*C} + T_{cw,w,*C}) \quad (3.117) \]

An estimate of the working fluid side wall temperature is used, since the actual value is not yet known. In case the working fluid is a pure vapor, the bulk temperature and heat
Figure 3.13: Representation of the heat transfer process across the interface within the working fluid channels of the heat exchanger. For absorption or condensation, the heat flows \((Q)\) will be in the opposite direction of the indicated arrows, and the values are thus negative.

transfer coefficient near the wall of the working fluid in the next relations should be those of the vapor and not the liquid phase. In general, this should not be the case, but it is possible near the inlet under certain conditions. The subscript ‘\(w f\)’ is used to denote the general phase of the working fluid that is in contact with the wall in the following equations, so \(T_{w f, b}\) refers to the bulk phase temperature of the working fluid phase that is in contact with the wall. Heat flow from the working fluid liquid bulk to the wall should be equal to the heat flow through the wall and the heat flow from the wall to the cooling water liquid bulk. This energy continuity condition holds, since conduction of heat in the direction of flow is neglected both inside the wall and in the regions of both fluids near the wall. Convection in axial direction in the regions near the wall is also neglected, it only occurs in the bulk of the phases. The three heat flows discussed before are given respectively by the equations:

\[
Q_{w f, w} = \alpha_{w f} \cdot A_w \cdot (T_{w f, w} - T_{w f, b})
\] (3.118)

\[
Q_{w, con} = \frac{\lambda_w}{d_p} \cdot A_w \cdot (T_{cw, w} - T_{w f, w})
\] (3.119)

\[
Q_{cw, w} = \alpha_{cw} \cdot A_w \cdot (T_{cw, b} - T_{cw, w})
\] (3.120)

Here, \(\alpha_{w f}\) and \(\alpha_{cw}\) are the effective heat transfer coefficients near the wall of the working fluid and cooling water phase in contact with the wall respectively. The latter one is an estimate, based on the value computed in the previous cooling water iteration loop. Also, the temperature of the cooling water bulk \((T_{cw, b})\) is based on a similar estimation. There are now three independent equations with two unknowns: the temperature at either side of the wall. These temperatures are isolated by combining and rearranging equations. The wall temperatures for the working fluid and cooling water side are given respectively as:

\[
T_{w f, w} = \frac{(\alpha_w^* + \alpha_{w f, w}^*) \cdot T_{w f, b} + T_{cw, b}}{1 + \alpha_w^* + \alpha_{w f, w}^*}
\] (3.121)

\[
T_{cw, w} = \frac{(1/\alpha_w^* + \alpha_{cw, w}^*) \cdot T_{cw, b} + T_{w f, b}}{1/\alpha_w^* + \alpha_{cw, w}^*}
\] (3.122)

In the relations above, three heat transfer ratios are defined. The ratio of heat transfer in the working fluid to the heat transfer in the cooling water \(\left(\alpha_w^*\right)\), the ratio of heat transfer
in the working fluid to the wall heat transfer (\(\alpha_{wf,w}^*\)) and the ratio of the cooling water heat transfer to the wall heat transfer (\(\alpha_{cw,w}^*\)) are given respectively as:

\[
\alpha_{w}^* = \frac{\alpha_{wf}}{\alpha_{cw}}
\]

(3.123)

\[
\alpha_{wf,w}^* = \frac{\alpha_{wf} d_p}{\lambda_w}
\]

(3.124)

\[
\alpha_{cw,w}^* = \frac{\alpha_{cw} d_p}{\lambda_w}
\]

(3.125)

A check is performed to see ensure that the energy balance across the wall is satisfied. This is done by setting up the energy balances within the wall and near the wall at the working fluid side and computing the absolute difference in heat flux across the wall (\(q_{w,\text{diff}}\)):

\[
q_{w,\text{diff}} = \alpha_{wf,w} \cdot (T_{wf,w} - T_{wf,b}) - \frac{\lambda_w}{d_p} \cdot (T_{cw,w} - T_{wf,w})
\]

(3.126)

The difference relative to the heat flux in the working fluid to the wall is given by:

\[
q_{w,\text{diff}}^* = \frac{q_{w,\text{diff}} \cdot A_w}{Q_{wf,w}}
\]

(3.127)

The heat transfer process across the wall is represented schematically in Figure 3.14. Similar parameters as for the heat transfer across the interface are indicated. Also, the heat flow through the wall itself (\(Q_w\)) and the thermal conductivity of the wall (\(\lambda_w\)) are indicated.

![Figure 3.14: Representation of the heat transfer process across the wall between the working fluid and cooling water channels of the heat exchanger. For absorption or condensation, the heat flows (Q) will be in the opposite direction of the indicated arrows, and the values are thus negative.](image)

**Overall and Species Mass Balance for the Control Volume**

The overall mass balance of the control volume is to be solved as long as the flow is still two-phase. In case the flow is fully condensed, all parameters related to mass and heat transfer across the interface are set to zero. The mass flow rates of both phases out of the control volume, in case that the flow is still two-phase, are calculated by:

\[
\dot{m}_{L,CV,out} = \dot{m}_{L,CV} - (\dot{m}_{i,1} + \dot{m}_{i,2})
\]

(3.128)

\[
\dot{m}_{V,CV,out} = \dot{m}_{V,CV} + (\dot{m}_{i,1} + \dot{m}_{i,2})
\]

(3.129)
3.6. SOLUTION PROCEDURE

The mass fractions out of the control volume of both phases are obtained by setting up and rearranging the mass balance for one of either species within each phase inside the control volume. The mass balances for ammonia inside the liquid and vapor phase respectively are given by:

\[ x_{b,\text{out}} \dot{m}_{L,\text{CV},\text{out}} = x_b \dot{m}_{L,\text{CV}} - \dot{m}_{i,1} \]  
(3.130)

\[ y_{b,\text{out}} \dot{m}_{V,\text{CV},\text{out}} = y_b \dot{m}_{V,\text{CV}} + \dot{m}_{i,1} \]  
(3.131)

The new mass fractions of ammonia inside the liquid and vapor phase respectively are then given by:

\[ x_{b,\text{out}} = x_b \frac{\dot{m}_{L,\text{CV}} - \dot{m}_{i,1}}{\dot{m}_{L,\text{CV},\text{out}}} \]  
(3.132)

\[ y_{b,\text{out}} = y_b \frac{\dot{m}_{V,\text{CV}} + \dot{m}_{i,1}}{\dot{m}_{V,\text{CV},\text{out}}} \]  
(3.133)

Schematic representations of the overall and species mass balance within the working fluid channel control volume are given respectively in Figure 3.15 and Figure 3.16. Mass flows of either phase are indicated at the inlet and outlet, as well as the species mass flows across the interface. The same is true for the species mass fractions in Figure 3.16. The molar fractions are calculated from the mass fractions using Equation A.21. The outlet vapor quality is calculated using Equation A.27.

Figure 3.15: Representation of the overall mass balance within a control volume of the working fluid channel of the heat exchanger. For absorption or condensation, the mass flows across the interface (\( \dot{m}_{i} \)) will be in the opposite direction of the indicated arrows, and the values are thus negative.

OVERALL ENERGY BALANCE FOR THE CONTROL VOLUME

The overall heat transfer coefficient (\( U \)) for transport of energy from the working fluid bulk to the cooling water bulk is obtained by taking into account the three heat transfer resistances encountered in series:

\[ U = \frac{1}{\frac{1}{\alpha_{cw,w}} + \frac{d_p}{\lambda_w} + \frac{1}{\alpha_{w,f,w}}} \]  
(3.134)

The three resistances encountered correspond to the three regions of heat transfer, described previously when discussing the heat transfer across the wall. The heat flow through a single wall within the control volume is then given by:
Figure 3.16: Representation of the species mass balance within a control volume of the working fluid channel of the heat exchanger. For absorption or condensation, the mass flow across the interface ($\dot{m}_i$) will be in the opposite direction of the indicated arrows, and the values are thus negative.

$$Q_w = U \cdot A_w \cdot (T_{cw,b} - T_{wf,b}) \quad (3.135)$$

This heat flow should be equal to the heat flows within each individual region discussed:

$$Q_w = Q_{w,\text{cond}} = Q_{wf,w} = Q_{cw,w} \quad (3.136)$$

The heat flows into the control volume of either phase are given by:

$$Q_{L,\text{in},CV} = \dot{m}_{L,\text{CV}} h_{L,b,in} \quad (3.137)$$

$$Q_{V,\text{in},CV} = \dot{m}_{V,\text{CV}} h_{V,b,in} \quad (3.138)$$

The total heat flow into the control volume is the sum of the contributions of both phases:

$$Q_{\text{in},CV} = Q_{V,\text{in},CV} + Q_{L,\text{in},CV} \quad (3.139)$$

The outlet bulk enthalpy of the vapor and liquid phases are obtained by setting up and rearranging the overall energy balance for each phase. The energy balance for respectively the liquid and vapor phase are given by the following equations, where the left hand side gives the heat flow at the outlet of the control volume:

$$Q_{L,\text{out},CV} = h_{L,b,\text{out}} \dot{m}_{L,\text{out},CV} = Q_{L,\text{in},CV} - Q_{L,i} + Q_w \quad (3.140)$$

$$Q_{V,\text{out},CV} = h_{V,b,\text{out}} \dot{m}_{V,\text{out},CV} = Q_{V,\text{in},CV} + Q_{V,i} \quad (3.141)$$

The phase bulk specific enthalpies at the outlet are then given by:

$$h_{L,b,\text{out}} = \frac{Q_{L,\text{in},CV} - Q_{L,i} + Q_w}{\dot{m}_{L,\text{out},CV}} \quad (3.142)$$

$$h_{V,b,\text{out}} = \frac{Q_{V,\text{in},CV} + Q_{V,i}}{\dot{m}_{V,\text{out},CV}} \quad (3.143)$$
The total heat flow out of the control volume is given as the sum of the outlet heat flows of both individual phases:

\[ Q_{out, CV} = Q_{V, out, CV} + Q_{L, out, CV} \quad (3.144) \]

The outlet temperature of each phase is calculated using FluidProp through the obtained outlet enthalpy value, mass fraction value and pressure in the control volume:

\[ T_{L, out} = f(P, h_{L, b, out}, x_b) \quad (3.145) \]

\[ T_{V, out} = f(P, h_{V, b, out}, y_b) \quad (3.146) \]

In Figure 3.17, a schematic representation is given of the total and phase energy balance within the working fluid control volume. A part of the cooling water channel is also indicated to emphasize the driving force behind the heat transfer. Parameters are indicated at the inlet, outlet or in the bulk, and include mass flow rates, specific enthalpies, heat flows and temperatures.

![Figure 3.17: Representation of the overall energy balance within a control volume of the working fluid channel of the heat exchanger. For absorption or condensation, the heat flows across the wall and interface (Qw and Qi) will be in the opposite direction of the indicated arrows, and the values are thus negative.](image)

**PRESSURE DROP IN THE CONTROL VOLUME**

The pressure drop in each control volume follows from the contributions of friction, elevation and deceleration. Equations are presented in section 3.3. Since the pressure drop on the working fluid side is very small, the effects are by default neglected in the model by leaving the pressure unchanged for each control volume.

**STORAGE OF VALUES AND NEW CONTROL VOLUME VALUES**

After all the calculations for a control volume are completed, all parameters are saved in a Matlab structure. The control volume outlet parameters that were calculated are set as the inlet values of the next control volume. The fluid properties in the next control volume are based on these values, and are calculated through the equations presented in Appendix A, section A.2. The mass flow rates of the phases and species are calculated through the equations in subsection 3.6.3. All other relevant parameters are calculated next, through the combination of relations discussed in the previous sections.
3.6.5. COOLING WATER MAIN CALCULATIONS

The main steps in the cooling water loop are presented in Figure 3.18. The mass flow rate in a single cooling water channel is given by Equation 3.68. The area available for heat transfer is the same as that for the working fluid side, and can be expressed as:

\[ A_w = \Gamma \cdot \Delta z \cdot B_p \]  

(3.147)

The mass flux is calculated in a similar way as for the working fluid. Fluid properties are calculated using the relations given in Appendix A. In case pure water is used relations are found in subsection A.2.1, or in subsection A.2.4 when water with a certain content of salts is used. The actual velocity, Reynolds and Prandtl number are calculated using similar relations as for the working fluid. The friction factor, Nusselt number, heat transfer coefficient and pressure drop in the control volume are calculated using the relations for single phase flow, presented in sections 3.3 and 3.4. The temperature at the wall is determined in a similar way to the working fluid, only in this case the estimates of the working fluid bulk temperature \( (T_{w,f,b}) \) and working fluid heat transfer coefficient near the wall \( (\alpha_{w,f,w}) \) are used, obtained from the values computed in the previous working fluid iteration loop. Also, the approach towards the energy balance is similar to the working fluid side. The outlet enthalpy of the control volume is thus calculated by:

\[ h_{out} = h_{in} + \frac{Q_w}{m_c} \]  

(3.148)

Based on the enthalpy and pressure at the control volume outlet, the outlet temperature is calculated using FluidProp:

\[ T_{out} = f(P, h_{out}) \]  

(3.149)

All parameter values are saved and the outlet values of the pressure, temperature and enthalpy are used as inlet values of the next control volume.

**Figure 3.18:** The substeps related to the computation of the cooling water loop parameter values in the solution algorithm of the model.

3.6.6. OVERALL HEAT EXCHANGER PARAMETERS

In the following paragraphs, calculation procedures for certain overall parameters of the heat exchanger are described. These parameters are used either as a check to ensure that the overall energy balance is satisfied, or as a performance parameter for the entire heat exchanger. The first is the case for the resulting wall heat flux deviation, described next. This value is computed for every iteration loop, to check if the solution is close to convergence.
After sufficient convergence is reached, the overall energy balance for each control volume and for an entire channel is set up for both the working fluid and cooling water side, to compare values on either side. When the energy balance is satisfied, the overall heat transfer coefficient of the heat exchanger is computed to quantify the overall performance of the heat exchanger, and to compare the value with expected values from literature or experiments.

**Resulting Wall Heat Flux Deviation**

After the loops are completed for all control volumes of the working fluid and cooling water side, the difference in the energy balance through the wall is calculated based on the values of the total wall heat flow of both streams. The relative deviation between the total heat flows across the wall \( q^*_{dev,w,tot} \) is calculated by taking the sum of the heat flow difference divided by the working fluid side heat flow for each control volume:

\[
q^*_{dev,w,tot} = \sum_{i=1}^{\gamma} \frac{Q_{wf,w,i} - Q_{cw,w,i}}{Q_{wf,w,i}}
\]

The average relative deviation \( q^*_{dev,w,avg} \) is obtained by dividing the previous equation by the number of control volumes \( \gamma \):

\[
q^*_{dev,w,avg} = \frac{q^*_{dev,w,avg}}{\gamma} = \frac{1}{\gamma} \sum_{i=1}^{\gamma} \frac{Q_{wf,w,i} - Q_{cw,w,i}}{Q_{wf,w,i}}
\]

The iterations are stopped after the computed deviation drops below a certain specified value. Besides the difference in computed heat flux, the absolute value of the heat flow across the wall of either stream should be computed in each iteration step. This is used to check if these parameters converge to a constant value, and do not continue to drift to lower or higher values during the course of the iterative loop, even though the difference between them has become small. The absolute values of the total heat flow across the wall for the working fluid and cooling water side \( Q_{wf,w,tot} \) and \( Q_{cw,w,tot} \) are given respectively by the following equations:

\[
Q_{wf,w,tot} = \sum_{i=1}^{\gamma} Q_{wf,w,i}
\]

\[
Q_{cw,w,tot} = \sum_{i=1}^{\gamma} Q_{cw,w,i}
\]

**Overall Energy Balance**

When the iterations are stopped, the values of all parameters are saved for each control volume, but only for the last iteration loop. To check if all calculations are performed correctly, the overall energy balances are set up for both the working fluid and cooling water side, and the computed changes in heat flow from inlet to outlet of either stream are compared. The heat flow change between the inlet and outlet of the control volume within a phase is computed by:

\[
\Delta Q = \dot{m}_{CV, out}h_{out} - \dot{m}_{CV, in}h_{in} = \dot{m}_{CV, out}c_{p, out}T_{out} - \dot{m}_{CV, in}c_{p, in}T_{in}
\]
This relation is valid because the pressure change ($\Delta P$) within the control volume is negligible, which makes the pressure term in the specific enthalpy relation also negligible as shown in the next equation:

$$dh = c_p dT + v(1 - \beta T)dP \approx c_p dT$$  \hfill (3.155)

Here, $dh$, $dT$ and $dP$ are the changes in enthalpy, temperature and pressure over an infinitesimal change in position. $v$ is the specific volume and $\beta$ is the volumetric expansion coefficient. Heat flow changes within the control volume liquid and vapor phase of the working fluid are thus given by:

$$\Delta Q_{wf,L} = \dot{m}_{wf,L,\text{CV},\text{out}}c_{p,\text{p},L,\text{out}}T_{wf,L,\text{out}} - \dot{m}_{wf,L,\text{CV},\text{in}}c_{p,\text{p},L,\text{in}}T_{wf,L,\text{in}}$$

$$= \dot{m}_{wf,L,\text{CV},\text{out}}h_{wf,L,\text{out}} - \dot{m}_{wf,L,\text{CV},\text{in}}h_{wf,L,\text{in}}$$  \hfill (3.156)

$$\Delta Q_{wf,V} = \dot{m}_{wf,V,\text{CV},\text{out}}c_{p,\text{p},V,\text{out}}T_{wf,V,\text{out}} - \dot{m}_{wf,V,\text{CV},\text{in}}c_{p,\text{p},V,\text{in}}T_{wf,V,\text{in}}$$

$$= \dot{m}_{wf,V,\text{CV},\text{out}}h_{wf,V,\text{out}} - \dot{m}_{wf,V,\text{CV},\text{in}}h_{wf,V,\text{in}}$$  \hfill (3.157)

The total heat flow changes within the control volume for the working fluid and cooling water are given by:

$$\Delta Q_{wf} = \Delta Q_{wf,L} + \Delta Q_{wf,V}$$  \hfill (3.158)

$$\Delta Q_{cw} = \dot{m}_{cw,\text{CV},\text{in}}c_{p,\text{cw},\text{in}}T_{cw,\text{in}} - \dot{m}_{cw,\text{CV},\text{out}}c_{p,\text{cw},\text{out}}T_{cw,\text{out}}$$

$$= \dot{m}_{cw,\text{CV},\text{in}}h_{cw,\text{in}} - \dot{m}_{cw,\text{CV},\text{out}}h_{cw,\text{out}}$$  \hfill (3.159)

The cumulative heat flow change for all control volumes is then computed for both streams:

$$\Delta Q_{wf,\text{tot}} = \sum_{i=1}^{\gamma} \Delta Q_{wf,i}$$  \hfill (3.160)

$$\Delta Q_{cw,\text{tot}} = \sum_{i=1}^{\gamma} \Delta Q_{cw,i}$$  \hfill (3.161)

Similarly, cumulative heat flow changes for the working fluid liquid and vapor phase ($\Delta Q_{wf,L,\text{tot}}$ and $\Delta Q_{wf,V,\text{tot}}$) are computed. The difference in cumulative heat flows between the streams is calculated through the following equation:

$$\Delta Q_{tot} = \Delta Q_{wf,\text{tot}} - \Delta Q_{cw,\text{tot}}$$  \hfill (3.162)

The heat flows are calculated only for a single set of control volumes along the channel length, which includes half an entire working fluid and cooling water channel, and the wall between the channels. To calculate the total amount of heat transferred within the heat exchanger, the previous heat flows need to be multiplied with the amount of plates ($N_p$) to account for all channels:
\[ \Delta Q_{wf,HX} = \Delta Q_{wf,tot} \tilde{N}_p \] (3.163)

\[ \Delta Q_{cw,HX} = \Delta Q_{cw,tot} \tilde{N}_p \] (3.164)

In this case it is assumed that no heat transfer occurs between the environment and the outer two channels, where the cooling water is present. Also, the same symmetry assumption is used for the flow, heat and mass transfer in the outer two channels. In practice this does not hold, because on the inner side of the channel heat exchange with the working fluid takes place through the wall, where on the outer side the wall is assumed to be adiabatic.

**OVERALL MASS BALANCE**

The mass balance is checked in each control volume of the working fluid by taking the change of mass flow between the inlet and outlet for either phase:

\[ \Delta \dot{m}_{wf,L} = \dot{m}_{wf,L,CV,\text{out}} - \dot{m}_{wf,L,CV,\text{in}} \] (3.165)

\[ \Delta \dot{m}_{wf,V} = \dot{m}_{wf,V,CV,\text{out}} - \dot{m}_{wf,V,CV,\text{in}} \] (3.166)

The sum of both changes should be zero, since the mass can not leave the control volume other than through the fluid flow inlet and outlet. No accumulation of mass takes place, since steady-state is assumed:

\[ \Delta \dot{m}_{wf} = \Delta \dot{m}_{wf,L} + \Delta \dot{m}_{wf,V} = 0 \] (3.167)

If the mass balance is zero for each control volume, the mass balance for the sum of all control volumes should also be zero. This is the mass balance over the entire working fluid channel:

\[ \Delta \dot{m}_{wf,tot} = \sum_{i=1}^{\gamma} \Delta \dot{m}_{wf,i} = \sum_{i=1}^{\gamma} (\Delta \dot{m}_{wf,L} + \Delta \dot{m}_{wf,V}) = 0 \] (3.168)

**OVERALL HEAT TRANSFER COEFFICIENT**

The overall heat transfer coefficient of the heat exchanger is calculated for each control volume separately, as previously described. The logarithmic mean temperature difference (LMTD) method may be used for larger sections of the heat exchanger. An assumption in this method is that the fluid properties and mass flow rate are constant over the heat exchanger or heat exchanger section of interest. This assumption is not generally valid for the heat exchanger considered in this study. Especially at the working fluid side, this approximation is considered quite rough. It can however be used as a method to obtain estimates of the heat transfer performance in the absorption and subcooling section, or for the entire heat exchanger. If full condensation of the working fluid is predicted in the model to occur at control volume \( \zeta \), the length of the condensation section \( L_{\text{cond}} \) is given by:

\[ L_{\text{cond}} = \frac{\zeta}{\gamma} L_p \] (3.169)
Here, \( \gamma \) is the total number of control volumes and \( L_p \) is the total length of a plate or channel. The terms ‘absorption’ and ‘condensation’ are used interchangeably in this section, since all relations are valid for either phenomenon. The length of the subcooled section is then given by the following equation:

\[
L_{\text{subc}} = L_p - L_{\text{cond}}
\]  

The total enlarged heat transfer area in the heat exchanger \((A_{p,\text{tot}})\) is given by Equation A.92. The enlarged heat transfer areas of the condensation section \((A_{\text{cond}})\) and the subcooled section \((A_{\text{subc}})\) are given by the following equations:

\[
A_{\text{cond}} = \frac{L_{\text{cond}}}{L_p} A_{p,\text{tot}} = \frac{\zeta}{\gamma} A_{p,\text{tot}}
\]

\[
A_{\text{subc}} = \frac{L_{\text{subc}}}{L_p} A_{p,\text{tot}} = \frac{\gamma - \zeta}{\gamma} A_{p,\text{tot}}
\]

The temperature of the working fluid is defined at the inlet of the working fluid as \( T_{w,f,\text{top}} \) and at the outlet as \( T_{w,f,\text{bot}} \). The subscripts ‘\text{top}’ and ‘\text{bot}’ are chosen to respectively indicate the top side and bottom side of the heat exchanger, to avoid confusion, since the inlet and outlet of the working fluid and cooling water channel are on opposite sides. The temperatures at the location inside the heat exchanger where the full condensation occurs for either stream are given as \( T_{w,f,\text{FC}} \) and \( T_{c,w,\text{FC}} \). The temperature differences between the working fluid and cooling water side at each of these points are given by the following equations:

\[
\Delta T_{\text{top}} = T_{w,f,\text{top}} - T_{c,w,\text{top}}
\]

\[
\Delta T_{\text{bot}} = T_{w,f,\text{bot}} - T_{c,w,\text{bot}}
\]

\[
\Delta T_{\text{FC}} = T_{w,f,\text{FC}} - T_{c,w,\text{FC}}
\]

The logarithmic mean temperature difference \( T_{LMTD} \) for each section is defined as function of the temperature difference at both end points of that section:

\[
LMTD_{\text{cond}} = \frac{\Delta T_{\text{top}} - \Delta T_{\text{FC}}}{\ln \left( \frac{\Delta T_{\text{top}}}{\Delta T_{\text{FC}}} \right)}
\]

\[
LMTD_{\text{subc}} = \frac{\Delta T_{\text{FC}} - \Delta T_{\text{bot}}}{\ln \left( \frac{\Delta T_{\text{FC}}}{\Delta T_{\text{bot}}} \right)}
\]

\[
LMTD_{\text{tot}} = \frac{\Delta T_{\text{top}} - \Delta T_{\text{bot}}}{\ln \left( \frac{\Delta T_{\text{top}}}{\Delta T_{\text{bot}}} \right)}
\]

The values of the heat flow across the wall in the absorption and subcooled section are given as the sum of the heat flows within all control volumes in the specified section, multiplied by the number of plates across which the heat transfer takes place:
3.6. Solution Procedure

\[ Q_{w,\text{cond}} = N_p \cdot \sum_{i=1}^{\zeta} Q_{w,i} \]  

(3.179)

\[ Q_{w,\text{subc}} = N_p \cdot \sum_{i=\zeta}^{\gamma} Q_{w,i} \]  

(3.180)

The total heat flow across the wall in the heat exchanger is given as:

\[ Q_{w,\text{tot}} = N_p \cdot \sum_{i=1}^{\gamma} Q_{w,i} = Q_{w,\text{cond}} + Q_{w,\text{subc}} \]  

(3.181)

The overall heat transfer coefficients are then calculated by rearranging the general heat transfer relation \( Q = U A \Delta T \):

\[ U_{\text{cond}} = \frac{Q_{w,\text{cond}}}{A_{\text{cond}} LMTD_{\text{cond}}} \]  

(3.182)

\[ U_{\text{subc}} = \frac{Q_{w,\text{subc}}}{A_{\text{subc}} LMTD_{\text{subc}}} \]  

(3.183)

\[ U_{\text{tot}} = \frac{Q_{w,\text{tot}}}{A_{\text{p,tot}} LMTD_{\text{tot}}} \]  

(3.184)
NUMERICAL RESULTS

In section 4.1, the results of the model are presented for a selected base case. The model results were produced before any experiments were performed, since the planned experiments were delayed by several months. The results of the model are compared to models implementing empirical correlations for two-phase flows with absorption or condensation in section 4.2. A sensitivity analysis is discussed for two cases that are a modification of the base case in section 4.3. Finally, in section 4.4, a future outlook for the modelling and experimental work is given. This section includes a critical reflection on the assumptions made in the model and the added value of the model, and presents several recommendations for future work.

4.1. NUMERICAL BASE CASE

The base case corresponds to test 1 in the experimental plan, which is presented in Table 5.4 in the chapter on the experimental results. The ammonia mass fraction is $w_{NH_3} = 96\%$, this is the predicted optimum in terms of the cycle performance by Kirkenier (2014) [3]. The working fluid mass flow rate ($\dot{m}_{wf}$) is set to 0.0045 kgs$^{-1}$. This value is chosen since it results in the model predicting that full condensation does not occur in the channel and there is thus no liquid subcooling, but the working fluid is close to being fully condensed towards the end of the channel. This corresponds to an effective use of the heat exchanger. Full condensation relatively early in the channel is unwanted for practical applications, since it implies that a subcooled liquid is present in a large section of the channel. Relatively little heat will be exchanged in that section. The other settings for the base case are related to the cooling water and hot water mass flow rates, temperatures and salinity: $\dot{m}_{cw} = 0.079$ kgs$^{-1}$, $\dot{m}_{hw} = 0.204$ kgs$^{-1}$, $T_{cw} = 5^\circ C$, $T_{hw} = 29^\circ C$, $S = 0$ gkg$^{-1}$. These values are based on the values of test 1 in the study by Gudjonsdottir (2015) [98].

Results for the base case are discussed in the following subsections, starting with the overall energy balances in the system. Next, the text will move into the other relevant mass and energy balances and the resulting profiles for a selection of parameters.

4.1.1. ENERGY BALANCES AND HEAT FLOWS

The base case was run for the model settings summarized in Table 3.1. The geometrical details of the heat exchanger are kept the same as the specifics of the heat exchanger used for the experimental work.
Wall heat flux difference

After 21 iterations the relative average wall heat flux difference ($q_{dev, w, avg}^*$) is below 0.1%. The heat flux difference of each iteration step is monitored for both streams, and the same is done for the total heat flow across the wall computed for both streams ($Q_{w, w, tot}$ and $Q_{cw, w, tot}$). The results are presented in figures 4.1 and 4.2. The computation of these parameters is presented in subsection 3.6.6, Equation 3.150 and Equation 3.151, and the calculation loops of the working fluid and cooling water stream are discussed respectively in subsection 3.6.4 and subsection 3.6.5. The relative heat flux difference at the working fluid side is set manually to 100% for the first iteration, since an actual value cannot be computed before the model is run for the cooling water side. This explains the peak observed for the relative difference on the working fluid side (red line). The calculated relative differences on the working fluid side are in the first iterations higher than those of the cooling water side. The relative difference increases for each working fluid loop from the cooling water loop in the iteration steps until the 10th. It is slightly difficult to observe this from the figure, but the heat flux difference value of the red line (working fluid) for step number $i + 1$ is higher than the level of the blue line (cooling water) for step number $i$, until $i = 10$.

The direct cause for this trend is not known, but could be related to the higher numerical complexity of the working fluid side. After the 10th iteration step, the relative differences calculated for either side become close and eventually very small, after 21 or more iterations (<0.1%).

For the total heat flow across the heat exchanger walls, which can also be described as the ‘heat exchanged’, it is observed that the values calculated on either side approach each other and move towards a constant value in the final iteration steps in Figure 4.2. The heat flow values are negative since the positive heat flow is defined in the direction from the cooling water towards the working fluid side in the model control volumes, similar to the approach described in Sieres and Fernández-Seara (2007) [78]. Therefore, heat flows will effectively be negative for absorption or condensation and positive for desorption or evaporation. On the cooling water side, the value starts to move quickly towards the constant value in the first few iteration steps, and the profile gradually flattens out in the later iterations steps. The working fluid profile starts to move towards the constant value in the first iterations steps as well, but a relatively flat profile is observed from the 2nd up to the 10th iteration step. After the 10th iteration, the value starts to move more quickly towards the constant value, before flattening out again as it approaches the constant value. The constant heat flow value of 4049 W represents the final estimate of the total heat exchanged by the entire heat exchanger.

Total and wall heat flows

The methods used to compute the overall energy balance and overall heat transfer coefficient are discussed in subsection 3.6.6. Results are only obtained and discussed for the last iteration step of the simulation, since the remaining difference in the energy balances are at the lowest point here and the solution of the simulation is considered converged. The cumulative heat flow changes or ‘heat gained’ of the cooling water ($\Delta Q_{cw, cum}$) and working fluid ($\Delta Q_{w, f, cum}$) stream are presented in Figure 4.3. The presented values are those per channel, and in total 8 working fluid and 9 cooling water channels are present. The total value of the heat exchanged is thus over 4 kW, which corresponds to the presented value in Figure 4.2. Also, the heat gained of the liquid ($\Delta Q_{w, f, L, cum}$) and vapor ($\Delta Q_{w, f, V, cum}$) phase of the working fluid are presented in the figure. The profiles of the total heat flows of ei-
ther stream should be exactly opposite, since all the heat lost by the working fluid steam is gained by the cooling water stream. The final or total cumulative heat flow change values should be of the same magnitude but also opposite, which is the case in the figure. Heat is transferred from the liquid phase of the working fluid to the cooling water across the wall. However, since mass is absorbed from the vapor phase to the liquid phase the mass flow is increased and, as a result, effectively the heat flow increases along the length of the heat exchanger. The vapor phase of the working fluid constantly loses heat and mass along the length of the heat exchanger, as is observed in the heat flow profile.

The heat flow values of the working fluid and cooling water in and out of the control volume are presented in Figure 4.4. The values are based on equations 3.137 to 3.141 and 3.144 for the working fluid side and similar equations for the cooling water side. The values are presented for each control volume, consisting of half the working fluid and cooling water channel, separated by a single plate. In total, 18 plates are present in the heat exchanger, and also from this figure the calculated total value of the heat exchanged is more than 4 kW. Considering the large amount of control volumes, the difference between inlet and outlet values is not observed in the graph, and both appear to lie on the same line. The profiles for the working fluid heat flow ($Q_{wf,CV}$) and cooling water heat flow ($Q_{cw,CV}$) appear to be similar, but are shifted by a certain constant value along the y-axis. This confirms that the overall and control volume energy balances are satisfied, as an equal amount of heat should leave the working fluid and enter the cooling water in each control volume. The heat flow of the working fluid is the sum of the liquid ($Q_{wf,L,CV}$) and vapor phase ($Q_{wf,V,CV}$) contributions. The removal of heat from the vapor phase through absorption of ammonia is the main contribution to the exchanged heat. It is observed that the heat content of the vapor phase is reduced by a large amount over the channel length. The energy content of the vapor phase is reduced by 83%, from 375 W at the inlet to 65 W at the outlet of a single channel.

**Working Fluid Interface Heat Flows**

Heat flows related to the transport across the working fluid interface are presented in Figure 4.5, calculated based on Equation 3.104 up to 3.111. This includes the total heat flow across the interface, heat flows to and from the interface from the bulk phases, and species
4. NUMERICAL RESULTS

and phase heat flows associated with the transport across the interface. The heat flow associated with the transport of water (species 2) across the interface in either phase \(Q_{L,1,2}\) and \(Q_{V,1,2}\) is negligible and thus close to zero. The same is true for the heat flow that results from heat transfer in the vapor phase from the bulk to the interface \(Q_V\). The heat flow resulting from the heat transfer of the liquid interface to the bulk \(Q_L\) is of a higher magnitude. The total heat flows in either phase associated with transport across the interface \(Q_{L,i}\) and \(Q_{V,i}\) follow the same profile, as these two should be equal everywhere to satisfy the energy balance. The heat flow associated with the transport of ammonia (species 1) across the interface from the vapor phase \(Q_{V,1,1}\) is approximately equal to the total heat flow associated with transport across the interface in either phase, since it is the only significant contribution to the total heat flow of all three contributions given in Equation 3.105.

The heat flow associated with the transport of ammonia into the liquid phase \(Q_{L,1,1}\) is smaller in magnitude. Together with \(Q_L\) these two contributions add up to the total liquid phase heat flow across the interface, following Equation 3.104.

4.1.2. WORKING FLUID MASS FLOW RATES AND PHASE COMPOSITIONS

PHASE MASS FLOW RATES

The profiles of the mass flow rate within the control volume of the working fluid liquid and vapor phase \(\dot{m}_{w_f,LCV}\) and \(\dot{m}_{w_f,LCV}\), and of the total working fluid are presented in Figure 4.6. The total mass flow remains constant, since the system is in steady-state and the mass can not escape the channels other than through the inlet and outlet. The same amount of mass that enters a control volume thus also leaves the control volume within a set time. The total mass flow rate should equal the sum of the contributions of both phases. The profiles of either phase should be opposite, mirrored along the horizontal line representing half the total mass flow rate. It is observed in the figure that this is the case. This implies that the working fluid overall mass balance is satisfied for each control volume and thus for the entire channel and the whole heat exchanger.
4.1. Numerical base case

Species mass flow rates across the interface
The mass flow rates of both species across the interface in the working fluid are presented in Figure 4.7. It is observed that the transport of water (species 2) is negligible when compared to ammonia (species 1). This is expected, since the vapor phase consists of nearly pure ammonia. The magnitude of the mass flow rate across the interface increases along the length of the channel, due to an increasing driving force for mass transfer. This driving force is the difference between interface and bulk concentration in the phases of the working fluid.

Molar fractions at the bulk and interface
The molar fractions of the liquid and vapor phase (\(\bar{x}\) and \(\bar{y}\)) in the bulk and at the interface are presented in Figure 4.8. The liquid phase ammonia molar fractions at both the bulk (\(\bar{x}_b\)) and interface (\(\bar{x}_i\)) increase throughout the length of the working fluid channel, as a result of the absorption of ammonia into the liquid phase. The ammonia fraction at the interface is determined by the assumed thermodynamic equilibrium at the interface (Equation A.22), and the ammonia fraction in the bulk is based on the species and total mass balance for each control volume (Equation 3.132). The molar fractions of ammonia in the vapor phase bulk (\(\bar{y}_b\)) and at the interface (\(\bar{y}_i\)) are indistinguishable, and both appear to be equal to unity throughout the length of the heat exchanger. When zooming in on the molar fraction profiles in the vapor phase, it is observed that they follow a similar trend to the molar fractions in the liquid phase. The molar fractions at the interface and in the vapor phase bulk are calculated respectively by Equation A.23 and Equation 3.132. The molar fraction of ammonia is higher at the interface. Referring back to figures 3.9 and 3.10, it is concluded that the situation is typical of absorption or condensation (case 2).
4. NUMERICAL RESULTS

Figure 4.6: Working fluid total mass flow rate and mass flow rates of both phases at the control volume inlet and outlet, as function of the position inside the heat exchanger.

Figure 4.7: Working fluid mass flow rates of both species across the interface in each control volume, as function of the position inside the heat exchanger.

VAPOR QUALITY

Since the largest part of the ammonia from the vapor phase is absorbed throughout the length of the heat exchanger, the vapor quality of the working fluid ($X_{wf}$) is reduced by a large amount. At the inlet the vapor fraction is 0.74 and at the outlet 0.09, which implies that more than 88% is absorbed. The outlet value is close to zero, at which point the working fluid would become fully condensed. The vapor quality of the working fluid and cooling water are presented in Figure 4.9. The cooling water is a pure liquid throughout the heat exchanger, so the vapor quality is zero everywhere.

Figure 4.8: Molar fractions of the working fluid liquid and vapor phase at both the interface and bulk, as function of the position inside the heat exchanger.

Figure 4.9: Vapor quality of the working fluid and cooling water, as function of the position inside the heat exchanger.

4.1.3. TEMPERATURE AND HEAT TRANSFER COEFFICIENT PROFILES

Temperatures

Figures 4.10 and 4.11 present the temperatures of the working fluid and cooling water respectively as function of the length in the heat exchanger and as function of the heat exchanged between the streams for the entire heat exchanger. Temperatures are given in the
bulk of the phases, on either side of the wall and at the working fluid vapor-liquid interface. The profiles of the working fluid bulk vapor phase and interface are different from what is expected. The vapor phase temperature at a certain point inside the heat exchanger, roughly half way through, drops below the liquid phase temperature. It continues to drop faster than the temperatures of the liquid phase bulk and the wall, and eventually drops below the wall temperature at the working fluid side. The profile stays just above the profile of the temperature at the cooling water side wall, and flattens out towards the end of the heat exchanger. The interface temperature lies above the liquid and vapor phase bulk temperatures. Initially, the profile decreases slowly and starts to flatten out roughly 75% into the channel, after which the temperature starts to rise.

Most of the heat is exchanged at the bottom of the heat exchanger, towards the working fluid outlet and cooling water inlet, which is observed by the steeper temperature profile of the cooling water bulk phase ($T_{cw,b}$). Also, the working fluid liquid phase bulk temperature ($T_{wf,l,b}$) and cooling water bulk temperature are much further apart at the bottom, corresponding to a larger driving force for heat transfer. Qualitatively, the two profiles are similar, but the temperature difference between inlet and outlet is quite different for both streams. For the working fluid flow, the bulk temperatures of the liquid and vapor are different. An average bulk temperature may be specified, but the liquid phase bulk temperature should provide a good indication of this temperature. At the inlet, both temperatures are assumed equal, and near the outlet the vapor quality is low. The temperatures at either side of the wall ($T_{wf,w}$ and $T_{cw,w}$) follow a similar profile to the liquid phase bulk profiles on either side. Both profiles lie close, due to the high heat transfer across the wall.

Figure 4.10: Temperatures of the working fluid and cooling water in the bulk phase, at the wall or at the phase interface, as function of the position inside the heat exchanger.

Figure 4.11: Temperatures of the working fluid and cooling water in the bulk phase, at the wall or at the phase interface, as function of the heat exchanged between the two streams for the entire heat exchanger.

**Heat Transfer Coefficients**

The heat transfer coefficients in each of the working fluid and cooling water phases are presented in Figure 4.12, based on the calculations discussed in section 3.4. The heat transfer coefficient in the working fluid ($\alpha_{wf}$) is the heat transfer coefficient of the liquid phase ($\alpha_{wf,l}$), since liquid contacts the walls everywhere on the working fluid side. Values of this heat transfer coefficient and the heat transfer coefficient at the cooling water side ($\alpha_{cw}$)
Numerical results are based on the loops of either side, discussed in subsections 3.6.4 and 3.6.5. The profiles should overlap, which is observed in the figure. Also, the overall heat transfer coefficient \( (U) \) is calculated based on both loops, and the two profiles overlap as well. Based on Figure 4.12, the overall heat transfer coefficient is estimated to be roughly \( 3 \cdot 10^3 \text{ Wm}^{-2} \text{K}^{-1} \) along the whole length of the heat exchanger. The average value of \( U \) can also be estimated using Equation 3.184, based on the LMTD method discussed at the end of subsection 3.6.6. The value is then \( 1.87 \cdot 10^3 \text{ Wm}^{-2} \text{K}^{-1} \). The calculated value using the average of the overall heat transfer coefficients in each control volume is large when compared to the predictions based on the model in combination with experimental results in the work by Gudjonsdottir (2015) [98]. The values agree better with the estimation using the LMTD method. Application of the LMTD method is strictly speaking not valid for absorption or condensation in heat exchangers, because of the latent heat associated with phase changes processes. However, the method is used in literature for computation of overall heat transfer coefficients of heat exchangers, including absorbers and condensers. A direct comparison can not be made between the results of the studies, since the presented results in the study by Gudjonsdottir (2015) [98] are based on an experiment using slightly different operating conditions (test 3) and pure ammonia instead of the ammonia-water mixture. In chapter 6 of the work by Gudjonsdottir (2015) [98], predictions are also made for the use of the ammonia-water mixture, but under different conditions than in this work. The presented figure in the study (6.10) with the temperature profiles of the condenser cooling water and working fluid as function of the heat exchanged does show large similarities with the observed profiles in this work.

It is observed in Figure 4.12 that the vapor phase heat transfer is the lowest. This does not directly affect the amount of heat that is transported across the wall to the other stream, but it does affect the heat transfer across the interface to the liquid phase. The heat transfer coefficient of the working fluid liquid phase is higher than that of the cooling water. The overall heat transfer coefficient is lower, since it encompasses the total heat transfer resistance caused by three different regions within the control volume. Based on the calculated heat transfer coefficients, the heat transfer resistances are compared in the different regions of the control volume. The following four regions for heat transfer are distinguished:

1. From the working fluid vapor bulk to the interface.

2. From the working fluid interface to the working fluid liquid bulk, which is the same heat transfer resistance as from the working fluid bulk to the wall at the working fluid side.

3. From the wall at the working fluid side across the wall to the cooling water side.

4. From the wall at the cooling water side to the cooling water liquid bulk.

With the exception that fouling is not considered in this work, these four regions correspond to those described in Tovazhnyanski and Kapustenko (1984) [50] and Tovazhnyansky et al. (2004) [92], as discussed in subsection 2.4.2 and 2.7.1 respectively. The relative ratios of heat transfer coefficients are used to calculate the temperature at the vapor-liquid interface and at either side of the wall. These calculations are discussed in subsection 3.6.4, and the relative heat transfer parameters are calculated through equations 3.114, 3.123, 3.124 and 3.125. Results for the values of the heat transfer ratios are presented in Figure 4.13.
4.1. Numerical base case

Figure 4.12: Heat transfer coefficients related to both streams based on both the working fluid and cooling water side, as function of the position inside the heat exchanger.

Figure 4.13: Heat transfer ratios at the specified regions around the working fluid interface and the wall, based on local heat transfer on both the working fluid and cooling water side, as function of the position inside the heat exchanger.
4.1.4. OTHER PARAMETERS RELATED TO HEAT TRANSFER

Before discussing the mass transfer effects, some additional parameters are discussed related to the heat transfer. This includes the film thickness, velocities, friction factors and Reynolds numbers.

FILM THICKNESS AND VELOCITIES

The liquid phase heat transfer in case of the two-phase film flow is determined by the film thickness \(d_{film}\) of the liquid film. This characteristic flow length scale, and that of the vapor phase flow \(d_V\) are calculated through equations 3.46 and 3.49 respectively. Both are presented in Figure 4.14, together with the channel gap \(d_g\) for comparison. The sum of two films thicknesses, one on either side of the channel, and the vapor phase characteristic length should together equal the channel gap. In case the working fluid fully condenses, the whole channel becomes filled with liquid. The actual velocities of the liquid and vapor phase \(u_{act,L}\) and \(u_{act,V}\) are related to the characteristic flow lengths of either phase, through equations 3.82 and 3.83. Throughout the whole channel the actual velocity of the vapor phase is higher than that of the liquid phase, and both values approach each other towards the outlet. This is expected, since vapor flows in the center of the channels, and the liquid flows in a thin film along the wall. Towards the end of the channel the vapor phase is almost completely absorbed, and the velocity is reduced as result of the reduced volumetric flow rate \(\dot{V}_V\). The actual velocities are plotted together with the superficial velocities of both working fluid phases \(u_{sup,L}\) and \(u_{sup,V}\) in Figure 4.15. The superficial velocities are smaller than the actual velocities, since they are based on the cross-sectional area of the channel \(A_c\) or the channel gap, and not on the characteristic flow length scales of the phases. For single phase flow, the superficial and actual velocity are equal, as is the case for the cooling water flow.

![Figure 4.14: Film thickness of the liquid phase film, characteristic flow length scale of the vapor phase flow and the average channel gap, as function of the position inside the heat exchanger.](image1)

![Figure 4.15: Superficial and actual velocities of the working fluid liquid and vapor phase and the cooling water, as function of the position inside the heat exchanger.](image2)

FRICTION FACTOR AND REYNOLDS NUMBER

For the single phase flow, the heat transfer depends on the friction factor. The frictional pressure drop also depends on the friction factor. This dependency is only implemented
at the cooling water side, since the pressure drop in the working fluid is assumed negligible. The calculated pressure drops are discussed further ahead in this section. The values of both the Fanning friction factor \( f \) and Darcy friction factor \( \xi \) are presented in Figure 4.16. The friction factor of the working fluid \( f_{wf} \) that is used in the model is the relation proposed in Yan et al. (1999) [68], presented in Equation 3.10. Alternatively, the friction factor may be calculated using Equation A.121, based on the work by Han et al. (2003) [99]. The second relation predicts a much larger friction factor than the first relation. The relation of the cooling water friction factor for single phase flow is given in the VDI Heat Atlas ([8]), and the calculations are discussed in section 3.3, equations Equation 3.16 to Equation 3.27. The proposed relations are valid for a limited range of Reynolds numbers. The Reynolds numbers of the cooling water \( (Re_{cw}) \), the working fluid liquid and vapor phase \( (Re_{wf,L} \) and \( Re_{wf,V} \)) and equivalent Reynolds number \( (Re_{eq,wf}) \) are presented in Figure 4.17. The cooling water friction factor relations for laminar flow \( (Re < 2000) \) are used, and the values of the cooling water Reynolds number are well within the specified limits in the heat exchanger. For the working fluid, the friction factor relation by Yan et al. (1999) [68] is valid for \( Re_{eq,wf} < 6000 \), which is the case throughout the whole heat exchanger. The relation by Han et al. (2003) [99] is valid within the range \( 250 < Re < 750 \). This holds for the largest part of the heat exchanger. Only towards the working fluid channel outlet, the Reynolds number drops below this minimum value.

4.1.5. Cumulative Pressure Drop

The pressure drop in the working fluid and cooling water channels also depends on the friction factor. The total cumulative pressure drop of the working fluid and the cooling water \( (\Delta P_{tot,cum}) \) are presented respectively in Figure 4.18 and Figure 4.19. The cumulative pressure drop is set to zero at the inlet of either channel. A positive cumulative pressure drop value implies that the pressure decreases along the direction of the flow. For the working fluid, the contributions of deceleration \( (\Delta P_{dec,cum}) \) and friction \( (\Delta P_{ele,cum}) \) result in an increasing pressure drop along the direction of the flow, and the contribution due to elevation \( (\Delta P_{ele,cum}) \) to a decrease in pressure drop. Since the flow is vertically downwards, the hydrostatic pressure increases from inlet to outlet, corresponding to the decreasing pressure

Figure 4.16: Darcy and Fanning friction factors \( (\xi = 4f) \) of the working fluid and cooling water, as function of the position inside the heat exchanger.

Figure 4.17: Reynolds numbers of the working fluid and cooling water flow, as function of the position inside the heat exchanger.
drop. The flow decelerates from the channel inlet to outlet and experiences friction with the channel walls, which both result in a decrease in pressure and thus an increase in pressure drop. For the cooling water flow, the frictional pressure drop and elevation pressure drop are positive, since the liquid also experiences friction from the walls but the liquid is moved vertically upwards. In terms of both absolute and relative magnitude of the pressure drop, the working fluid side pressure drop is low. The total cumulative pressure drop in the channel is 220 Pa. Relative to an absolute pressure of 6.732 bar this is just over 0.03%, and thus assumed negligible. In the cooling water channel, the total pressure drop is close to 3000 Pa on an absolute pressure of around 1 bar. This is 3%, and the pressure drop in each control volume is accounted for in the model.

**Figure 4.18:** Total cumulative pressure drop of the working fluid, and the cumulative contributions due to elevation, deceleration and friction, as function of the position inside the heat exchanger.

**Figure 4.19:** Total cumulative pressure drop of the cooling water, and the cumulative contributions due to elevation and friction, as function of the position inside the heat exchanger.

### 4.1.6. Mass Transfer Limitations

Returning to the mass transfer in the working fluid, the ratio of the liquid phase mass transfer coefficient \( k_L \) to the vapor phase mass transfer coefficient \( k_V \) is presented in Figure 4.20. It is observed that the liquid phase mass transfer coefficient is lower than the vapor phase mass transfer coefficient throughout the channel by at least a factor 4. The liquid phase mass transfer is thus expected to be the limiting factor in the mass transfer across the interface during absorption. This observation is discussed in more depth in section 4.3.

The effects of heat and mass transfer in a phase may be compared by defining parameter \( \kappa \), relating the total mass transfer to total heat transfer in a phase:

\[
\kappa = \frac{Sh \cdot Pr}{Nu \cdot Sc} = Le^{-2/3}
\]  

(4.1)

The Prandtl, Nusselt and Schmidt numbers were discussed previously in the text. The general definitions of the Sherwood (\( Sh \)) and Lewis (\( Le \)) number are given respectively in equations A.10 and A.5 in Appendix A. The two expressions in the previous equation are related through the Chilton-Colburn analogy between heat and mass transfer. By computing the value in each control volume of the simulation for both the liquid and vapor phases, it can be determined whether mass transfer is expected to significantly limit the heat transfer. This is the case when the Lewis number is large \( (Le \gg 1) \) or the value of \( \kappa \) is small \( (\kappa \ll 1) \).
The values of $\kappa$ in the working fluid vapor and liquid phase along the length of the heat exchanger are presented in Figure 4.21. In the liquid phase, the mass transfer is lower than the heat transfer and mass transfer is thus expected to limit the actual heat transfer rate. In the vapor phase the mass transfer is higher than the heat transfer, and heat transfer is expected not to be limited by the mass transfer.

**4.1.7. RELATIVE HEAT AND MASS TRANSFER RESISTANCES**

The relative heat transfer resistances of the working fluid and cooling water near the wall and in the wall itself are presented in Figure 4.22. The heat transfer resistance or thermal resistance ($R$) is defined as the reciprocal of the heat transfer coefficient:

$$ R = \frac{1}{\alpha} $$

In computing the total heat transfer resistance across the wall ($R_{w,tot}$), the three resistances encountered in series are added, which is equivalent to taking the reciprocal of the overall heat transfer coefficient:

$$ R_{w,tot} = \frac{1}{U} = R_{cw,w} + R_{w} + R_{wf,w} = \frac{1}{\alpha_{cw,w}} + \frac{d_p}{\lambda_w} + \frac{1}{\alpha_{wf,w}} $$

(4.3)

Heat transfer resistances are defined relative to the total heat transfer resistance. As an example, the relative heat transfer resistance of the cooling water ($R_{cw,w}^*$) for the heat transfer across the wall is:

$$ R_{cw,w}^* = \frac{R_{cw,w}}{R_{w,tot}} $$

(4.4)

In the figure, the resistances are presented as function of the length in the heat exchanger for 50 different control volumes. The top of the y-axis represents the top of the channels, which translates to the working fluid inlet and cooling water outlet. The relative magnitude of each resistance is represented by different colors, and the total sum of all resistances is always equal to unity. It is observed that the largest resistance for heat transfer
across the wall is on the cooling water side, and the relative thermal resistance slightly decreases towards the bottom of the heat exchanger. The smallest thermal resistance is in the wall, and the relative magnitude of this resistance is approximately constant along the length of the heat exchanger.

A similar figure is constructed for the heat transfer resistances on either side of the interface, presented in Figure 4.23. In this figure, the heat transfer resistances relative to the total heat transfer resistance across the interface ($R_{t, tot}$) are given. The liquid phase heat transfer is thought to be limited by the mass transfer, which is in line with the observations reported in Thonon and Bontemps (2002) [69] for low Reynolds numbers. In the vapor phase, the effect of mass transfer on the heat transfer is smaller, and considered insignificant. The relative mass transfer resistance is indicated together with the remaining resistance. Both are of roughly equal magnitude, since the value $\kappa V \approx 2$ along the working fluid channel. It is observed that the relative thermal resistance of the liquid phase for heat transfer across the interface first increases along the working fluid channel, but quickly stops increasing and starts to decrease slowly. Over the whole channel, the vapor phase resistance is much larger than the liquid phase resistance, as is also expected from the comparatively low heat transfer coefficient in the vapor phase.

![Figure 4.22: Total relative thermal resistances of the cooling water, wall and working fluid for the total heat transfer across the wall, as function of the position inside the heat exchanger.](image1)

![Figure 4.23: Working fluid relative thermal resistances of the liquid and vapor phase for the total heat transfer across the interface, as function of the position inside the heat exchanger.](image2)

### 4.1.8. SURFACE TENSION EFFECTS

Surface tension is assumed negligible in the model for the working fluid flow. To check this assumption, three dimensionless numbers related to surface tension are calculated for each control volume as described in subsection 3.4.1, equations 3.50 to 3.53. The resulting values of the two-phase Eötvös and Capillary numbers ($Eo$ and $Ca$) are presented in Figure 4.24 and Figure 4.25. The profiles of the liquid and vapor phase Weber numbers ($We_L$ and $We_V$) are presented in Figure 4.26. As stated before, the effects of surface tension in comparison to the effects of respectively gravity, viscous forces and inertia are relevant for $Eo < 1$, $Ca < 10^{-5}$ and $We < 1$. The Eötvös number is in the range $2.39 - 2.54$ and thus outside the range where surface tension is considered relevant compared to the gravity effects in the flow. The Capillary number is at its lowest point equal to $2 \cdot 10^{-4}$, which is higher than...
the range for which surface tension is considered relevant compared to viscous forces. The Weber numbers of both the liquid and vapor phase are small enough to be considered relevant in comparison to the effects of inertia. The effects of inertia itself are however small compared to the viscous forces, which is apparent from the low Reynolds numbers of either phase, discussed previously. It is concluded that the surface tension and inertia effects are negligible compared to the dominant effects of gravity and viscous forces in the working fluid flow through the heat exchanger.

Figure 4.24: The working fluid Eötvös number, as function of the position inside the heat exchanger.  
Figure 4.25: The working fluid Capillary number, as function of the position inside the heat exchanger.  
Figure 4.26: The working fluid Weber number, as function of the position inside the heat exchanger.
4.1.9. Full condensation

For the previous case discussed, the model does not predict the working fluid to fully condense within the heat exchanger. For other cases, the model will predict full condensation. If the mass flow rate of the working fluid ($\dot{m}_{wf}$) is decreased, the retention time of the working fluid in the channels is longer and thus more time is available for heat exchange with the cooling water. Alternatively, if the inlet pressure of the working fluid is increased, or the inlet temperature is decreased, full condensation will occur sooner since the initial vapor quality is lower. Another approach is decreasing the relative ammonia content of the system, since the absorption of ammonia from the vapor to the liquid phase releases most of the energy. A second case is run using the same conditions as the previous base case, but changing the ammonia content to $w_{\text{NH}_3} = 95\%$. This case is discussed in section 4.3. Other parameters are monitored within the heat exchanger control volumes besides those that were discussed in this section, such as fluid properties, dimensionless numbers, and other quantities related to heat and mass transfer. Some of these are discussed further ahead in this chapter. Other parameters for the base case are presented in Appendix C.

4.2. Comparison with Empirical Models

In several studies heat transfer correlations are developed for condensing two-phase flow in plate heat exchangers, for pure fluids or for mixtures (absorption). In this study, the detailed model is compared to these direct two-phase correlations. These are all derived empirically based on a set of experiments using certain fluids under specified conditions, and a plate heat exchanger of specific geometry. Four studies are chosen, which are the same studies as discussed in the work by Gudjonsdottir (2015) ([35, 68, 69, 100]). Detailed specifics of each study are found in the original publications, and the most important heat transfer correlations are discussed in Appendix A, subsection A.4.1. The correlations were implemented into the numerical model, and for each control volume the values of the direct two-phase heat transfer coefficient ($\alpha_{TP}$), overall heat transfer coefficient ($U$) and heat flow across the wall ($Q_w$) are computed for different studies. The values are calculated using the parameter values predicted by the detailed model for each control volume. Total, mean, minimum and maximum predicted values over the entire heat exchanger are calculated, and relevant parameter values are compared to those predicted by the detailed model. Results are presented in Table 4.1.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Units</th>
<th>Model</th>
<th>Winkelmann</th>
<th>Difference</th>
<th>Yan et al.</th>
<th>Difference</th>
<th>Thonon and Bontemps</th>
<th>Difference</th>
</tr>
</thead>
<tbody>
<tr>
<td>$Q_{w,\text{tot}}$</td>
<td>W</td>
<td>$-245$</td>
<td>$-257$</td>
<td>$8$</td>
<td>$-204$</td>
<td>$-41$</td>
<td>$-226$</td>
<td>$-17$</td>
</tr>
<tr>
<td>$Q_{w,\text{min}}$</td>
<td>mW</td>
<td>$-12.3$</td>
<td>$-12.7$</td>
<td>$0.4$</td>
<td>$-10.2$</td>
<td>$-2.0$</td>
<td>$-14.4$</td>
<td>$-0.8$</td>
</tr>
<tr>
<td>$Q_{w,\text{max}}$</td>
<td>mW</td>
<td>$-41.2$</td>
<td>$-43.1$</td>
<td>$1.9$</td>
<td>$-32.2$</td>
<td>$-0.9$</td>
<td>$-50.9$</td>
<td>$9.7$</td>
</tr>
<tr>
<td>$U$</td>
<td>W m$^{-2}$ K$^{-1}$</td>
<td>$2.99 \cdot 10^4$</td>
<td>$3.25 \cdot 10^4$</td>
<td>$-260$</td>
<td>$2.67 \cdot 10^4$</td>
<td>$320$</td>
<td>$2.58 \cdot 10^4$</td>
<td>$420$</td>
</tr>
<tr>
<td>$U_{\text{min}}$</td>
<td>W m$^{-2}$ K$^{-1}$</td>
<td>$2.72 \cdot 10^4$</td>
<td>$2.90 \cdot 10^4$</td>
<td>$-180$</td>
<td>$2.17 \cdot 10^4$</td>
<td>$550$</td>
<td>$2.18 \cdot 10^4$</td>
<td>$540$</td>
</tr>
<tr>
<td>$U_{\text{max}}$</td>
<td>W m$^{-2}$ K$^{-1}$</td>
<td>$3.14 \cdot 10^4$</td>
<td>$3.36 \cdot 10^4$</td>
<td>$-220$</td>
<td>$2.80 \cdot 10^4$</td>
<td>$340$</td>
<td>$3.43 \cdot 10^4$</td>
<td>$390$</td>
</tr>
<tr>
<td>$\alpha_{\text{in}}$</td>
<td>W m$^{-2}$ K$^{-1}$</td>
<td>$1.10 \cdot 10^4$</td>
<td>$1.56 \cdot 10^4$</td>
<td>$-4.56 \cdot 10^3$</td>
<td>$7.86 \cdot 10^3$</td>
<td>$3.36 \cdot 10^4$</td>
<td>$7.94 \cdot 10^4$</td>
<td>$3.10 \cdot 10^4$</td>
</tr>
<tr>
<td>$\alpha_{\text{min}}$</td>
<td>W m$^{-2}$ K$^{-1}$</td>
<td>$0.77 \cdot 10^4$</td>
<td>$1.10 \cdot 10^4$</td>
<td>$-2.17 \cdot 10^3$</td>
<td>$4.81 \cdot 10^3$</td>
<td>$3.96 \cdot 10^4$</td>
<td>$4.57 \cdot 10^4$</td>
<td>$4.20 \cdot 10^4$</td>
</tr>
<tr>
<td>$\alpha_{\text{max}}$</td>
<td>W m$^{-2}$ K$^{-1}$</td>
<td>$1.25 \cdot 10^4$</td>
<td>$1.69 \cdot 10^4$</td>
<td>$-4.46 \cdot 10^3$</td>
<td>$8.44 \cdot 10^3$</td>
<td>$4.04 \cdot 10^4$</td>
<td>$2.61 \cdot 10^4$</td>
<td>$-1.36 \cdot 10^4$</td>
</tr>
</tbody>
</table>
4.2.1. Heat Flow and Heat Transfer Coefficient

The profiles along the heat exchanger channels of the wall heat flow and the overall heat transfer coefficient are presented in Figure 4.27 and Figure 4.28. The overall heat transfer coefficient and the heat transfer coefficients of the liquid and vapor phase using the detailed model are plotted together with the direct two-phase heat transfer coefficients of the different approaches in Figure 4.29. When comparing the heat flow profiles between different approaches, the profile of the detailed model appears to correspond well with three of the alternative approaches. The method described by Palmer et al. (2000) [100] predicts values of the heat flow that are several orders of magnitude smaller than the other predictions, which was also the case for the different approaches compared in the work of Gudjonsdottir (2015) [98]. The results of this approach are left out of the discussion. For the predicted total heat transfer across the wall \( Q_{w,tot} \), the difference is smallest when compared to the approach by Winkelmann (2010) [35], of just 8 W per wall, which translates to just over 10%. The difference with Thonon and Bontemps (2002) [69] is 17 W or 7%, and 41 W or 17% with Yan et al. (1999) [68]. Differences in the mean, minimum or maximum predicted value of the heat flow across the wall \( Q_w \) in a control volume are also presented in Table 4.1. When comparing the results for the overall heat transfer coefficient \( (U) \) or the liquid and two-phase heat transfer coefficient \( (\alpha_L \text{ and } \alpha_{TP}) \) the profile predicted by the detailed model lies in between the profiles of Winkelmann (2010) [35] and Yan et al. (1999) [68], which show a similar trend. In Winkelmann (2010) [35], slightly higher values of the overall heat transfer coefficient are predicted of on average 9%. In Yan et al. (1999) [68] the predicted values are slightly lower, on average by 11%. The values are thus of the same order of magnitude, and the same is true when comparing values with the study of Thonon and Bontemps (2002) [69]. Here, the average difference is 14%, but the profile shows an opposite trend to the studies discussed previously.

![Figure 4.27: Heat flow across the wall using different methods, as function of the position inside the heat exchanger.](image1)

![Figure 4.28: Overall heat transfer coefficient using different methods, as function of the position inside the heat exchanger.](image2)

4.2.2. Recommendations

By comparing the correlations using the approach described above, conclusions could be drawn about to what extent they correspond in each control volume. To properly compare the consequences of implementation of each of the heat transfer correlations (and com-
Figure 4.29: The overall, liquid and vapor phase heat transfer coefficient of the model and two-phase direct heat transfer coefficients using different methods, as function of the position inside the heat exchanger.
4.3. Sensitivity analysis

A sensitivity analysis is performed for two cases that are based on the base case, but where a single parameter is set to another values. Both cases are discussed next, and some recommendations for a future sensitivity analysis are presented in section 4.4.

4.3.1. Mixture composition

As discussed before, lowering the ammonia content of the working fluid will cause the ammonia to be fully absorbed into the liquid phase sooner in the working fluid channels, and full condensation will result. Results of a case that was run using all the same settings and inputs as the base case, but with an ammonia mass fraction of $w_{NH_3} = 0.95$, are presented in figures 4.30 to 4.39. The simulation is run for 25 iteration steps, which is the same as in the base case. The resulting relative wall heat flux deviation ($q^*_dew,w,avg$) between both streams is however larger, 1.88% compared to 0.0053%. For this reason, in Figure 4.30 it is observed that both heat flow profiles do not end at the same level in the figure. A figure with the total heat flow across the wall ($Q_{w,tot}$) for each iteration step is not included for this case, but similarly as for the base case the profiles of both streams converge to a constant value. The convergence process is however slower, which results in the larger relative error after the same amount of iteration steps.

Heat and mass flows

The cumulative heat flow changes of the working fluid and cooling water flows ($Q_{wf,cum}$ and $Q_{cw,cum}$) are presented in Figure 4.31. Both profiles are similar, but appear to be mirrored in the x-axis, which indicates that the differences in the energy balances for each control volume are negligible. Just over 0.19 m (77%) into the channel, the working fluid fully condenses, which is observed by the flattening of the heat flow profiles. This indicates that little heat is still transferred between both streams after the working fluid has fully condensed, as is expected.

The profiles of the heat flow in and out of the control volume for the working fluid
and the cooling water ($Q_{w,f, CV}$ and $Q_{cw, CV}$) are presented in Figure 4.32 and the mass flow profiles of the working fluid liquid and vapor phase flow ($\dot{m}_{w,f,L, CV}$ and $\dot{m}_{w,f,V, CV}$) in Figure 4.33. For the heat flows, it is observed that the profiles for both streams flatten out at the point where full condensation of the working fluid occurs. At the same point, the heat flow of the working fluid vapor ($Q_{w,f,V, CV}$) becomes zero and the total heat flow of the working fluid becomes equal to the liquid phase heat flow ($Q_{w,f,L, CV}$). The difference between the starting and final heat flow values for either stream is approximately the same, which indicates a small difference between the energy balances. The mass flow profiles of the working fluid liquid and vapor phase are opposite. At the full condensation point the liquid phase mass flow rate becomes equal to the total mass flow rate of the working fluid ($Q_{w,f, CV}$), and the vapor phase mass flow rate becomes equal to zero.

The total heat exchanged is just over 2.7% higher for the base case with 95% ammonia in terms of mass fraction, as compared to the case discussed here with 96%. This is calculated using the total cumulative heat flow change ($Q_{cum, tot}$). The values are respectively 504 W and 490 W for a single channel, and these values may differ from the values that are computed when the 95% case is converged to the same extent as the 96% case. Apart from the total heat transferred, the total rate of heat transfer is compared. For the 95% case, the heat transferred at the full condensation point is 472 W. For the 96% case, the heat transferred at the same length in the heat exchanger is 233 W, which is only 49% of that value. The heat transfer up to the point of full condensation is thus roughly two times faster for the 95% case. This is mostly the result of the increased mass transfer across the interface for this case, due to the increased driving force.

**Temperature and heat transfer coefficient**

As observed in Figure 4.34, the working fluid vapor quality ($X_{w,f}$) is reduced from close to 0.7 at the start to zero at the full condensation point. The relevant temperature profiles are presented in Figure 4.35. At the full condensation point the temperatures of the working fluid vapor phase bulk ($T_{w,f,V,b}$) and of the working fluid interface ($T_{i}$) are discontinued, since these are no longer of physical significance. The temperatures at the wall and in the bulk of the liquids in both streams after full condensation start to drop rapidly. This is a
result of the decrease in heat that can still be transferred in the remainder of the channel. After this point, no more heat is transferred due to condensation or absorption of a vapor but only due to subcooling of the liquid. This also results in the temperature level of either stream to reach approximately the same value towards the end of the working fluid channel, close to the inlet temperature of the cooling water.

The overall heat transfer coefficient and other relevant heat transfer coefficients are presented in Figure 4.36, and the heat transfer ratios around the wall and interface are presented in Figure 4.37. After the working fluid is fully condensed, the vapor phase heat transfer coefficient \( \alpha_{w,f,v} \) is still computed but has no practical significance anymore. The correlations for the liquid phase heat transfer coefficient \( \alpha_{w,f,L} \) change from two-phase liquid film flow to single phase liquid flow at this point. This is observed in both figures by a large jump in the profiles that represent this heat transfer coefficient. In practice, it is more likely that the transition is gradual than sharp and instantaneous. The heat transfer coefficient becomes smaller by a factor of roughly 5 at the point of full condensation. This is explained by the fact that in the model the liquid changes from flowing in a thin liquid film to occupying the entire channel, since other two-phase flow patterns are not taken into account. The heat transfer thus suddenly has to occur over a much larger distance in the liquid phase, which increases the resistance to heat transfer and thus lowers the corresponding heat transfer coefficient. The overall heat transfer coefficient \( U \) is also affected by this sudden change, and becomes roughly a factor 2 smaller. The heat transfer coefficient of the cooling water side \( \alpha_{cw} \) is not directly affected by the full condensation point. Due to the strong decrease in working fluid liquid phase heat transfer coefficient, the cooling water heat transfer coefficient becomes dominant and the working fluid heat transfer becomes limiting on the overall heat transfer. Based on the LMTD method, the overall heat transfer coefficient of the condensation section is \( 2.1 \times 10^3 \) Wm\(^{-2}\)K\(^{-1}\), which is higher by 12% than for the base case. For the subcooled section, however, the overall heat transfer coefficient is \( 1.0 \times 10^3 \) Wm\(^{-2}\)K\(^{-1}\), which is roughly a factor 2 lower than for the base case and the condensation section.
Figure 4.34: Vapor quality of the working fluid and cooling water, as function of the position inside the heat exchanger, for 95% ammonia in the mixture.

Figure 4.35: Temperatures of the working fluid and cooling water in the bulk phase, at the wall or at the phase interface, as function of the position inside the heat exchanger, for 95% ammonia in the mixture.

Figure 4.36: Heat transfer coefficients related to both streams based on both the working fluid and cooling water side, as function of the position inside the heat exchanger, for 95% ammonia in the mixture.
4.3. Sensitivity Analysis

Figure 4.37: Heat transfer ratios at the specified regions around the working fluid interface and the wall, based on local heat transfer on both the working fluid and cooling water side, as function of the position inside the heat exchanger for 95% ammonia in the mixture.
Friction factor and Reynolds number

A sudden change in value is also observed for the friction factor of the working fluid \((f_{wf})\), presented together with the friction factor of the cooling water \((f_{cw})\) in Figure 4.38. Here, the correlation for the working fluid friction factor changes from two-phase (described in Yan et al. (1999) [68]) to single-phase (described in Stephan et al. (2010) [8]). The friction factor is enlarged at full condensation, roughly by a factor of 20. This large value is explained by the fact that the liquid flow changes from occupying a small film near the wall to fully occupying the entire channel. The liquid will experience more internal shear or friction, thus increasing the friction factor. The Reynolds number profiles of the different streams and phases are presented in Figure 4.39. The working fluid vapor phase Reynolds number \((Re_{wf,V})\) becomes zero at full condensation, since the velocity also becomes zero. The liquid phase Reynolds number \((Re_{wf,l})\) becomes equal to the two-phase equivalent Reynolds number \((Re_{wf,eq})\) and remains approximately stable at a value just above 100. The cooling water Reynolds number \((Re_{cw})\) also becomes stable after slowly decreasing, at a value just below 100. Both streams may thus be assumed laminar throughout the channel, both before and after the working fluid fully condenses.

Figure 4.38: Darcy and Fanning friction factors of the working fluid and cooling water, as function of the position inside the heat exchanger for 95% ammonia in the mixture.

Figure 4.39: Reynolds numbers of the working fluid and cooling water flow, as function of the position inside the heat exchanger for 95% ammonia in the mixture.

Recommendations

It is recommended to compare the results of the model to cases for other working fluid compositions in the high ammonia fraction range \((90 – 100\%)\). The values of \(95 – 96\%\) used in this work are based on the work by Kirkenier (2014) [3]. This is based on the full OTEC cycle analysis, but the optimum value specifically for use in the absorber may differ from this value. The developed numerical model is at the moment not capable of use with pure ammonia, but some modifications to the code may allow this. Attempts were made to run the model for higher ammonia mass fractions \((> 98\%)\). This led to instabilities in the solution procedure, which caused the simulation to run slowly or in certain cases stop as result of an error. The exact cause for this is not known and was not investigated. This is an interesting study for the future, along with an investigation on how the model should be modified to work with pure ammonia, or ammonia concentrations in the limit of \(w_{NH_3} \rightarrow 100\%\). Also,
4.3. Sensitivity Analysis

An investigation can be made on how to modify the model to make the profiles of the friction factors, heat transfer coefficients, and other resulting parameters continuous without any large, sudden jumps in values in case full condensation occurs.

4.3.2. Vapor Phase Mass Transfer

The mass transfer coefficient of the working fluid vapor phase \( k_V \) in the base case is higher than that of the liquid phase \( k_L \). Depending on the position in the heat exchanger, the difference factor is between 4 and 60. The ratio of both coefficients is presented in Figure 4.20. In Figure 4.21, the ratio of the total mass transfer to the heat transfer is presented, showing that in the vapor phase the mass transfer is not expected to significantly limit the heat transfer. The mass transfer coefficient of the vapor is increased manually by a factor 10 in another run of the model, to compare the results with the base case. All other settings and inputs are kept the same. The goal is to investigate if the vapor phase mass transfer resistance is negligible, which may be concluded if the results of both cases are similar. Several parameter plots for this case are presented in figures 4.40 to 4.45.

Mass Transfer Coefficients

In Figure 4.40, the ratio of both mass transfer coefficients in the working fluid is given. As expected, this ratio of \( k_L \) to \( k_V \) is a factor 10 smaller than in the base case (Figure 4.20), since the vapor phase mass transfer coefficient is increased by a factor 10. The heat transfer resistances on either side of the interface are presented in Figure 4.41. The heat transfer in the liquid phase is still considered fully limited by the mass transfer. In the vapor phase, the heat transfer resistance that results from mass transfer is reduced further, and no heat transfer limitations are expected to arise as a result of the mass transfer. The ratio of the total vapor phase heat transfer resistance to the total liquid phase heat transfer resistance is the same as for the base case.

![Figure 4.40: Ratio of the liquid phase to the vapor phase mass transfer coefficient, as function of the position inside the heat exchanger, for an enlarged vapor phase mass transfer coefficient.](image)

![Figure 4.41: Working fluid relative thermal resistances of the liquid and vapor phase for the total heat transfer across the interface, as function of the position inside the heat exchanger, for an enlarged vapor phase mass transfer coefficient.](image)

Other Parameters

In Figure 4.42, the cumulative heat flow changes of the working fluid and cooling water are given. The final values are the same as for the base case, where the profiles are presented...
in Figure 4.3. The profiles of the different heat flows also overlap, which indicates that the vapor phase mass transfer does not play a limiting role in either of the cases. The profiles of the species mass transfer across the interface, vapor quality and temperatures are given in Figure 4.43, Figure 4.44 and Figure 4.45 respectively. All of the observed profiles are similar to the base case, increasing the vapor phase mass transfer coefficient does not affect the process in the heat exchanger.

![Figure 4.42: Cumulative heat flow changes of the working fluid and cooling water channels (two control volumes), as function of the position inside the heat exchanger, for an enlarged vapor phase mass transfer coefficient.](image1)

![Figure 4.43: Working fluid mass flow rates of both species across the interface in each control volume, as function of the position inside the heat exchanger, for an enlarged vapor phase mass transfer coefficient.](image2)

![Figure 4.44: Vapor quality of the working fluid and cooling water, as function of the position inside the heat exchanger, for an enlarged vapor phase mass transfer coefficient.](image3)

![Figure 4.45: Temperatures of the working fluid and cooling water in the bulk phase, at the wall or at the phase interface, as function of the position inside the heat exchanger, for an enlarged vapor phase mass transfer coefficient.](image4)

**Recommendations**
The results do not correspond with the statement made in Panchal and Kuru (1997) [89] or Thonon and Bontemps (2002) [69], that the vapor phase mass transfer is controlling in the ammonia-water absorption system. However, in the study by Panchal and Kuru (1997) [89],
a different geometry was used (vertical tube) under different operating conditions. Another case could be studied where the vapor phase mass transfer is set to an even higher value, equal or close to infinity, so that the mass transfer resistance effectively becomes zero. This is equivalent to a perfectly mixed vapor phase, but this is not expected to significantly influence the results of the simulation. In the study by Sieres and Fernández-Seara (2007) [78], mass and heat transfer corrections are described. These are introduced to compensate for the distorted velocity, composition and temperature profiles by the transferred species. Values of the factors were computed using the relations described in the study, but these were not implemented to avoid additional complexity of the model. Besides, the influence of implementation of these factors turned out to be minimal, only 1-2% change at most in the original values of the heat and mass transfer coefficients was calculated. The effect of implementation of these factors may be studied after a proper validation of the model using experimental data is made.

4.4. Future outlook

In this section some critical reflections are made on the results of the model, combined with recommendations for the modelling work. It should be investigated if the assumption of direct change from film flow to single phase flow is valid, since this causes sudden jumps in the computed values of certain parameters. A smoother transition between the flow patterns and computed parameters is expected in practice. This could also be achieved in the model by defining a certain transition regime, where new correlations are set up that allow a more gradual change of the predicted profiles of the friction factor and heat transfer coefficient. As a result, profiles of other parameters will also follow a more gradual transition. A better approach might be to first investigate which flow patterns occur in the heat exchanger in practice under the studied conditions, following the recommendations made in subsection 2.5.3.

4.4.1. Model assumptions

Reflecting on the other assumptions made in section 3.2, the assumption of a steady-state process is also valid in case the model is compared to experiments where the fluctuations of input parameters are kept to a minimum. The flow in the working fluid channels is controlled by the effects of gravity and viscous forces (shear stress, friction). The assumption of no heat losses in the heat exchanger should be validated by obtaining reliable data of the inlet and outlet temperatures, pressures and mass flows of both streams through the heat exchanger. By setting up the overall mass balance across the heat exchanger and for both streams individually, the heat that is lost can be estimated. To investigate if the flow is distributed equally amongst the channels in the heat exchanger, flow meters should be positioned in the channels of the absorber. This might be difficult, since the flow meters should be very small to fit in the channels. Also, if they would fit, they should not significantly affect the flow in the channels and may thus need to be even smaller. It is also possible to study the flow distribution through visualization of the flow and determining the flow rates in different channels through imaging techniques. This could be achieved by making the casing of the plate heat exchanger from a transparent material, or only part of the casing on one side, through which the flow can then be visualized.
4.4.2. Added value of the model

The predictions of the model point out that the ammonia-water mixture is a suitable candidate for use as working fluid, regarding the computed high overall heat transfer coefficient. The difference with the measured and calculated values from experiments is however rather large, as will be discussed in section 5.5, and conclusions can only be made after the numerical model is validated properly using experiments. If the model is able to reasonably predict the results of experiments, it may be used for optimization of the operating conditions for the absorber. Alternatively, the model may be used to predict the performance of other working fluids, for a certain composition under specified conditions. These predictions should also be validated by performing experiments with the corresponding working fluid. Through this approach, the expected added value of the model to other methods can be investigated. The main idea behind the developed model is that predictions should be valid for a larger range of working fluids and operating conditions if the model is based on fundamental transfer relations and mass and heat balances.

The model can also be extended to function as a design tool for the heat exchanger. Using the current model, the effects of varying geometrical parameters can already be studied. In other studies, the effects of geometrical parameters on the heat exchanger performance are discussed. The influence of the chevron angle is discussed in the work by Zwerver (2015) [9], and the effects of the ratios of channel spacing and channel width to the channel length in Longo et al. (2015) [51]. Another method may be used in the model for the prediction of the transferred heat in each control volume. In this work, the logarithmic mean temperature difference method is only used for computation of the overall heat transfer coefficient of the entire heat exchanger. The approach can also be implemented into the model for each control volume, to allow predictions that compensate for the differences in parameter values over the length of a control volume. However, it is expected that the control volume size is chosen small enough to give good predictions, and using the LMTD method requires an additional iterative loop in each control volume.

4.4.3. Further recommendations

In a first comparison, the results of the model compare well to results predicted using correlations that were developed specifically for certain working fluids under specified conditions. A discussion of the model results compared to a selection of these ‘lumped’ approaches, using direct heat transfer correlations for condensing two-phase flows, is found in section 4.2. Recommendations for further investigations are also included there. An important point that should be investigated is the computation of the interface temperature in the model. In the current implementation of the model the value is initially based on the liquid phase temperature, before it is calculated based on the relations for heat and mass transfer across the interface. When computing the interface temperature based on the conditions at the interface, instabilities arise in the calculations by the model. The profiles of the interface and vapor phase temperature predicted using the current settings in the model follow an unexpected trend towards the outlet of the working fluid channel, as is discussed in section 4.1. The causes of the instabilities and unexpected profiles should be investigated, and modifications should be implemented into the model to predict results that are reasonable.

Besides further investigation into the physical background of the model, also the numerical approach should be developed further. Recommendations related to this point
are:

- Solving both the cooling water and working fluid loops simultaneously, instead of repeating calculations in two separate loops, continuing until the difference between the predicted heat flow for both loops is zero.

- Alternatively, methods may be introduced to make the iteration loops converge faster, since the convergence process occurs slowly in certain studied cases.

- Reducing the amount of control volumes in the model led to instability of the model, and large oscillations in certain values. It could be investigated what the exact cause for these oscillations is and how these are avoided when reducing the amount of control volumes.

- Make the model run faster by combining sets of equations into matrix forms, reducing the amount of operations in the code, and possibly other improvements that do not change the actual calculations.

- Finding faster methods for fluid property computation can make the model simulations run faster, the computational speed of the model is at the moment limited mostly by the FluidProp calculations made for the ammonia-water mixture. A balance between accuracy, reliability and speed of fluid property calculations methods is required.
5

EXPERIMENTAL RESULTS

An experimental setup is used for validation of the numerical model of the absorber. In section 5.1 a short description of the experimental setup is given, including the relevant equipment and measurement devices. In section 5.2, the experimental plan is discussed, before moving into the actual experimental results and an uncertainty analysis respectively in section 5.3 and section 5.4. Finally, a preliminary validation of the model using the results of the model and experiments is presented in section 5.5.

5.1. EXPERIMENTAL SETUP

The experimental setup is used for research and development of OTEC technology, and is the result of a collaboration between Delft University of Technology and the company Bluerise B.V. The setup is used by different researchers, students or staff at the University and company. During the course of this work the setup was upgraded with several new components and sensors. The layout of the experimental setup will be briefly discussed through the Process Flow Diagram (PFD) and Process and Instrumentation Diagram (P&ID) next.

5.1.1. SETUP PFD AND P&ID

The existing experimental setup is based upon the concept of the Kalina cycle. All the relevant components of the Kalina cycle for use with OTEC in general have already been discussed in chapters 1 and 2. Here, the actual layout of the experimental setup is discussed. The PFD consists of all the important pieces of equipment in the system, connected by lines which indicate which fluid flows through them and in which direction the fluid flows. The pieces of equipment indicated in the PFD include the three heat exchangers (desorber, recuperator and absorber), three pumps for the three different fluids used in the system (working fluid, cooling water and heating water), the separator, the turbine, a manual valve and the mixing section. The valve allows control of the flow through the recuperator from the separator. The PFD is depicted in Figure 5.1.

For the remainder of this section, the focus lies on the part of the setup around the absorber, including the absorber itself. The P&ID of this part of the system is presented in Figure 5.2. Since this is only a part of the complete P&ID of the entire system, the origin or destination equipment is indicated at the cut-off lines. In this figure, relevant pieces of
Figure 5.1: PFD of the experimental setup. The different pieces of equipment are depicted, together with the lines connecting them. The legend explains what different line styles represent in the figure.
equipment and measurement devices are indicated with their code. The symbols in the code correspond to the type of equipment or measurement device, explained in Table 5.1.

Table 5.1: Overview of the experimental setup P&ID symbols

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>E</td>
<td>Heat exchanger</td>
</tr>
<tr>
<td>P</td>
<td>Pump (or line number)</td>
</tr>
<tr>
<td>T</td>
<td>Tank</td>
</tr>
<tr>
<td>V</td>
<td>Valve</td>
</tr>
<tr>
<td>Y</td>
<td>Filter</td>
</tr>
<tr>
<td>DI</td>
<td>Density indicator</td>
</tr>
<tr>
<td>LI</td>
<td>Level indicator</td>
</tr>
<tr>
<td>FI</td>
<td>Flow indicator</td>
</tr>
<tr>
<td>PI</td>
<td>Pressure indicator</td>
</tr>
<tr>
<td>TI</td>
<td>Temperature indicator</td>
</tr>
</tbody>
</table>

The number ensures each piece of equipment or measurement device has a unique code. Two flow meters located outside the selected area are presented in the P&ID, since these are necessary for measuring the flows of both streams through the absorber. The relevant measurable quantities are indicated near their indicators in the P&ID. It must be noted that in the actual setup the lines P-006 and P-013 are introduced separately into the absorber. P-006 and P-013 contain purely or mostly respectively vapor and liquid, and the liquid phase is sprayed onto the heat exchanger plates at the inlet of the channel. The vapor is fed to the heat exchanger in a regular way. Since the details of the spraying process are unknown and it will require a model on its own to describe the process, the simplification is made that both streams are perfectly mixed before entering the heat exchanger.

5.1.2. OPERATING CONDITIONS AND HEAT EXCHANGER GEOMETRY

The geometrical details of the equipment and lines in the system are known, and fixed during experiments. An overview of the geometrical details of the heat exchanger is presented in Table 5.2. Four different operating conditions are determined or measured independently that are required as input data for the model. These are the mass flow rate ($\dot{m}$), the pressure ($P$), the temperature ($T$) and the composition. For the working fluid the mixture overall mass or molar composition ($w$ or $\bar{w}$) is determined, and in case of the seawater the salinity ($S$). Values of the temperature and pressure are obtained for both streams, both at the inlet and outlet of the heat exchanger. The following quantities in the system are required as input data or as parameter for validation of the model, and are measured either directly or calculated based on other measured parameters:

- The absorber inlet and outlet temperature of the cooling water ($T_{cw,in}$ and $T_{cw,out}$)
- The absorber inlet and outlet temperature of the working fluid ($T_{wf,in}$ and $T_{wf,out}$)
- The absorber inlet and outlet pressure of the cooling water ($P_{cw,in}$ and $P_{cw,out}$)
- The absorber inlet and outlet pressure of the working fluid ($P_{wf,in}$ and $P_{wf,out}$)
- The mass flow rate of the cooling water in the absorber ($\dot{m}_{cw}$)
Figure 5.2: P&ID of the experimental setup, of the area around the absorber including the absorber itself. Besides the equipment, instrumentation and lines the measured parameters are indicated. At certain cut-off lines, the origin or destination equipment is indicated.
• The mass flow rate of the working fluid in the absorber ($\dot{m}_{wf}$)
• The salinity of the cooling water ($S$)
• The overall mass fraction of ammonia in the working fluid ($w_{NH_3}$)

Table 5.2: Overview of the experimental setup heat exchanger specifications, including all relevant quantities, their symbols and their values. The same values are used for the modelling work.

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Symbol</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of passes</td>
<td>$N$</td>
<td>1</td>
<td>–</td>
</tr>
<tr>
<td>Total number of channels</td>
<td>$N_c$</td>
<td>17</td>
<td>–</td>
</tr>
<tr>
<td>Number of cooling water channels</td>
<td>$N_{c,cw}$</td>
<td>9</td>
<td>–</td>
</tr>
<tr>
<td>Number of working fluid channels</td>
<td>$N_{c,wf}$</td>
<td>8</td>
<td>–</td>
</tr>
<tr>
<td>Total number of plates</td>
<td>$N_p$</td>
<td>18</td>
<td>–</td>
</tr>
<tr>
<td>Width of plates</td>
<td>$B_p$</td>
<td>0.111</td>
<td>m</td>
</tr>
<tr>
<td>Length of plates corrugated section</td>
<td>$L_p$</td>
<td>0.25</td>
<td>m</td>
</tr>
<tr>
<td>Plate thickness</td>
<td>$d_p$</td>
<td>0.4</td>
<td>mm</td>
</tr>
<tr>
<td>Compressed plate pitch</td>
<td>$p_{cp}$</td>
<td>2.4</td>
<td>mm</td>
</tr>
<tr>
<td>Corrugation amplitude</td>
<td>$\hat{a}$</td>
<td>1.0</td>
<td>mm</td>
</tr>
<tr>
<td>Channel gap</td>
<td>$d_g$</td>
<td>2.0</td>
<td>mm</td>
</tr>
<tr>
<td>Port diameter</td>
<td>$d_{port}$</td>
<td>23–30</td>
<td>mm</td>
</tr>
<tr>
<td>Area enlargement factor</td>
<td>$\Gamma$</td>
<td>1.17</td>
<td>–</td>
</tr>
<tr>
<td>Chevron angle to horizontal</td>
<td>$\varphi$</td>
<td>22</td>
<td>°</td>
</tr>
<tr>
<td>Chevron angle to flow</td>
<td>$\varphi_{flow}$</td>
<td>68</td>
<td>°</td>
</tr>
<tr>
<td>Channel cross-sectional area</td>
<td>$A_c$</td>
<td>$2.23 \cdot 10^{-4}$</td>
<td>m$^2$</td>
</tr>
<tr>
<td>Single plate surface area</td>
<td>$A_0$</td>
<td>0.0278</td>
<td>m$^2$</td>
</tr>
<tr>
<td>Enlarged plate surface area</td>
<td>$A_p$</td>
<td>0.0333</td>
<td>m$^2$</td>
</tr>
<tr>
<td>Total plate surface area</td>
<td>$A_{tot}$</td>
<td>0.5004</td>
<td>m$^2$</td>
</tr>
<tr>
<td>Total enlarged plate surface area</td>
<td>$A_{p,tot}$</td>
<td>0.5994</td>
<td>m$^2$</td>
</tr>
<tr>
<td>Hydraulic diameter</td>
<td>$d_h$</td>
<td>3.4</td>
<td>mm</td>
</tr>
<tr>
<td>Flow wetted perimeter</td>
<td>$\Pi$</td>
<td>0.266</td>
<td>m</td>
</tr>
</tbody>
</table>

5.1.3. Measurement devices

Five types of measurement devices present in the system are discussed here: for pressure, temperature, mass or volumetric flow rate rate, density and liquid level. The last one is present in the system for measurement of the liquid level in the working fluid buffer tank. The sensors used for each type of measurement are discussed here. The same sensors and procedures are used as in the work of Gudjonsdottir (2015) [98], with a few additions or modifications.

**Pressure sensors**

The pressure sensors are all gauge pressure sensors produced by Kobold, of type SEN-3272 with an internal diaphragm. Each sensor has a span of 0 to 10 bar gauge, with an accuracy of 0.5% within their span. The sensors work up to 100°C. The pressure sensors were calibrated by using all the different sensors for the same pressure measurement, and choosing the sensor that gave the pressure values closest to the average as the reference sensor.
Certain pressure sensors were added to the system as compared to the study by Gudjonsdottir (2015) [98] which was performed using the same setup. The same type of sensors are used.

**Temperature sensors**
The temperature sensors are resistance thermometers produced by TC Direct of the type PT-100 class B. The temperature sensors have a tolerance of $\pm 0.3^{\circ}C$ around $0^{\circ}C$ and $\pm 0.8^{\circ}C$ around $100^{\circ}C$. Calibration was done using a custom made adapter plate with a reference calibrated PT-100 element with an accuracy of $0.01^{\circ}C$, and up to 8 other sensors. All sensors were placed in the adapter plate, and a four point measurement was done of the resistance and temperature of each sensor for a thermostatic bath at $5^{\circ}C$, $15^{\circ}C$, $25^{\circ}C$ and $35^{\circ}C$. The same sensors are used as in Gudjonsdottir (2015) [98], only some are added in new locations in the system and the code of the sensor is changed in certain cases.

**Flow sensors**
There are three relevant flow sensors in the system: one for the cooling water volumetric flow rate (Fi-101), one for the working fluid pure vapor phase mass flow rate (Fi-201) and one for the working fluid pure liquid phase mass flow rate (Fi-202). The flow meter for the cooling water is a DF140 flow sensor produced by AE Sensors, with a 3/4” size. The measuring range is $30 – 3000 \text{ Lh}^{-1}$, which for water translates to approximately $0.0083 – 0.83 \text{ kgs}^{-1}$. The accuracy of the sensors is $\pm 1\%$ over their full range.

The flow meter for the vapor phase mass flow rate is a pure ammonia thermal bypass mass flow meter of type T14-AGD-44-E, produced by Bronkhorst. The measuring range at $T = 26.5^{\circ}C$ and $P = 9.5$ bar is $0.0001 – 0.005 \text{ kgs}^{-1}$, with an accuracy of $\pm 1\%$ over the full scale plus $\pm 1\%$ for the reading. The flow meter for the liquid phase mass flow rate is a Coriolis flow sensor produced by Bronkhorst, of type M15-AGD-22-0-S. It is able to measure mass flow rates from $0.72 – 36 \text{ kgh}^{-1}$ with an accuracy of $\pm 0.2\%$ for liquid flows of $91\%NH_3 + 9\%H_2O$ or for compositions that lie close. Unfortunately, the sensor was not working after installation and during the experiments. Through testing of the sensor in collaboration with the manufacturer, it was established that the installed sensor was not functioning correctly. A method to estimate the mass flow rate at the same location is described further ahead in this chapter.

**Density sensors**
The density sensor (Di-1) in the system is located inside the same device as the mass flow sensor for the liquid phase flow of the working fluid (Fi-202). The accuracy is $\pm 0.53 \text{ kgm}^{-3}$, which was based on a calibration using water with a density of $998.2 \text{ kgm}^{-3}$. Since the mass flow sensor in the device was not functional during the experiment, the density sensor was assumed to be functioning incorrectly as well or at least expected to give unreliable measurement results. The sensor did however measure values that appeared reasonable during testing, and the sensor values were carefully used to provide estimates of other parameters as discussed further ahead in this section.

**Level sensors**
The level sensor in the working fluid buffer tank (Li-2) is of type T-FM121 and produced by Liquicap. It will measure the level for conductive liquids between $-40^{\circ}C$ and $100^{\circ}C$, for a maximum kinematic viscosity of $0.002 \text{ m}^2\text{s}^{-1}$. The accuracy is $1\%$. 
5.1. EXPERIMENTAL SETUP

OVERVIEW
An overview of the relevant measurement devices in the system for the calculations related to this study is given in Table 5.3, related to the P&ID presented in Figure 5.2 and the sensor descriptions that were provided. Here, $T_{wf,sep,L}$, $P_{wf,sep,L}$ and $\rho_{wf,sep,L}$ refer respectively to the temperature, pressure and density of the liquid phase outlet of the separator. This part of the system is not included in the P&ID, but the measured values of these parameters are used for calculation of other parameters. This is discussed further ahead in this chapter.
Table 5.3: Overview of the relevant measurements devices in the system. The sensor name, type, manufacturer, measured parameter, measurement range and accuracy are presented.

<table>
<thead>
<tr>
<th>Sensor name</th>
<th>Parameter</th>
<th>Sensor type</th>
<th>Units</th>
<th>Manufacturer and specific sensor type</th>
<th>Measurement range</th>
<th>Accuracy</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ti-2</td>
<td>(T_{wf, in})</td>
<td>Temperature</td>
<td>°C</td>
<td>TC Direct PT-100 class B</td>
<td>0 – 100°C</td>
<td>0.3 – 0.8°C</td>
</tr>
<tr>
<td>Ti-3</td>
<td>(T_{cw, in})</td>
<td>Temperature</td>
<td>°C</td>
<td>TC Direct PT-100 class B</td>
<td>0 – 100°C</td>
<td>0.3 – 0.8°C</td>
</tr>
<tr>
<td>Ti-8</td>
<td>(T_{cw, out})</td>
<td>Temperature</td>
<td>°C</td>
<td>TC Direct PT-100 class B</td>
<td>0 – 100°C</td>
<td>0.3 – 0.8°C</td>
</tr>
<tr>
<td>Ti-9</td>
<td>(T_{wf, sep, L})</td>
<td>Temperature</td>
<td>°C</td>
<td>TC Direct PT-100 class B</td>
<td>0 – 100°C</td>
<td>0.3 – 0.8°C</td>
</tr>
<tr>
<td>Ti-10</td>
<td>(T_{wf, in, L})</td>
<td>Temperature</td>
<td>°C</td>
<td>TC Direct PT-100 class B</td>
<td>0 – 100°C</td>
<td>0.3 – 0.8°C</td>
</tr>
<tr>
<td>Ti-13</td>
<td>(T_{wf, out})</td>
<td>Temperature</td>
<td>°C</td>
<td>TC Direct PT-100 class B</td>
<td>0 – 100°C</td>
<td>0.3 – 0.8°C</td>
</tr>
<tr>
<td>Pi-2</td>
<td>(P_{wf, in, V})</td>
<td>Pressure</td>
<td>bar gauge</td>
<td>Kobold SEN-3272 internal diaphragm</td>
<td>0 – 10 bar gauge (100°C max.)</td>
<td>0.5%</td>
</tr>
<tr>
<td>Pi-6</td>
<td>(P_{wf, in, L})</td>
<td>Pressure</td>
<td>bar gauge</td>
<td>Kobold SEN-3272 internal diaphragm</td>
<td>0 – 10 bar gauge (100°C max.)</td>
<td>0.5%</td>
</tr>
<tr>
<td>Pi-7</td>
<td>(P_{wf, out})</td>
<td>Pressure</td>
<td>bar gauge</td>
<td>Kobold SEN-3272 internal diaphragm</td>
<td>0 – 10 bar gauge (100°C max.)</td>
<td>0.5%</td>
</tr>
<tr>
<td>Pi-9</td>
<td>(P_{wf, sep, L})</td>
<td>Pressure</td>
<td>bar gauge</td>
<td>Kobold SEN-3272 internal diaphragm</td>
<td>0 – 10 bar gauge (100°C max.)</td>
<td>0.5%</td>
</tr>
<tr>
<td>Pi-20</td>
<td>(P_{cw, in})</td>
<td>Pressure</td>
<td>bar gauge</td>
<td>Kobold SEN-3272 internal diaphragm</td>
<td>0 – 10 bar gauge (100°C max.)</td>
<td>0.5%</td>
</tr>
<tr>
<td>Pi-21</td>
<td>(P_{cw, out})</td>
<td>Pressure</td>
<td>bar gauge</td>
<td>Kobold SEN-3272 internal diaphragm</td>
<td>0 – 10 bar gauge (100°C max.)</td>
<td>0.5%</td>
</tr>
<tr>
<td>Fi-101</td>
<td>(V_{cw})</td>
<td>Volumetric flow rate</td>
<td>L h⁻¹</td>
<td>AE Sensors DF140 flow sensor, 3/4” size</td>
<td>30 – 3000 L h⁻¹</td>
<td>±1%</td>
</tr>
<tr>
<td>Fi-201</td>
<td>(m_{wf, V})</td>
<td>Mass flow rate</td>
<td>kgs⁻¹</td>
<td>Bronkhorst T14-AGD-44-E thermal bypass mass flow meter</td>
<td>0.0001 – 0.005 kgs⁻¹</td>
<td>1% full scale</td>
</tr>
<tr>
<td>Fi-202</td>
<td>(m_{wf, L})</td>
<td>Mass flow rate</td>
<td>kgh⁻¹</td>
<td>Bronkhorst M15-AGD-22-0-S Coriolis flow sensor</td>
<td>0.72 – 36 kgh⁻¹</td>
<td>±0.2%</td>
</tr>
<tr>
<td>Di-1²</td>
<td>(P_{wf, sep, L})</td>
<td>Density</td>
<td>kgm⁻³</td>
<td>Bronkhorst M15-AGD-22-0-S Coriolis flow sensor</td>
<td>not specified</td>
<td>±0.53 kgm⁻³</td>
</tr>
</tbody>
</table>

¹This sensor did not work during the performed experiments.
²It is uncertain whether this sensor was functioning correctly during the experiments.
³At \(T = 26.5°C\) and \(P = 9.5\) bar, 100% \(NH₃\).
⁴Liquid flows of 91% \(NH₃\) + 9% \(H₂O\).
⁵Additional ±1% reading error is specified.
⁶For \(\rho = 998.2 \text{ kgm}^{-3}\).
5.1. EXPERIMENTAL SETUP

OPERATING CONDITIONS
The pressure and temperature of the flows are measured by their respective meters, presented in Table 5.3. The temperature of the hot and cold water is adjusted by heating or cooling the basins where the water is pumped from. This in turn also affects the temperature of the working fluid in different locations, due to the transfer of heat within the heat exchangers. In this way, the temperature of the different streams is controlled within certain bounds. These bounds are set by the boiling and freezing temperatures of the fluids, the effectiveness of heat transfer in all the components, the mass flow rates of all flows and the heating and cooling power of the water basins. The pressure at different locations in the system results from the combination of the other operating conditions and system settings, such as valves being open or closed.

The inlet stream of the working fluid consists of two separate streams, P-006 and P-013, which are combined in the line before the inlet. The stream in P-006 is assumed to be a pure vapor, and P-013 is assumed to contain purely liquid, since the streams originate from respectively the top and bottom of the separator. For this reason, the subscript \( V \) is used for stream 1 and \( L \) for stream 2. The temperature and pressure in the combined line are calculated by assuming homogeneous mixing of both streams, and using the known temperature, pressure, mass flow rate and composition of both separate streams. As stressed before, this is a simplification of the actual situation in the absorber, where the working fluid liquid is sprayed on the heat exchanger plates. To allow this, first the specific heat of the liquid and vapor phase \( (c_{p,wf,L} \text{ and } c_{p,wf,V}) \) are calculated based on Equation A.42 and Equation A.43. The heat flows of the phases are then calculated through the following equations:

\[
Q_{wf,L} = \dot{m}_{wf,L} c_{p,wf,L} T_{wf,in,L}
\]

\[
Q_{wf,V} = \dot{m}_{wf,V} c_{p,wf,V} T_{wf,in,V}
\]

The total heat and mass flow \( (Q_{tot} \text{ and } \dot{m}_{tot}) \) of both phases combined are computed by simply adding the individual phase heat and mass flows:

\[
Q_{wf,tot} = Q_{wf,L} + Q_{wf,V}
\]

\[
\dot{m}_{wf,tot} = \dot{m}_{wf,L} + \dot{m}_{wf,V}
\]

The specific enthalpy of the total working flow \( (h_{wf,tot}) \) is obtained by dividing the heat flow by the mass flow:

\[
h_{wf,tot} = \frac{Q_{wf,tot}}{\dot{m}_{wf,tot}}
\]

The pressures of both streams are roughly equal, the numerical average of both pressures should give a good approximation of the pressure of the combined stream:

\[
P_{wf,in} = \frac{P_{wf,in,L} + P_{wf,in,V}}{2}
\]

The inlet temperature of the working fluid \( (T_{wf,in}) \) is calculated as function of the enthalpy and pressure of the stream, using FluidProp:
\[ T_{wf,in} = f(P_{wf,in}, h_{wf,tot}) \] (5.7)

The assumption on which this method is based should be validated, perfect mixing of the vapor and liquid phases in the line that feeds the stream to the absorber and the working fluid inlet section of the absorber. The model also allows the input of different vapor and liquid phase bulk temperatures, and this may appear to be a more realistic approach. The flow meter in the liquid phase stream can also measure the density \( \rho_{wf,L,sep} \) of the stream. The composition of the liquid phase stream is based on the measurements of the density, temperature \( T_{wf,L,sep} \) and pressure \( P_{wf,L,sep} \) in that stream, which together give the ammonia molar fraction \( \bar{x} \) through the assumed thermodynamic equilibrium. This is done by taking Equation A.40 and varying the composition until a match is found for the computed density. Since it was uncertain whether Di-1 worked correctly, this method is not the most reliable. However, as stated before, the sensor did measure probable values and stable values during test runs for longer time scales.

The overall composition of the working fluid is determined by monitoring the amounts fed of the pure ammonia and water when filling the system. The equation for the overall ammonia mass fraction \( w \) based on the measured mass of the ammonia and water \( m_1 \) and \( m_2 \) fed to the system is given as:

\[ w = \frac{m_1}{m_1 + m_2} \] (5.8)

When the system is in steady-state, it is assumed that the overall molar composition at the inlet of the condenser is equal to the overall molar composition in the whole system. This assumption is valid given that on this location the vapor and liquid phases are not separated into different streams, and assuming no leakage or hold-up of one specific phase or component at a certain location in the system. In the working fluid streams after the separator, the compositions of the vapor and liquid phases are different. The salinity of the cooling water is measured or determined by the amount of salts added to the water. In the case of the experimental setup, the value is assumed to be zero. Demineralized water is used instead of seawater, which has a negligible concentration of salts. The salinity value of the cooling water will affect fluid properties such as density and viscosity. Estimates of the fluid properties for salt water can be made using the method described by Sharqawy et al. (2011) [48], through equations A.66 to A.69.

The mass flow rate of the working fluid and cooling water are measured using the flow meters presented in Table 5.3. The cooling water flow meter measures the flow as a volumetric flow rate. In this case the density should be calculated based on the measured pressure and temperature at the same location, using Equation A.14. The mass flow rate is then calculated using Equation 3.63. The mass flow rates in the system are controlled by adjusting the speed of the pumps. The mass flow rate of the working fluid, hot water or cold water is bounded by a minimum and maximum flow rate, due to the minimum and maximum amount of power that the pump is able to supply.

**Working fluid liquid phase mass flow rate**

Another method had to be used to estimate the working fluid liquid phase mass flow rate, since the flow sensor Fi-202 was not functional during the performed experiments. This may be done in different ways, using mass or energy balances. Using only mass balances, the liquid phase mass flow rate is determined based on vapor phase mass flow rate and
composition, and on the liquid phase and total composition. Three balances are set up for the total flow that is split into the liquid and vapor phase flow. These are the total mass balance and the mass balances of both species, of which two are independent:

\[ \dot{m} = \dot{m}_L + \dot{m}_V \]  
\[ \dot{m}_1 = \dot{m}_{1,L} + \dot{m}_{1,V} \]  
\[ \dot{m}_2 = \dot{m}_{2,L} + \dot{m}_{2,V} \]

The first equation can be expressed as the sum of the first two:

\[ \dot{m} = \dot{m}_L + \dot{m}_V = \dot{m}_{1,L} + \dot{m}_{2,L} + \dot{m}_{1,V} + \dot{m}_{2,V} = \dot{m}_1 + \dot{m}_2 \]

The total mass fraction of ammonia \((w)\), and the mass fractions of ammonia in the vapor and liquid phase \((x\) and \(y\)) are expressed respectively as:

\[ w = \frac{\dot{m}_1}{\dot{m}_1 + \dot{m}_2} \]  
\[ x = \frac{\dot{m}_{1,L}}{\dot{m}_{1,L} + \dot{m}_{2,L}} \]  
\[ y = \frac{\dot{m}_{1,V}}{\dot{m}_{1,V} + \dot{m}_{2,V}} \]

All the different mass flow rates and compositions corresponding to a certain phase or certain species are represented in Figure 5.3. The total flow in the system can be split up into two phases, and each of the phases contains two species, or only a single species in certain specific cases.

![Figure 5.3: Mass flow rates and compositions corresponding to a two-phase, two-component stream.](image)

The species mass flow rates of the liquid phase \((\dot{m}_{1,L} \text{ and } \dot{m}_{2,L})\) are then given by combining and rearranging the previous relations. By using \(y = 1\) through assumption of pure ammonia flow in the vapor phase it follows that \(\dot{m}_V = \dot{m}_{1,V}\), and the equations are:

\[ \dot{m}_{2,L} = \frac{\dot{m}_{1,V}(1 - w)(1 - x)}{w - x} \]  
\[ \dot{m}_{1,L} = \frac{\dot{m}_{2,L}x}{1 - x} \]

Using Equation 5.12 the total liquid phase mass flow rate \((\dot{m}_{w,f,L})\) is then found. The method to calculate the working fluid liquid phase flow rate using energy balances is much
more extensive. It may give a more accurate prediction, but it is also subject to the uncertainty of many different parameters in the system. Therefore, this method is not applied and not described here.

5.2. EXPERIMENTAL PLAN

The experimental work to be performed is based on an experimental plan consisting of multiple experimental sets, each set consisting of the different input parameters. These sets are constructed in a way that there is always an experimental set that differs only in one parameter from another experimental set. There are limits to the possible range of experimental sets, that can be conducted in practice. This is caused by the possible ranges of parameters themselves, or by the limits to combinations of different parameters that allow a steady-state operation of the system. The second cause may lead to that an experiment can only be performed for a set of parameters that is slightly different from the planned set. For the first cause, the ranges of values of parameters that are possible can be either dependent on or independent of the other parameters.

The possible range of parameters is naturally restricted by physical limits. Compositions do not exceed unity or fall below zero. Besides, as discussed before, the power limits of the pump and heating and cooling equipment respectively limit the possible range of flow rates and temperatures. The pressure and temperature are restricted to certain levels at locations in the system where exceeding this level might break or damage the equipment, instrumentation or piping. Finally, the values of parameters should be within the range of practical significance. A cold water source temperature of 4 – 7 °C and warm water source temperature of 24 – 29 °C correspond to conditions that are encountered in an actual location where OTEC is applied. The mass flow rate and temperature are selected to be close to the optimal conditions identified by Kirkenier (2014) [3].

Different experiments are performed using the setup with 100% ammonia in the work of Gudjonsdottir (2015) [98]. For validation of the detailed model, where phase change is taken into account as well as heat and mass transfer in both of the phases, experiments with the ammonia-water mixture should be performed. The optimal mixture composition can be based on a full system analysis of the OTEC cycle, but this does not necessarily imply that this composition is also optimal for each specific component in the system, such as the absorber. For this reason, the mixture composition should be between values of 90% and 100% ammonia, since the optimum is expected to lie somewhere in this range. A composition of 96% is taken as a starting point, and experimental results for this composition are compared with using pure (100%) ammonia. The mass flow rates of the working fluid are set to values 2.5 times higher than in the set by Gudjonsdottir (2015) [98], since the numerical simulations predict more effective heat transfer in this range within the heat exchanger. More about this is explained in Chapter 4, on the numerical results. The final experimental plan, containing six experimental sets is presented in Table 5.4. Here, also the temperature and mass flow rate of the hot water stream are specified. All components need to be running when operating the system, and these values are required as set points for the heaters and pumps on the hot water side.

Due to many technical difficulties in the preparations for the experimental work, the actual experiments were delayed by several months. The caused delay in combination with the planning of the project, resulted in that only a single experimental set could be performed using the setup. The results are discussed in the next section.
5.3. EXPERIMENTAL RESULTS

The performed experiment was conducted under steady-state conditions, which is essential for comparison with the numerical results. The sensors described earlier were used to perform measurements, and were coupled to LabVIEW to monitor the measured values over time and write the values to a file. Two major technical difficulties were encountered during the experiment. The first difficulty is related to the broken flow meter (Fi-202) and the density meter in the same device (Di-1), which may not have been functioning correctly. The alternative methods to calculating the mass flow rate and composition in this flow are discussed in subsection 5.1.3.

The other technical difficulty was related to regulation of the cooling water temperature. The cooling water is pumped from a basin (T-3), which is kept at constant temperature using two heat pumps. Both heat pumps can only be turned fully on or off, there is no regulation for the delivered power by the heat pumps. The power of the smaller heat pump alone is for typical operating conditions insufficient to cool the water to the desired temperature and keep it at the temperature. On the other hand, the power of the large heat pump is larger than required to cool the water to desired temperature and keep it there. This results in the requirement of a control loop that turns both heat pumps on and off at the correct times to keep the cooling water temperature close to the desired value. The frequency in which the pumps can be turned on or off is limited, and besides that there is a time delay in switching the pumps on or off. Also, the heat pumps have to run for a minimum amount of time before they can again be switched off. This results in periodic behavior of the measured temperature of the cooling water over time, but the time-averaged value is constant over longer time scales and close to the set-point. The profiles of the cooling water temperatures ($T_{cw,in}$ and $T_{cw,out}$) are presented in Figure 5.4, along with the other temperature profiles. Time-averaged values are used instead of instantaneous values since these are expected to better reflect the operating conditions. Measured instantaneous values strongly depend on the exact time of the performed measurement. Also, due to the fact that the fluctuations in different measured parameters are out of phase, the set of operating conditions measured instantaneously are not expected to correspond to each other for the assumption of a steady-state of the system. The measurements of temperature and other parameters are discussed in the next subsections.

5.3.1. TEMPERATURE

The periodic profile of the temperature over time is also observed in other temperature profiles than the cooling water. The temperature profiles of all relevant temperature sensors

Table 5.4: The experimental plan, containing the values for each experimental set of all parameters that can be controlled.

<table>
<thead>
<tr>
<th>Test</th>
<th>$\dot{m}_{cw}$ [kgs$^{-1}$]</th>
<th>$\dot{m}_{hw}$ [kgs$^{-1}$]</th>
<th>$\dot{m}_{wf}$ [kgs$^{-1}$]</th>
<th>$T_{hw}$ [°C]</th>
<th>$T_{cw}$ [°C]</th>
<th>$\omega_{NH_3}$ [%]</th>
<th>$S$ [gkg$^{-1}$]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.079</td>
<td>0.204</td>
<td>0.0045</td>
<td>29</td>
<td>5</td>
<td>92-98</td>
<td>0</td>
</tr>
<tr>
<td>2</td>
<td>0.079</td>
<td>0.204</td>
<td>0.0045</td>
<td>27</td>
<td>5</td>
<td>92-98</td>
<td>0</td>
</tr>
<tr>
<td>3</td>
<td>0.079</td>
<td>0.204</td>
<td>0.0045</td>
<td>24</td>
<td>7</td>
<td>92-98</td>
<td>0</td>
</tr>
<tr>
<td>4</td>
<td>0.079</td>
<td>0.204</td>
<td>0.0050</td>
<td>27</td>
<td>5</td>
<td>92-98</td>
<td>0</td>
</tr>
<tr>
<td>5</td>
<td>0.079</td>
<td>0.079</td>
<td>0.0045</td>
<td>27</td>
<td>5</td>
<td>92-98</td>
<td>0</td>
</tr>
<tr>
<td>6</td>
<td>0.079</td>
<td>0.079</td>
<td>0.0045</td>
<td>27</td>
<td>5</td>
<td>92-98</td>
<td>0</td>
</tr>
</tbody>
</table>
for the performed experiment, with a total sample time of 8616 s or roughly 2.4 h, are pre-
sented in Figure 5.4. The temperature of the cooling water at the inlet of the heat exchanger
is measured using Ti-3, and the direct influence of the temperature fluctuations is clearly
visible there. The fluctuations are more or less in phase and of equal magnitude at the out-
let of the cooling water flow (Ti-8). Fluctuations are most irregular at the liquid outlet of the
separator (Ti-9), but they still appear to have the same characteristic time scale and fluctu-
ate around a constant temperature. The temperatures of the liquid and vapor working fluid
streams that are mixed before the inlet (Ti-10 and Ti-2) display the same fluctuating behav-
ior in terms of their time scale, but vary in magnitude. This is caused by differences in mass
flow or the heat capacity. The temperature of the working fluid outlet (Ti-13) displays more
irregular fluctuation behavior, with sharp downward peaks interchanged by more smooth
peaks upward. It is expected that the downward peaks are caused by the subcooling of the
liquid after full condensation of the ammonia-water mixture. There is no direct evidence
for this observation, but it is a plausible explanation considering the difference with the
other observed temperature profiles. As alternative to using Ti-13 as an indication of the
working fluid temperature at the heat exchanger outlet, Ti-4 could be used, which is lo-
cated after the buffer tank and has smaller fluctuations in the temperature profile. This
temperature profile is not included in the figure. The average measured temperature using
this sensor is slightly higher, by roughly 0.5°C. The cause for this higher temperature is not
known, but should be investigated. For now, the temperature from sensor Ti-13 is used as
the indication of the working fluid outlet temperature, despite the large fluctuations in the
measured profile. The average value and other relevant values corresponding to each tem-
perature profile are summarized in Table 5.5, along with all other measured parameters or
parameters calculated using the measured data.

5.3.2. Pressure
The relevant pressure profiles of six different sensors are presented in Figure 5.5. Certain
profiles overlap or lie very close. Both the inlet (Pi-20) and outlet (Pi-21) gauge pressure
of the cooling water are close to zero, meaning that both absolute pressures are around
atmospheric pressure, as expected. The profiles at the working fluid inlet of the liquid (Pi-6)
and vapor (Pi-2) phase and at the working fluid outlet (Pi-7) are also similar. This confirms
that the pressure drop is small on the working fluid side of the heat exchanger. The pressure
drops are discussed in more detail in section 5.5. The pressures also fluctuate with the same
characteristic timescale as the temperature. The pressure at the outlet of the separator and
recuperator, before the expansion valve (Pi-9) is higher than the other pressures and shows
more irregular fluctuations.

5.3.3. Flow rates
The volumetric and mass flow rate measured respectively of the cooling water flow and the
working fluid flow are presented in Figure 5.6 and Figure 5.7. The cooling water flow is
reasonably stable, even though two small steps appear in the displayed profile, where the
flow rate quickly increases over a short time period. The fluctuations in the working fluid
vapor flow are relatively large. This indicates that the liquid and vapor flow rates leaving
the separator over time vary strongly. The cause for this is not known and was not investi-
gated, since the available time for the experiments did not allow this. The variations may
completely or for a part be a result of the fluctuations of the temperature in different parts
of the system. In case the same behavior is also observed in future experiments, the cause should be investigated and an attempt should be made to minimize the variations. The mass flow rate of the liquid leaving the separator cannot be measured, only estimated using the method previously described. It is assumed that the mass flow rate of both phases combined, which is fed to the absorber, shows less fluctuations. The only way to know if this assumption is true is to measure the mass flow rate of the liquid phase or the full working fluid. For all calculations in the model or for the experimental work steady-state of the system is assumed.

5.3.4. DENSITY AND LIQUID LEVEL
The density and liquid level plots are presented in Figure 5.8 and Figure 5.9. The measured density value of the working fluid over time using sensor Di-1 shows on average little fluctuations, but many sharp upward or downward peaks are observed. These may be caused by the presence of vapor in the fluid, which the sensor cannot handle, resulting in values that are off. As stressed before, it was not known if the density meter was functioning correctly during the experiments, so some carefulness is required when discussing results related to the density measurements. The liquid level of the working fluid buffer tank after the absorber outlet is measured using level sensor Li-2, and the observed profile shows slightly irregular repeating fluctuations. This may be caused by the fluctuations in the pressure and temperature of the working fluid temperature at the inlet and outlet of the condenser, which will result in a different distribution of the phases inside the buffer tank. Also, a varying mass flow rate of the working fluid may cause these fluctuations. It is not possible to conclude if this was the case, since the only flow meter measuring the flow rate of the working fluid flow measures only a single phase of the flow.

5.3.5. OTHER PARAMETERS
In Table 5.5 certain parameters are presented which are used to compare the results of the numerical model with the measured values of the experiments. This is discussed in section 5.5. These are the measured pressure drops ($\Delta P$) and temperature changes ($\Delta T$) of both streams between the inlet and outlet:

$$\Delta P_{cw} = P_{cw,in} - P_{cw,out}$$ (5.18)
$$\Delta P_{wf} = P_{wf,in} - P_{wf,out}$$ (5.19)
$$\Delta T_{cw} = T_{cw,out} - T_{cw,in}$$ (5.20)
$$\Delta T_{wf} = T_{wf,out} - T_{wf,in}$$ (5.21)

Also, the value of the total heat transferred across the wall ($Q_{w,\text{tot}}$) is calculated based on the cooling water side using the following equation:

$$Q_{w,\text{tot}} = Q_{cw,out} - Q_{cw,in} = \dot{m}_{cw,out}c_{p,cw,out}T_{cw,out} - \dot{m}_{cw,in}c_{p,cw,in}T_{cw,in}$$
$$= \dot{m}_{cw}(c_{p,cw,out}T_{cw,out} - c_{p,cw,in}T_{cw,in})$$ (5.22)
Here, it is used that the mass flow on the cooling water side is constant and equal to $\dot{m}_{cw}$. The specific heat of the cooling water ($c_{p,cw}$) is calculated at the channel inlet and outlet using Equation A.16. A check can be done of the total heat transferred across the wall using the estimated inlet and outlet heat flows of the working fluid. However, in this case the vapor quality should be known at the outlet. As discussed before, the temperature of the working fluid at the outlet ($T_{WF,out}$) shows large fluctuations with large downward peaks, as is observed in Figure 5.4. The peaks are thought to be caused by subcooling of the liquid, and the remainder of the measurement values for a liquid-vapor mixture. Also, good estimations are needed of the mass flow rate and the temperature of the liquid and vapor phase at the inlet. As discussed before, the estimations made for these values are very rough and the uncertainties are large. Thus, it is better to base the exchanged heat on the cooling water side. To calculate the total heat transfer coefficient ($U$), the LMTD method is used and the logarithmic mean temperature difference over the entire heat exchanger is taken. Equations 3.173, 3.174 and 3.178 are used to calculate $\Delta T_{top}$, $\Delta T_{bot}$ and $LMTD_{tot}$. Next, Equation A.92 is used to calculate $A_{p,tot}$ and Equation 3.184 to calculate $U_{tot}$. All average calculated values and their uncertainties are presented in Table 5.5.

![Figure 5.4: Temperature profiles of different relevant temperature sensors in the system, as function of the time since the start-up of the system.](image)

### 5.4. Uncertainty Analysis

In the previous section, the fluctuations in the measured values are discussed. These fluctuations are for the largest part caused by actual physical changes of parameters, since it is known that certain input parameters of the system strongly fluctuate. For a proper uncertainty analysis, instrumentation and repeatability uncertainty needs to be taken into account. The measured fluctuations are in fact related to the repeatability uncertainty, since a single measurement value is based on a large set of measurements with an interval of 1 second. In this case, variations in the outcome of the measurements are for the largest part explained by the known fluctuation of the cooling water temperature, due to the difficulty with regulation of the heat pumps discussed earlier.

Of all the measured parameters, certain values related to the distribution of measured
Figure 5.5: Pressure profiles of different relevant pressure sensors in the system, as function of the time since the start-up of the system.

Figure 5.6: Volumetric flow rate profile of flow sensor Fi-101, as function of the time since the start-up of the system.

Figure 5.7: Mass flow rate profile of flow sensor Fi-201, as function of the time since the start-up of the system.

Figure 5.8: Density profile of density sensor Di-1, as function of the time since the start-up of the system.

Figure 5.9: Liquid level profile of liquid level sensor Li-2, as function of the time since the start-up of the system.
values are presented in Table 5.5. The measured value in the dataset of a parameter for
measurement $i$ is referred to as $t_i$. If a total number of $N$ values are measured over the time
of the experiment for a parameter, the mean value of the parameter ($\bar{t}$) is given as:

$$\bar{t} = \frac{1}{N} \sum_{i=1}^{N} t_i$$  \hspace{1cm} (5.23)

The standard deviation of the parameter ($s$) is given as:

$$s = \sqrt{\frac{1}{N-1} \sum_{i=1}^{N} (t_i - \bar{t})^2}$$  \hspace{1cm} (5.24)

The minimum and maximum of the measured parameters are also computed and included in Table 5.5. The uncertainty in the measurement data, or repeatability uncertainty ($\tau_{rep}$), is given within 99% certainty by:

$$\tau_{rep} = 3s$$  \hspace{1cm} (5.25)

The instrumentation uncertainty values ($\tau_{instr}$) are given in the last column of Table 5.3. In case the relative uncertainty ($\tau_{* instr}$) is given, the absolute value is obtained by multiplying the relative uncertainty with the mean measured value:

$$\tau_{instr} = \tau_{* instr} \bar{t}$$  \hspace{1cm} (5.26)

The total absolute uncertainty ($\tau_{tot}$) and relative uncertainty ($\tau_{* tot}$) in each measured parameter value are given by:

$$\tau_{tot} = \sqrt{\tau_{instr}^2 + \tau_{rep}^2}$$  \hspace{1cm} (5.27)

$$\tau_{* tot} = \frac{\tau_{tot}}{\bar{t}}$$  \hspace{1cm} (5.28)

Besides parameters of which the values are measured directly, parameters are presented in Table 5.5 of which the values are based on calculations using the measured values. The used methods and equations are discussed in section 5.1 and section 5.3. Uncertainties in the values of the directly measured parameters will propagate into the uncertainties of the other parameters. This occurs through the rules of error propagation or propagation of uncertainty. The following analysis is performed using similar methods as discussed in the work of Gudjonsdottir (2015) [98] and Yuan et al. (2013) [101]. An arbitrary parameter $t_0$ is taken, which is a function of several other parameters:

$$t_0 = f(t_1, t_2 \ldots)$$  \hspace{1cm} (5.29)

The general equation for the error propagation is then given as:

$$\tau_{t_0} = \sqrt{\left(\frac{\partial t_0}{\partial t_1} \tau_{t_1}\right)^2 + \left(\frac{\partial t_0}{\partial t_2} \tau_{t_2}\right)^2 + \ldots}$$  \hspace{1cm} (5.30)

In certain cases the relation between the parameters to be determined and the other parameters is known exactly. For example, the uncertainty in $Q_{w f, L}$ discussed in Equation 5.1, is given as:
\[ \tau_Q = \sqrt{\left( \frac{\partial Q}{\partial \dot{m}} \tau_{\dot{m}} \right)^2 + \left( \frac{\partial Q}{\partial c_p} \tau_{c_p} \right)^2 + \left( \frac{\partial Q}{\partial T} \tau_T \right)^2} \]  

(5.31)

Here, \( Q, \dot{m}, c_p \) and \( T \) are used respectively instead of \( Q_{wf,L}, \dot{m}_{wf,L}, c_{p, wf,L} \) and \( T_{wf, in,L} \).

The relative error \( (\tau_Q^*) \) is given as:

\[ \tau_Q^* = \sqrt{\tau_{m}^* + \tau_{c_p}^* + \tau_{T}^*} \]  

(5.32)

For certain fluid properties, FluidProp is used to determine the value. The relation is only know in the form of input and output parameters, as for example when calculating the working fluid inlet liquid phase specific heat \( (c_{p, wf,L, in}) \). In the next equation, \( P \) is used for \( P_{wf, in, 2} \):

\[ c_p = f(x, P, T) \]  

(5.33)

The equation for the absolute error can be written in the same form as Equation 5.30. The relative error in \( c_p \) \( (\tau_{c_p}^*) \) is given as:

\[ \tau_{c_p}^* = \sqrt{(c_p(x + \tau_x, P, T) - c_p(x, T, P))^2 + (c_p(x, P + \tau_P, T) - c_p(x, T, P))^2 + (c_p(x, P, T + \tau_T) - c_p(x, T, P))^2} \]  

(5.34)

Following the methods described above, average values and uncertainty parameters are presented for relevant quantities in Table 5.5. Certain uncertainty parameters are known or calculated for the measured parameters \( (\tau_{rep}, \tau_{instr} \) and \( \tau_{instr}^* \)), but are not presented in the table due to space limitations.

### 5.5. Preliminary experimental validation of the model

The results of the model are compared to the experimental set discussed in section 5.3, of which the relevant values are presented in Table 5.5. An uncertainty analysis of the experimental results is presented in section 5.4. The uncertainty in the experimental results is high compared to the measured values. This is mostly a consequence of the large fluctuations of certain measured parameters, due to the problem with the cooling water heat pumps discussed in section 5.3. This makes it difficult to perform a proper validation of the model using the experimental results. For future studies, it is recommended that experiments with less unnecessary fluctuations in the input parameters are performed to better compare the experiments with numerical results. This should be possible, since the cause of the fluctuations is known, or at least the cause of the largest fluctuations. A careful comparison is thus made here. The model was run for the input settings of the base case, with input values that correspond with the experimental measurements, for a total of 25 iterations. Results of both the model and the experiments are summarized together in Table 5.6. The most relevant model results are presented in figures 5.10 to 5.16. As observed in the figures, the model predicts full condensation and thus subcooling of the working fluid liquid from close to 0.18 m into the channel for the average measured conditions. As discussed before, the working fluid is either expected to be a subcooled liquid or a two-phase vapor-liquid mixture at the channel outlet, depending on the exact time of the performed
Table 5.5: Overview of the relevant measured and calculated parameters in the system, used either as input for the model or to calculate other parameters. Uncertainty values are included where relevant, where ‘St. dev.’ stands for standard deviation.

<table>
<thead>
<tr>
<th>General</th>
<th>Sensor name</th>
<th>Parameter</th>
<th>Model input</th>
<th>Units</th>
<th>Values</th>
<th>Mean (±)</th>
<th>Maximum</th>
<th>Minimum</th>
<th>St. dev. (±)</th>
<th>r</th>
<th>r + t</th>
<th>r - t</th>
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<tbody>
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<td>Ti-8</td>
<td>Tref,int</td>
<td>Yes</td>
<td>°C</td>
<td></td>
<td>10.52</td>
<td>11.18</td>
<td>9.90</td>
<td>0.33</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
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<td>°C</td>
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<td>4.02</td>
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<td></td>
<td></td>
<td></td>
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<tr>
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<td>°C</td>
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<td></td>
</tr>
<tr>
<td>Pt-7</td>
<td>Pef,ref</td>
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<td>bar gauge</td>
<td></td>
<td>5.372</td>
<td>5.588</td>
<td>5.201</td>
<td>0.074</td>
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<td></td>
<td></td>
<td></td>
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<td>bar gauge</td>
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<td>K</td>
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<tr>
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<td>bar</td>
<td></td>
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<td>6.508</td>
<td>6.201</td>
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<td>0.035</td>
<td>6.596</td>
<td>6.147</td>
</tr>
<tr>
<td>Pt-21</td>
<td>Pef,ref</td>
<td>No</td>
<td>bar</td>
<td></td>
<td>0.973</td>
<td>0.981</td>
<td>0.965</td>
<td>0.002</td>
<td>0.008</td>
<td>0.008</td>
<td>0.981</td>
<td>0.965</td>
</tr>
<tr>
<td>Pt-20</td>
<td>Pef,in</td>
<td>No</td>
<td>bar</td>
<td></td>
<td>0.032</td>
<td>0.055</td>
<td>0.021</td>
<td>0.002</td>
<td>0.008</td>
<td>0.008</td>
<td>1.040</td>
<td>1.024</td>
</tr>
<tr>
<td>Ti-9</td>
<td>Tref,in,V</td>
<td>No</td>
<td>°C</td>
<td></td>
<td>15.84</td>
<td>19.18</td>
<td>12.44</td>
<td>1.24</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Ti-2</td>
<td>Tref,in,V</td>
<td>No</td>
<td>°C</td>
<td></td>
<td>16.07</td>
<td>16.49</td>
<td>15.72</td>
<td>0.15</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Ti-10</td>
<td>Tref,in,L2</td>
<td>No</td>
<td>°C</td>
<td></td>
<td>12.82</td>
<td>13.56</td>
<td>11.99</td>
<td>0.35</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Pt-9</td>
<td>Pef,in,L2</td>
<td>No</td>
<td>bar gauge</td>
<td></td>
<td>8.039</td>
<td>8.263</td>
<td>7.658</td>
<td>0.083</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Pt-6</td>
<td>Pef,in,L1</td>
<td>No</td>
<td>bar gauge</td>
<td></td>
<td>5.379</td>
<td>5.518</td>
<td>5.211</td>
<td>0.074</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Pt-2</td>
<td>Pef,in,Y</td>
<td>No</td>
<td>bar gauge</td>
<td></td>
<td>5.396</td>
<td>5.539</td>
<td>5.222</td>
<td>0.074</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Fl-201</td>
<td>mref,Y</td>
<td>No</td>
<td>kgm⁻¹</td>
<td></td>
<td>0.158</td>
<td>0.337</td>
<td>0.087</td>
<td>0.035</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Di-01</td>
<td>pef,Y</td>
<td>No</td>
<td>kg⁻¹</td>
<td></td>
<td>450.0</td>
<td>962.1</td>
<td>354.8</td>
<td>16.3</td>
<td>48.8</td>
<td>0.075</td>
<td>698.9</td>
<td>601.2</td>
</tr>
<tr>
<td>Ti-9</td>
<td>Tref,in,Y</td>
<td>No</td>
<td>K</td>
<td></td>
<td>288.99</td>
<td>292.34</td>
<td>285.95</td>
<td>1.24</td>
<td>3.72</td>
<td>0.013</td>
<td>292.72</td>
<td>285.27</td>
</tr>
<tr>
<td>Ti-2</td>
<td>Tref,in,Y</td>
<td>No</td>
<td>K</td>
<td></td>
<td>289.22</td>
<td>289.64</td>
<td>288.87</td>
<td>0.53</td>
<td>0.002</td>
<td>0.002</td>
<td>289.75</td>
<td>288.69</td>
</tr>
<tr>
<td>Ti-10</td>
<td>Tref,in,L2</td>
<td>No</td>
<td>K</td>
<td></td>
<td>285.97</td>
<td>286.71</td>
<td>285.14</td>
<td>0.33</td>
<td>1.10</td>
<td>0.004</td>
<td>287.07</td>
<td>284.88</td>
</tr>
<tr>
<td>Pt-9</td>
<td>Pef,in,L2</td>
<td>No</td>
<td>bar</td>
<td></td>
<td>9.033</td>
<td>9.263</td>
<td>8.656</td>
<td>0.083</td>
<td>0.254</td>
<td>0.028</td>
<td>9.283</td>
<td>8.784</td>
</tr>
<tr>
<td>Pt-6</td>
<td>Pef,in,L1</td>
<td>No</td>
<td>bar</td>
<td></td>
<td>6.379</td>
<td>6.518</td>
<td>6.211</td>
<td>0.074</td>
<td>0.225</td>
<td>0.035</td>
<td>6.604</td>
<td>6.154</td>
</tr>
<tr>
<td>Pt-2</td>
<td>Pef,in,Y</td>
<td>No</td>
<td>bar</td>
<td></td>
<td>6.396</td>
<td>6.539</td>
<td>6.222</td>
<td>0.074</td>
<td>0.223</td>
<td>0.035</td>
<td>6.619</td>
<td>6.173</td>
</tr>
<tr>
<td>Fl-201</td>
<td>mref,Y</td>
<td>No</td>
<td>kg⁻¹</td>
<td></td>
<td>0.00263</td>
<td>0.09562</td>
<td>0.08143</td>
<td>0.00039</td>
<td>0.00177</td>
<td>0.6733</td>
<td>0.00441</td>
<td>0.000866</td>
</tr>
</tbody>
</table>

1 Ti-4 could be used to measure the temperature, in the text it is motivated why Ti-13 is used.
2 The exact uncertainty in the weight measured is not known, so a measurement uncertainty of 10% is assumed.
3 The maximum and minimum values cannot be determined using FluidProp due to the large uncertainties in other parameters. The lowest and highest values are taken within the uncertainty limits of $T_{ref,in,Y}$ and $T_{ref,in,L1}$ since these are the temperatures of the two streams that are combined into this stream.
4 The values are based on the assumption that $\gamma_{NH_3} = 1$.
5 The calculated value is below zero.
6 The calculated value is larger than unity.
measurement. This is supported by the experimental measurements and specifically the measured temperature profile at the outlet of the working fluid channel \(T_{wf,\text{out}}\) presented in Figure 5.4.

**Overall Heat Transfer Coefficient**

The overall heat transfer coefficient \((U)\) is predicted by the model both through the LMTD method and through the average of the values in all control volumes. The overall heat transfer coefficient profile along the length of the heat exchanger channel predicted by the model is presented in Figure 5.10. The difference between the computed values is large, and the same is concluded for the difference with the value calculated using the experiments. It appears that the overall heat transfer coefficient is overpredicted by a large amount using the model. However, the calculated value of 407 Wm\(^{-2}\)K\(^{-1}\) from the experiment is smaller than expected for the heat exchanger. In the work of Gudjonsdottir (2015), the calculated value of \(U\) using different methods is between 869 and 1910 Wm\(^{-2}\)K\(^{-1}\). Even though the experiments in this work were done for pure ammonia, under slightly different operating conditions, the factor 2 – 5 difference in the computed overall heat transfer coefficient is large.

**Heat Exchanged**

When comparing the total heat transferred across the wall \((Q_{w,\text{tot}})\) between the experimental work and the model, the difference is 26%, which is reasonable for a first guess of the developed model. The question is whether the reasonable prediction by the model is due to a good agreement with the actual processes in the heat exchanger, or also partly due to the fact that differences in parameters that are combined to calculate the exchanged heat cancel out. When calculating the total exchanged heat based on the heat flow across the wall for each control volume, the model value is even slightly larger and equal to roughly 3.2 kW. This is also observed in Figure 5.11, by taking into account that there are 8 working fluid channels. As stated at the end of section 5.3, due to the large uncertainties in certain inlet and outlet parameters of the working fluid stream, it is difficult to make a good estimate of the exchanged heat in the heat exchanger based on the working fluid side for the experiment. For the model, the absorbed mass is calculated to be roughly 2.64 gs\(^{-1}\). This is also observed from Figure 5.12, where the working fluid mass flow rates in a control volume are presented, and again taking into account that there are 8 working fluid channels. In Figure 5.13 the bulk specific enthalpy of vaporization of the mixture is presented, of which the value is approximately \(1.25 \cdot 10^6\) Jkg\(^{-1}\). The heat exchanged due to absorption from the vapor phase is thus roughly 3.3 kW, which is practically all the heat that is exchanged. The value is even higher than the total heat exchanged here, due to errors in rounding off the numbers. The subcooled section can be neglected in terms of heat transfer.

**Logarithmic Mean Temperature Difference**

There is also a large difference in the calculated logarithmic mean temperature difference \((LMTD)\) value, as observed from the presented values in Table 5.6. The temperature profiles of the working fluid and cooling water bulk phase streams predicted by the model are presented in Figure 5.14. Through the model, the outlet temperature of either stream is predicted to lie much closer to the inlet temperature of the other stream, which is also a possible consequence of the overprediction of the overall heat transfer coefficient. Also, because the model predicts full condensation the temperature values lie much closer near
the outlet of the working fluid channel, since the liquid there is subcooled. From the experimental data, it was observed that the working fluid changes from a subcooled liquid to a vapor-liquid at the outlet over time due to the fluctuations in temperature and pressure. In case the model does not predict full condensation, such as for the base case, the temperatures near the working fluid outlet lie much further apart.

**Pressure drop**

The cumulative pressure drop profiles in both streams predicted by the model are presented in Figure 5.15 and Figure 5.16. The model predicts a positive pressure drop for the cooling water flow. In the experiment, also a positive pressure drop is measured and the total values differ by roughly 47%. For the working fluid pressure drop, the prediction of the model is negative due to a higher contribution of the negative elevation pressure drop than the positive friction and deceleration contributions. The measured value differs largely from the value predicted by the model. At the point where full condensation occurs in the channel, the deceleration pressure drop stops to play a role since the velocity of the liquid phase flow remains constant from that point on. The elevation pressure drop sharply decreases due to the large density of the liquid phase compared to the vapor phase. The pressure drop profiles are observed in figures 5.15 and 5.16. The parameter profiles in the other figures are not discussed in more detail here, another case where full condensation occurs is described in subsection 4.3.1.

**Recommendations**

To properly investigate the correspondence between the model and experiments, more experimental work should be done first. It is important to keep the measured parameters as constant as possible over time, to ensure steady-state of the system and reduce the uncertainties in the measured values. Also, as a result of the model input parameters obtained for the experimental measurements, the model predicts full condensation inside the working fluid channel. It is recommended to perform experiments outside the range where this is predicted. The experimental plan was designed to achieve this, but the actual values measured during the experiment were different from the experimental set. From a first comparison, it seems probable that the heat transfer is overpredicted using the model. However, actual conclusions can only be drawn after a larger set of reliable experimental data is obtained.
5.5. Preliminary Experimental Validation of the Model

Table 5.6: Overview of the relevant parameters and their values predicted using the numerical model and based on the performed experiment, either measured directly or calculated. The differences are given relative to the experimental value.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Units</th>
<th>Prediction</th>
<th>Difference</th>
<th>Measurement uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\Delta P_{cw}$</td>
<td>bar</td>
<td>0.031</td>
<td>0.016</td>
<td>0.013 0.072 0.046</td>
</tr>
<tr>
<td>$\Delta P_{wf}$</td>
<td>bar</td>
<td>−0.001</td>
<td>0.016</td>
<td>0.001 0.016 0.015</td>
</tr>
<tr>
<td>$\Delta T_{cw}$</td>
<td>K</td>
<td>9.20</td>
<td>6.49</td>
<td>0.04 6.53 6.46</td>
</tr>
<tr>
<td>$\Delta T_{wf}$</td>
<td>K</td>
<td>−9.87</td>
<td>−4.19</td>
<td>0.06 −4.14 −4.25</td>
</tr>
<tr>
<td>$\Delta T_{top}$</td>
<td>K</td>
<td>0.68</td>
<td>3.28</td>
<td>0.03 3.41 3.35</td>
</tr>
<tr>
<td>$\Delta T_{bot}$</td>
<td>K</td>
<td>0.00</td>
<td>5.69</td>
<td>0.06 5.75 5.63</td>
</tr>
<tr>
<td>LMTD $D_{rot}$</td>
<td>K</td>
<td>0.13</td>
<td>10.21</td>
<td>0.15 10.36 10.07</td>
</tr>
<tr>
<td>$Q_{w,tot}$</td>
<td>W</td>
<td>3130</td>
<td>2489</td>
<td>81 2570 2408</td>
</tr>
<tr>
<td>$U_{tot}$ (LMTD)</td>
<td>Wm$^{-2}$K$^{-1}$</td>
<td>4.31 $\cdot 10^4$</td>
<td>407</td>
<td>4.27 $\cdot 10^4$ $-105%$</td>
</tr>
<tr>
<td>$U_{tot}$ (average)</td>
<td>Wm$^{-2}$K$^{-1}$</td>
<td>2.48 $\cdot 10^3$</td>
<td>407</td>
<td>2.07 $\cdot 10^3$ $-509%$</td>
</tr>
</tbody>
</table>

Figure 5.10: Heat transfer coefficients related to both streams based on both the working fluid and cooling water side, as function of the position inside the heat exchanger for the experimental case.
5. Experimental Results

Figure 5.11: Cumulative heat flow changes of the working fluid and cooling water channels (two control volumes), as function of the position inside the heat exchanger for the experimental case.

Figure 5.12: Working fluid total mass flow rate and mass flow rates of both phases at both the control volume inlet and outlet, as function of the position inside the heat exchanger, for the experimental case.

Figure 5.13: Specific enthalpies related to the working fluid flow, and the specific enthalpy of vaporization, as function of the position inside the heat exchanger for the experimental case.

Figure 5.14: Temperatures of the working fluid and cooling water in the bulk phase, at the wall or at the phase interface, as function of the position inside the heat exchanger for the experimental case.

Figure 5.15: Total cumulative pressure drop of the working fluid, and the cumulative contributions due to elevation, deceleration and friction, as function of the position inside the heat exchanger for the experimental case.

Figure 5.16: Total cumulative pressure drop of the cooling water, and the cumulative contributions due to elevation and friction, as function of the position inside the heat exchanger for the experimental case.
6

CONCLUSIONS AND RECOMMENDATIONS

6.1. CONCLUSIONS
In this section the most important conclusions from the performed work discussed in the previous chapters are summarized, and in the next section recommendations for future research are made. The main goal of this study was to develop and validate a flow pattern based model for the use of the ammonia-water mixture as working fluid in a vertical downflow plate heat exchanger functioning as absorber. Experimental data is obtained, but as stressed before, more experimental work should be performed for steady-state conditions with less fluctuations in operating conditions to properly validate the model. The results of the numerical model compared to the performed experiment, discussed in section 5.3, show large differences. This may for a large part be due to the fluctuations in the measured parameters over time and the resulting large uncertainties in values that are required as input for the model.

The results of the model, apart from the comparison with experimental work, are discussed extensively in section 4.1. The relevant mass and energy balances are satisfied for the model in case the simulation is run for a sufficient amount of iteration loops. The exchanged heat predicted by either stream converges to a constant value during the iteration loop. In terms of magnitude, the predicted amount of heat exchanged is larger than expected. However, more experiments performed under the same conditions as those used in the model and reliable measurements are required for a proper comparison.

The temperature profiles of the cooling water and working fluid liquid in the bulk and at the wall appear reasonable in terms of trend, magnitude of the differences between inlet and outlet and correspondence to the relative heat transfer resistances in different regions within the control volume. In case the working fluid fully condenses, the liquid phase becomes subcooled and the temperatures of the cooling water and working fluid streams lie very close towards the working fluid channel outlet. The profiles of the interface and vapor phase temperature show a different behavior from what is expected, but the precise cause for this was not investigated. The predictions of the pressure drop over the channel length agree well with what is expected in terms of order of magnitude, and with values measured in the experiment.
Considering the predicted values for the overall heat transfer coefficient and heat exchanged over the entire heat exchanger, it seems probable that the heat transfer is overestimated. This could be largely the consequence of the assumed film flow pattern throughout the section of the working fluid channel where the flow is two-phase. In terms of the flow regime and the effects of surface tension, the film flow assumption appears to be valid. When comparing the predicted overall heat transfer coefficient and heat exchanged using the film flow assumption to predictions using direct correlations for two-phase flow, the results correspond well with the correlations proposed in the work by Winkelmann (2010) [35] and Yan et al. (1999) [68]. Observations are discussed in section 4.2.

The heat transfer resistance in the liquid phase is much smaller than in the vapor phase for the working fluid. It appears that the heat transfer of the working fluid liquid phase is limited by mass transfer, but for the vapor phase the mass transfer does not appear to play a role on the heat transfer. This observation is supported by the case studied in section 4.3, where the vapor phase mass transfer coefficient is enlarged manually. Since the mass transfer practically only concerns transport of ammonia, and only the mass transfer in the liquid phase appears relevant, the mass transfer modelling can be significantly simplified. In the same section a case is discussed where the mixture composition is changed to 95% ammonia, as compared to 96% for the base case. It is observed that full condensation of the working fluid is predicted for this case. The relevant mass and energy balances are still satisfied, but discontinuities in certain parameter profiles arise due to different correlations used for two-phase film flow and the single phase film flow of the working fluid.

Even though the model may overpredict the heat transfer, the overall predictions of the heat transfer coefficient and exchanged heat are reasonable. Some other minor deficiencies are present in the model, such as the temperature profiles that appear unlikely and the jumps in certain parameter values for full condensation. Through further development of the model, it should be possible to predict the performance of the heat exchanger under varying operating conditions and the specified geometry. Also, predictions could be made for the performance of other working fluids if the model is expanded to include predictions of the necessary thermodynamic and transport properties. This provides an added value over previous approaches, where the predictions made using the implemented correlations into the model are limited to specific working fluids and operating conditions. This last point is discussed in more detail in the recommendations.

6.2. RECOMMENDATIONS
The added value of the model compared to lumped approaches should be investigated, using direct empirical heat transfer correlations for two-phase condensing flow or the absorption of a mixture. To achieve this, first a proper experimental validation of the model using the ammonia-water mixture should be performed. Correspondence of the detailed model to the lumped approaches should be investigated, as well as how different approaches compare with the experimental results. If the numerical and experimental results agree sufficiently, the model may be extended to work with other working fluids. By performing more experiments at different operating conditions and possibly for different heat exchanger configurations, it can be investigated within what ranges the model gives reasonable predictions. Where necessary, the model should be improved to give better predic-
6.2. RECOMMENDATIONS

tions for a wider range of conditions, through implementation of empirical or theoretical
correlations that are valid for the given conditions. Also, a proper assumption of the occur-
ring flow pattern under different conditions to which the correlations correspond should
lead to better predictions of the model. The range of applicability of models implement-
ing direct or lumped correlations is usually very limited. The investigation should include
finding promising mixtures and their optimal compositions. Given these findings, the best
possible heat exchanger configurations and operating conditions can be identified.

On the shorter term, the focus should be on performing reliable experiments and col-
lecting high quality data. To allow this, fluctuations in operating conditions over time
should be avoided where possible. Fluctuations in operating conditions conflict with the
steady-state assumption and introduces large measurement errors that make the compari-
son with numerical results more difficult. The cause of the fluctuations of the experiments
is known, and this cause can and should be eliminated from the experiments by modifying
the setup. Design of an experimental set with operating conditions that are possible for
steady-state operation of the system is crucial.

An important goal for future studies is the validation of the film flow pattern assump-
tion made in this work. In section 2.5, an extensive study was presented on the current work
related to flow pattern prediction for two-phase flow in plate heat exchangers. The main
conclusions and recommendations are listed in the same section. Experimental work is
required on the visualization of flow patterns, under varying operational conditions, for
different fluids and heat exchanger geometries. The focus may be on adiabatic, isothermal
flows in the first case, and may later be expanded to flows where phase change occurs. Bet-
ter predictions of the expected flow pattern of the working fluid under given conditions will
also lead to better predictions of the heat exchanger performance. Besides, it may lead to
the requirement of the implementation of more flow patterns and corresponding correla-
tions into the model. Also, this could improve the transition from the two-phase film flow
to the single phase liquid flow. This should lead to continuity of profiles of predicted values
of the heat transfer coefficient and friction factors, without large jumps.

For the modelling work, the focus in the near future should be on development of re-
liable, accurate and fast fluid property predictions. These may be based on relations spe-
cific for a working fluid combination, but also on more general relations, valid for different
fluid combinations. There might be trade-offs in for example speed and accuracy, as was
the case in this work for choosing between the method by Conde (2006) [36] and the pre-
dictions using FluidProp [33]. Results of different relations should be compared in terms of
their influence on the overall predictions of the heat exchanger performance, and the corre-
spondence with the experimental work. Besides relations for the working fluid properties,
suitable relations for the cooling water properties with a certain content of salts are also re-
quired for practical predictions. The relations proposed in Sharqawy et al. (2011) [48] could
be implemented for this purpose, and the sensitivity towards the salinity investigated. Also,
the effect of changing the plate material may be studied. To allow this, the model only re-
quires the plate thickness to be specified, and a value or relation for the thermal conduc-
tivity to be chosen.
The temperature profiles of the working vapor phase and interface are not as expected in terms of their trend compared to the other profiles. This may be a consequence of the method used to initially guess the interface temperature in each control volume of the model. The interface temperature in the current implementation is based on the liquid phase temperature, since basing the value on the interface conditions led to instability in the simulation results. The cause of this should be investigated, as well as the results of the model after changing the initial interface temperature estimation to a more suitable one. The influence on the predicted temperature profiles of the working fluid interface and vapor phase should be studied. Also, the inlet temperature of the working fluid liquid and vapor phase are now set to the same value, based on perfect mixing of both phases. In practice, both phases are separated before introduction to the heat exchanger, and the temperatures may be different. A separate model for the liquid that is sprayed onto the heat exchanger plates could be required to give an accurate prediction of the inlet temperatures. Alternatively, the initial liquid phase and vapor phase bulk temperatures may be set to the measured values of the individual streams, where the assumption is made that no heat exchange occurs between the phases during the spraying process. Likewise, the inlet composition may be based on the measured values instead of the assumed thermodynamic equilibrium at the inlet. To allow this, reliable experimental predictions are required.

The sensitivity analysis of the influence of mass transfer on the numerical model should be expanded. By further increasing the mass transfer coefficient of the vapor phase, it can be established that the heat transfer indeed is not affected by the mass transfer. Also, the fraction of ammonia in the mixture should be varied between 90 – 100% to investigate the optimum composition for the absorber and the dependence of this optimum on operating conditions. Predictions should again be validated using experiments. The cause for the observed instabilities at higher ammonia fractions should be investigated, and modifications may be implemented in the model to avoid these instabilities. Finally, the geometrical details of the heat exchanger may be varied to study the effects on the performance, and possibly investigate more suitable designs of the heat exchanger.
A.1. DIMENSIONLESS NUMBERS

General Capillary number:

\[ Ca = \frac{u \mu}{\sigma} \]  \hspace{1cm} (A.1)

General Eötvös number:

\[ Eo = \frac{d^2 g \Delta \rho}{\sigma} \]  \hspace{1cm} (A.2)

General Froude number:

\[ Fr = \frac{u^2}{gd} \]  \hspace{1cm} (A.3)

General Graetz number:

\[ Gz = \frac{d^2 u}{\alpha z} \]  \hspace{1cm} (A.4)

General Lewis number:

\[ Le = \frac{a}{D} = \frac{\lambda}{\rho c_p D} \]  \hspace{1cm} (A.5)

General Nusselt number:

\[ Nu = \frac{hd}{\lambda} \]  \hspace{1cm} (A.6)

General Prandtl number:

\[ Pr = \frac{\nu}{\alpha} = \frac{\mu c_p}{\lambda} \]  \hspace{1cm} (A.7)

General Reynolds number:

\[ Re = \frac{\rho u d}{\mu} \]  \hspace{1cm} (A.8)
General Schmidt number:

\[ Sc = \frac{v}{D} = \frac{\mu}{\rho D} \]  
(A.9)

General Sherwood number:

\[ Sh = \frac{kd}{D} \]  
(A.10)

General Weber number:

\[ We = \frac{\rho u^2 d}{\sigma} \]  
(A.11)

### A.2. FLUID PROPERTIES

#### A.2.1. PURE FLUIDS

Certain pure fluid properties are independent of the thermodynamics of the system, and have a constant value specific for the fluid species. These constant properties include the critical temperature \((T_{\text{crit}})\), critical pressure \((P_{\text{crit}})\), critical density \((\rho_{\text{crit}})\) and molecular weight \((M)\). The critical density may be calculated based on the critical temperature and pressure through an equation of state function:

\[ \rho_{\text{crit}} = f(T_{\text{crit}}, P_{\text{crit}}) \]  
(A.12)

Other pure fluid properties depend on the thermodynamics of the system, and can be specified as a function of for instance the pressure and temperature. For a pure fluid inside a system at constant and uniform pressure and temperature, it will either be a pure liquid or a pure vapor or gas. The vapor quality \((X)\) will thus either be zero, in the case of a pure liquid, or otherwise equal to unity for a pure vapor. In systems where pressure and temperature vary with time or position, the instantaneous local vapor quality is computed from the pressure and temperature:

\[ X = f(P, T) \]  
(A.13)

Other pure fluid properties in this work are the density \((\rho)\), dynamic viscosity \((\mu)\), specific heat capacity \((c_p)\) and thermal conductivity \((\lambda)\). For a pure fluid these all depend on \(P\) and \(T\) as stated before:

\[ \rho = f(P, T) \]  
(A.14)

\[ \mu = f(P, T) \]  
(A.15)

\[ c_p = f(P, T) \]  
(A.16)

\[ \lambda = f(P, T) \]  
(A.17)

For either pure fluids or mixtures, the thermal diffusivity \((\alpha)\) and kinematic viscosity \((\nu)\) of a phase are given as function of other fluid properties:
\[ a = \frac{\lambda}{\rho c_p} \]  \hfill (A.18)

\[ \nu = \frac{\mu}{\rho} \]  \hfill (A.19)

### A.2.2. Binary Mixtures

#### Composition

The overall mass fraction of species 1 \((w_1)\) in a binary mixture is calculated using molar fraction of species 1 \((\bar{w}_1)\) and the molecular weights of both species \((M_1 \text{ and } M_2)\):

\[ w = \frac{1}{1 + \frac{M_2}{M_1} \cdot \frac{1-\bar{w}}{\bar{w}}} \]  \hfill (A.20)

Similarly, the molar fraction of species 1 is calculated from the mass fraction of species 1:

\[ \bar{w} = \frac{1}{1 + \frac{M_1}{M_2} \cdot \frac{1-w}{w}} \]  \hfill (A.21)

These relations do not only hold for the overall mass and molar fractions \((w \text{ and } \bar{w})\) but also for the phase mass and molar fractions of the liquid phase \((x \text{ and } \bar{x})\) and vapor phase \((y \text{ and } \bar{y})\). The species fractions may not only differ between the liquid and vapor phases, but also between the bulk of the phases and the region near the vapor-liquid interface.

The interface liquid phase species molar fraction \((\bar{x}_i)\) and vapor phase species molar fraction \((\bar{y}_i)\) are based on the pressure \((P)\), interface temperature \((T_i)\) and total species molar fraction \((\bar{w})\):

\[ \bar{x}_i = f(P, T_i, \bar{w}) \]  \hfill (A.22)

\[ \bar{y}_i = f(P, T_i, \bar{w}) \]  \hfill (A.23)

The bulk liquid phase species molar fraction \((\bar{x}_b)\) and vapor phase species molar fraction \((\bar{y}_b)\) are based on the pressure \((P)\), phase bulk temperatures \((T_{L,b} \text{ and } T_{V,b})\) and total species molar fraction \((\bar{w})\):

\[ \bar{x}_b = f(P, T_{L,b}, \bar{w}) \]  \hfill (A.24)

\[ \bar{y}_b = f(P, T_{V,b}, \bar{w}) \]  \hfill (A.25)

The binary mixture vapor quality is computed using FluidProp, using overall species molar fraction \((\bar{w})\), pressure and temperature:

\[ X = f(\bar{w}_b, P, T) \]  \hfill (A.26)

A value of the temperature representative of the considered system or control needs to be chosen, since the temperature may vary between phases or locally within the phases. Alternatively, the vapor quality may be defined using the total \((w)\) and phase \((x \text{ and } y)\) mass fractions or the total \((h_{tot})\) and phase \((h_L \text{ and } h_V)\) specific enthalpy.
\[ X = \frac{w - x}{y - x} \]  
(A.27)

\[ X = \frac{h_{\text{tot}} - h_L}{h_{V} - h_L} \]  
(A.28)

**Enthalpies**

The liquid and vapor bulk phase specific enthalpy are based on the phase species molar fractions, bulk temperature and vapor quality set to either zero for the liquid phase or unity for the vapor phase.

\[ h_{L,b} = f(\bar{x}_b, T_{L,b}, X = 0) \]  
(A.29)

\[ h_{V,b} = f(\bar{y}_b, T_{V,b}, X = 1) \]  
(A.30)

The total mixture two-phase enthalpy is based on the individual phase specific enthalpy and on the relative mass flow rate of the phases. When this ratio \( \dot{m}_L / \dot{m}_V \) is unknown, the individual mass flow rates should be calculated first based on the total mass flow rate \( \dot{m} \) and the vapor quality. Relevant relations are found in Chapter 3. The total specific mixture two-phase enthalpy is computed as follows:

\[ h_{\text{tot}} = \frac{\dot{m}_V h_V + \dot{m}_L h_L}{\dot{m}_V + \dot{m}_L} = \frac{h_V + \frac{\dot{m}_L}{\dot{m}_V} h_L}{1 + \frac{\dot{m}_L}{\dot{m}_V}} \]  
(A.31)

The specific enthalpies of the vapor and liquid phases at the interface are calculated in a similar way as the bulk values, using the species molar fractions and temperatures at the interface:

\[ h_{L,i} = f(\bar{x}_i, T_i, X = 0) \]  
(A.32)

\[ h_{V,i} = f(\bar{y}_i, T_i, X = 1) \]  
(A.33)

The partial specific enthalpy \( (\tilde{h}) \) of a certain species in a certain phase specifies the enthalpy change of the mixture with changes in molar fraction of that species in that phase, for constant temperature and pressure. The liquid phase partial specific enthalpy of species 1 and 2 are calculated as function of the partial derivatives of the partial specific enthalpy of the phases with respect to the molar fraction of species 1:

\[ \tilde{h}_{L,1} = h_L + \bar{x}_2 \frac{\partial h_L}{\partial \bar{x}_1} \]  
(A.34)

\[ \tilde{h}_{L,2} = h_L - \bar{x}_1 \frac{\partial h_L}{\partial \bar{x}_1} \]  
(A.35)

A similar approach is used for the partial specific enthalpy of species 1 and 2 in the vapor phase:

\[ \tilde{h}_{V,1} = h_V + \bar{y}_2 \frac{\partial h_V}{\partial \bar{y}_1} \]  
(A.36)
\[ \tilde{h}_{V,2} = h_V - \tilde{y}_1 \frac{\partial h_V}{\partial \tilde{y}_1} \] (A.37)

The partial specific enthalpy change from the liquid to the vapor phase for species 1 and species 2 are then given as:

\[ \Delta \tilde{h}_1 = \tilde{h}_{V,1} - \tilde{h}_{L,1} \] (A.38)

\[ \Delta \tilde{h}_2 = \tilde{h}_{V,2} - \tilde{h}_{L,2} \] (A.39)

**OTHER PROPERTIES**

The liquid and vapor phase density can also be calculated using FluidProp, based on the phase species molar fraction, vapor quality corresponding to the phase and either the temperature (for the liquid phase) or the pressure (vapor phase).

\[ \rho_L = f(\tilde{x}_b, T_L, X = 0) \] (A.40)

\[ \rho_V = f(\tilde{y}_b, P, X = 1) \] (A.41)

The specific heat of the liquid and vapor phases in the bulk can also be calculated using FluidProp, using a similar approach as for the density.

\[ c_{p,L} = f(\tilde{x}_b, T_L, X = 0) \] (A.42)

\[ c_{p,V} = f(\tilde{y}_b, T_V, X = 1) \] (A.43)

The saturation temperature of the liquid phase at the interface is given for the interface species molar fraction in the liquid phase, the pressure and the vapor quality set to zero:

\[ T_{sat,L,i} = f(\tilde{x}_i, P, X = 0) \] (A.44)

For the vapor phase, the volumetric expansion coefficient can be approximated by that of an ideal gas:

\[ \beta_V = \frac{1}{T} \] (A.45)

Alternatively, the volumetric expansion coefficient may be calculated as a function of temperature and composition, using FluidProp.

\[ \beta_V = f(\tilde{y}_b, T_V, X = 1) \] (A.46)
**A.2.3. AMMONIA-WATER**

The bulk liquid and vapor phase dynamic viscosity ($\mu$), surface tension ($\sigma$) and thermal conductivity ($\lambda$) are computed using the method by Conde (2006) [36] as the following functions of phase bulk molar fraction and phase bulk temperature:

$$\mu_L = f(\bar{x}_b, T_{L,b}) \quad (A.47)$$

$$\mu_V = f(\bar{y}_b, T_{V,b}) \quad (A.48)$$

$$\sigma_L = f(\bar{x}_b, T_{L,b}) \quad (A.49)$$

$$\sigma_V = f(\bar{y}_b, T_{V,b}) \quad (A.50)$$

$$\lambda_L = f(\bar{x}_b, T_{L,b}) \quad (A.51)$$

$$\lambda_V = f(\bar{y}_b, T_{V,b}) \quad (A.52)$$

As a faster alternative to the FluidProp method, also the bulk density ($\rho$) and specific heat ($c_p$) can be calculated:

$$\rho_L = f(\bar{x}_b, T_{L,b}) \quad (A.53)$$

$$\rho_V = f(\bar{y}_b, T_{V,b}) \quad (A.54)$$

$$c_{p,L} = f(\bar{x}_b, T_{L,b}) \quad (A.55)$$

$$c_{p,V} = f(\bar{y}_b, T_{V,b}) \quad (A.56)$$

The liquid phase mass diffusivity ($D_L$) is also computed using the same method:

$$D_L = f(\bar{x}_b, T_L) \quad (A.57)$$

A relation for the vapor phase mass diffusivity ($D_V$) is not described in the method by Conde (2006) [36]. This value is computed using the method by Fuller et al. (1966) [102] instead, which uses the pressure and temperature of the vapor phase as inputs.

$$D_V = f(P, T_V) \quad (A.58)$$

The molar fractions of the liquid and vapor phase at the interface can also be based on the method by Conde (2006) [36], using the pressure ($P$) and interface temperature ($T_i$):

$$\bar{x}_i = f(P, T_i) \quad (A.59)$$

$$\bar{y}_i = f(P, T_i) \quad (A.60)$$
The same applies to the phase specific enthalpies, in the bulk or at the interface:

\[ h_{L,b} = f(\bar{x}_b, T_{L,b}) \]  (A.61)

\[ h_{V,b} = f(\bar{y}_b, T_{V,b}) \]  (A.62)

\[ h_{L,i} = f(\bar{x}_i, T_i) \]  (A.63)

\[ h_{V,i} = f(\bar{y}_i, T_i) \]  (A.64)

**A.2.4. Seawater**

The calculated vapor quality is a function of the seawater salinity \((S)\) and the temperature \((T)\) and should be equal to zero or very small:

\[ X = f(S, T) \approx 0 \]  (A.65)

The density, dynamic viscosity, specific heat capacity and thermal conductivity in case seawater is used of a certain salinity are given as function of \(S\) and \(T\) as well, through the relations proposed by Sharqawy et al. (2011) [48]:

\[ \rho = f(S, T) \]  (A.66)

\[ \mu = f(S, T) \]  (A.67)

\[ c_p = f(S, T) \]  (A.68)

\[ \lambda = f(S, T) \]  (A.69)

**A.3. Plate Heat Exchanger Geometry**

The channel gap is given by subtracting the plate thickness from the compressed plate pitch:

\[ d_g = p_{cp} - d_p \]  (A.70)

The corrugation amplitude is half the channel gap:

\[ \hat{a} = d_g / 2 \]  (A.71)

For non-contacting plates the ratio of \(d_g\) to \(\hat{a}\) will be larger than 2, but this case is not considered in this work. An overview of the important parameters related to the corrugation geometry is given in Figure A.1.

The width of the stacked plates is given by the following relation:

\[ W_p = d_o + N_p \cdot p_{cp} \]  (A.72)
Figure A.1: Representation of the corrugation profile including the compressed plate pitch ($p_{cp}$), plate thickness ($d_p$), channel gap ($d_g$) and channel amplitude ($\hat{a}$). The original figure is presented in Winkelman (2010) [35].

The number of channels is one less than the number of plates:

$$N_c = N_p - 1$$  \hspace{1cm} (A.73)

The number of walls through which the heat transfer occurs is two less than the number of plates, since the outer walls are assumed to be isolated on the outside resulting in no heat flux through these walls:

$$N_w = N_p - 2$$  \hspace{1cm} (A.74)

The number of working fluid channels, for an uneven number of channels and assuming the secondary channels to be the outer channels, is given by:

$$N_{c,wf} = \frac{1}{2}(N_c - 1)$$  \hspace{1cm} (A.75)

The number of cooling water channels is then:

$$N_{c,cw} = \frac{1}{2}(N_c + 1)$$  \hspace{1cm} (A.76)

The single plate plane projected area is given by:

$$A_0 = L_p \cdot B_p$$  \hspace{1cm} (A.77)

The total area of plates is obtained by summing up all plane projected areas:

$$A_{tot} = N_p \cdot A_0$$  \hspace{1cm} (A.78)

The single channel cross-sectional area is given through multiplication of the channel gap and channel width:

$$A_c = d_g \cdot B_p$$  \hspace{1cm} (A.79)

The total channels cross-sectional area is obtained by multiplying the amount of channels with the cross-sectional area per channel:

$$A_{c,tot} = N_c \cdot A_c$$  \hspace{1cm} (A.80)
The working fluid channels cross-sectional area is obtained by multiplying the amount of working fluid channels with the cross-sectional area per channel. A similar calculation is used for the cooling water channels:

\[ A_{wf} = N_{c,wf} \cdot A_c \] (A.81)

\[ A_{cw} = N_{c,cw} \cdot A_c \] (A.82)

The detailed geometry is discussed next, related to the corrugation pattern. Since most plates have sinusoidal corrugation patterns, or patterns very close to sinusoidal, the patterns may be defined by three main geometrical parameters: the corrugation pitch (\(p\)), the corrugation amplitude (\(\hat{a}\)) and the chevron angle (\(\phi\)).

The chevron angle to the main flow direction is shifted under an angle of 90° as compared to the chevron angle to the horizontal, since the main flow direction is vertically downward:

\[ \phi_{flow} = 90° - \phi \] (A.83)

The chevron angle can also be expressed in radians (\(\phi_{rad}\)), for both the chevron angle to flow and to the horizontal, using the following equation:

\[ \phi_{rad} = \frac{2\pi \phi}{360°} \] (A.84)

The pitch gives the length of a single corrugation of the periodic pattern along the flow direction. The corrugation wavelength gives the length of a single corrugation of the periodic pattern along the direction of the corrugation pattern itself. It is a function of the corrugation pitch and the chevron angle of the plates (Winkelmann, 2010 [35]):

\[ \Lambda = p \cdot \sin(\phi_{flow}) \] (A.85)

The difference between the corrugation pitch and wavelength, and their dependency on the chevron angle is illustrated in Figure A.2. Another length characteristic of the flow in the corrugated channels is given in Stephan et al. (2010) [8]:

\[ L_{corr} = \frac{\Lambda}{\sin(2\phi_{flow})} \] (A.86)

The wave number or corrugation parameter \(\Omega\) is a function of the corrugation amplitude \(\hat{a}\) and wavelength \(\Lambda\) (Stephan et al. (2010) [8]):

\[ \Omega = \frac{2\pi \hat{a}}{\Lambda} \] (A.87)

The channel gap is twice the amplitude, an alternative definition for the corrugation parameter is thus (Tribbe and Müller-Steinhagen (2001) [63]):

\[ \Omega = \pi d_g \Lambda \] (A.88)

The ratio of the wavy plate surface to its plane projection is defined by the area enlargement factor (\(\Gamma\)) (Stephan et al. (2010) [8]):

\[ \Gamma(\Omega) \approx \frac{1}{6} \left( 1 + \sqrt{1 + \Omega^2} + 4\sqrt{1 + \Omega^2/2} \right) \] (A.89)
Since the detailed geometry is not specified by manufacturers of commercial plate heat exchangers plates, the area enlargement factor is set to a typical value ($\Gamma = 1.17$). This then allows computation of the wave number ($\Omega$), corrugation wavelength ($\Lambda$) and corrugation pitch ($p$) through equations A.85, A.87 and A.89.

Different definitions exist for defining the hydraulic diameter of plate heat exchangers, in this work the definition of the hydraulic diameter presented by Stephan et al. (2010) [8] is used:

$$d_h = \frac{4 \hat{a}}{\Gamma}$$  \hspace{1cm} (A.90)

The hydraulic diameter represents the characteristic length scale for flow in a given geometry. The enlarged plate surface area ($A_p$) describes the effective surface area of the corrugated plates available for heat transfer (Stephan et al. (2010) [8]):

$$A_p = \Gamma \cdot A_0$$  \hspace{1cm} (A.91)

The total surface area available for heat transfer is then:

$$A_{p,\text{tot}} = N_p \cdot A_p$$  \hspace{1cm} (A.92)

The wetted perimeter of the flow of the channels is defined through the following relation:

$$\Pi = \frac{4 \cdot A_c}{d_h}$$  \hspace{1cm} (A.93)

**A.4. ALTERNATIVE MODEL CORRELATIONS**

**A.4.1. HEAT TRANSFER**

**DIRECT TWO-PHASE FLOW**

Direct two-phase heat transfer correlations for condensing two-phase flows are discussed here, used in Chapter 4 to compare the heat transfer predictions to those of the detailed model. First, several dimensionless numbers need to be specified that have not previously
been discussed in the main text or appendices of this thesis. The Lockhart-Martinelli parameter expresses the liquid fraction of a flowing fluid, and is commonly used in condensing two-phase heat transfer (and pressure drop) correlations. The parameter $\chi$ is defined as the ratio of the liquid frictional pressure drop over the channel length $(dP_f/dz)_L$ over that one for the gas or in this case the vapor $(dP_f/dz)_V$:

$$\chi^2 = \frac{(dP_f/dz)_L}{(dP_f/dz)_V}$$  \hspace{1cm} (A.94)

A relation for the parameter is obtained from knowing the mass flow rates and densities of both the phases:

$$\chi = \frac{\dot{m}_L}{\dot{m}_V} \sqrt{\frac{\rho_V}{\rho_L}}$$  \hspace{1cm} (A.95)

Here, the definition is used presented in the study by Palmer et al. (2000) [100]:

$$\chi = \left( \frac{1 - X}{X} \right)^{0.9} \left( \frac{\rho_V}{\rho_L} \right)^{0.5} \left( \frac{\mu_V}{\mu_L} \right)^{0.1}$$  \hspace{1cm} (A.96)

The two-phase multiplier ($\phi_L$) gives the two-phase frictional pressure drop relative to the liquid single phase pressure drop.

$$\phi_L^2 = \frac{(dP_f/dz)_TP}{(dP_f/dz)_L}$$  \hspace{1cm} (A.97)

The two-phase multiplier is usually defined as a function of the Lockhart-Martinelli parameter, and in the study by Palmer et al. (2000) [100] given as:

$$\phi_L = \left[ 1 + \frac{12}{\chi} + \frac{1}{\chi^2} \right]^{0.5}$$  \hspace{1cm} (A.98)

The Galileo number gives the ratio of gravitational to viscous forces and is described by the following relation:

$$Ga = \frac{gd^3\rho_L(\rho_L - \rho_V)}{\mu_L^2}$$  \hspace{1cm} (A.99)

The parameter is used in certain heat transfer relations for general or viscous flow. The Convection number is defined in Winkelmann (2010) [35]:

$$Co = \left( \frac{\rho_V}{\rho_L} \right)^{1/2} \left( \frac{1 - X}{X} \right)^{0.8}$$  \hspace{1cm} (A.100)

The reduced pressure is the ratio of the pressure to the critical pressure:

$$P_{red} = \frac{P}{P_{crit}}$$  \hspace{1cm} (A.101)

In the study by Palmer et al. (2000) [100], the acentric factor ($\omega$) is defined as function of the reduced pressure:

$$\omega = -\log_{10}(P/P_{crit})$$  \hspace{1cm} (A.102)
The heat transfer correlations can now be given in terms or previously defined parameters. For the study of Palmer et al. (2000) [100] the liquid phase heat transfer coefficient is defined as:

$$\alpha_L = \frac{\lambda_L}{d_h} \cdot 0.16 R e_L^{0.89} P r_L^{0.3}$$  \quad (A.103)

The two-phase heat transfer coefficient, valid for $13 < R e_L < 230$ is given as:

$$\alpha_{TP} = \alpha_L^{0.387} \phi_L^{0.0827} G a^{0.346} P r_{red}^{1.5} \omega^{1.5}$$  \quad (A.104)

The heat transfer correlation for the liquid phase and the two-phase flow by Thonon and Bontemps (2002) [69], valid for $50 < R e 2000$ are given by:

$$\alpha_L = \frac{\lambda_L}{d_h} \cdot 0.347 R e_L^{0.653} P r_L^{0.33}$$  \quad (A.105)

$$\alpha_{TP} = 1564 R e_{eq}^{-0.76} \alpha_L$$  \quad (A.106)

The relations by Winkelmann (2010) [35] are valid for $10 < R e < 1100$, and are given for a mixture and pure fluid respectively as:

$$\alpha_{TP,mix} = \frac{\lambda_L}{d_h} \cdot 16.8 R e_{eq}^{0.29} P r_L^{1/3}$$  \quad (A.107)

$$\alpha_{TP,pf} = \frac{\lambda_L}{d_h} \cdot 94 C a^{-0.46} P r_L^{1/3}$$  \quad (A.108)

The relation by Yan et al. (1999) [68] for two-phase flow is valid for $R e > 200$:

$$\alpha_{TP} = \frac{\lambda_L}{d_h} \cdot 16.8 R e_{eq}^{0.4} P r_L^{1/3}$$  \quad (A.109)

The direct two-phase heat transfer correlations can be used to alternatively calculate the overall heat transfer coefficient and heat flux or heat flow across the wall. The liquid and vapor phase temperature are not defined individually using the direct methods, only the fluid bulk temperature and the temperature near the wall. The effective bulk temperature for two-phase flow ($T_{TP,b}$) is estimated as the mass-weighted average temperature of the liquid and vapor phases:

$$T_{TP,b} = \frac{m_L T_{L,b} + m_V T_{V,b}}{m_L + m_V}$$  \quad (A.110)

This temperature had no direct physical relevance, but is used merely as a tool to estimate the effective temperature of the two-phase mixture. The overall heat transfer coefficient ($U$) is calculated similarly as described in Chapter 3, through the combined heat transfer resistances of the wall, working fluid and cooling water side given in Equation 3.134.

Here, $\alpha_{w f,w}$ is the heat transfer coefficient on the working fluid side near the wall, and is equal to $\alpha_{TP}$ of one of the studies discussed previously. The heat flow across the wall is given through Equation 3.135, where the effective two-phase bulk temperature ($T_{TP,b}$) is used as the working fluid bulk phase temperature ($T_{w f,b}$).
**TWO-PHASE FILM FLOW**

The used correlation for the liquid phase film Nusselt number is presented in Chapter 3. As an alternative to the relation for Nusselt laminar vapor flow over a plate, the local ($Nu_V$) and average Nusselt ($\overline{Nu}_V$) number for laminar flow in tubes may be used:

\[
Nu_V = 1.08(Gz_V)^{-1/3}, Gz < 0.05 \quad (A.111)
\]

\[
\overline{Nu}_V = 1.62(Gz_V)^{-1/3}, Gz < 0.05 \quad (A.112)
\]

\[
Nu_V = \overline{Nu}_V = 3.66, Gz > 0.1 \quad (A.113)
\]

The Graetz number of the vapor phase ($Gz_V$) as a function of length along the heat exchanger is given by:

\[
Gz_V = \frac{d_V Re_V Pr_V}{z} \quad (A.114)
\]

Here, the Reynolds ($Re_V$) and Prandtl ($Pr_V$) numbers of the vapor phase are given using their general definitions in equations A.8 and A.7.

Another alternative to calculating the vapor phase Nusselt number is the relation for a flat plate parallel to flow:

\[
Nu_V = 0.332 Re_V^{1/2} Pr_V^{1/3}, Re < 3 \cdot 10^5 \quad (A.115)
\]

The used correlation for the liquid phase film thickness is presented in Chapter 3. In case capillary forces are dominant, which is not expected to be the case for flow in the heat exchangers studied, other relations for the film thickness should be used. Relations taking into account surface tension for flow in corrugated channels have not been found, relations for tubes however are known. In Nefs (2013) [77] a model is constructed for ammonia-water absorption in small tubes, and the equation for capillary tubes described in Irandoust and Andersson (1989) [103] is used:

\[
d_{\text{film}} = 0.67 Ca^{2/3} d_h \quad (A.116)
\]

In this relation, the hydraulic diameter for the flow $d_h$ and the Capillary number $Ca$ should be known. More about the Capillary number and other dimensionless numbers related to the effects of surface tension are discussed further ahead in this section. Another length scale related to the film thickness, is described in Winkelmann (2010) [35]:

\[
d_{\text{film}} = \left( \frac{\mu_L/\rho_L}{g} \right)^{1/3} \quad (A.117)
\]

This number is however not the actual film thickness itself, but the two are correlated through an additional constant. It is assumed that the vapor and liquid phase move through the channels with a different average actual velocity, which justifies the use of the Nusselt laminar film theory for condensation relations. In case the vapor and liquid phases would move at equal velocity, the film thickness would depend only on the relative volumetric flows of the phases:
\[ d_{film} = d_g \frac{\dot{V}_{L, CV}}{\dot{V}_{L, CV} + \dot{V}_{V, CV}} = d_g \frac{\dot{m}_{L, CV}/\rho_L}{\dot{m}_{L, CV}/\rho_L + \dot{m}_{V, CV}/\rho_V} \]  

(A.118)

Here, \( d_g \) is the channel gap and thus the mean distance between the two plates of the channel.

**Single phase flow**

The used heat transfer correlations for single phase flow are given in Chapter 3. Alternative Nusselt numbers for single phase flow, independent of the axial distance in the channel, are given by the Dittus-Boelter and Coulson and Richardson equations. The Dittus-Boelter equation (used in Kanaris *et al.* (2006) \[104\] and Zhang and Che (2011) \[54\]) is valid for turbulent flow (\( Re \gtrsim 10000 \)), and a certain range of Prandtl numbers (0.6 \( \leq Pr \leq 160 \)). The position inside the channel needs to be larger than the entrance length by a factor 10 (\( \frac{z}{d_g} \gtrsim 10 \)). For cooling the fluid the exponent of the Prandtl number \( C = 0.3 \):

\[ Nu = 0.023 Re^{0.8} Pr^C \]  

(A.119)

The Coulson-Richardson equation (used in Kirkenier (2014) \[3\]) is valid for turbulent flows:

\[ Nu = 0.26 Re^{0.65} Pr^{0.4} \]  

(A.120)

**A.4.2 Pressure drop**

The two-phase flow friction factor (\( f_{TP} \)) is calculated using Equation 3.10, proposed by Yan *et al.* (1999) \[68\]. Alternatively, the two-phase friction factor may be based on the work by Han *et al.* (2003) \[99\]. Here, the two-phase friction factor is described by the following relation, valid for 250 < \( Re < 750 \):

\[ f_{TP} = C_1 Re_{eq}^{C_2} \]  

(A.121)

Here, \( Re_{eq} \) is given by Equation 3.11. The constants \( C_1 \) and \( C_2 \) are dependent on certain geometrical parameters, and given by the following equations:

\[ C_1 = 3521.1 \left( \frac{p_{cp}}{d_h} \right)^{4.17} \left( \frac{\pi}{2} - \varphi_{rad} \right)^{-7.75} \]  

(A.122)

\[ C_2 = -1.024 \left( \frac{p_{cp}}{d_h} \right)^{0.0925} \left( \frac{\pi}{2} - \varphi_{rad} \right)^{-1.13} \]  

(A.123)
FLOW PATTERNS FOR TWO-PHASE FLOW

This appendix chapter contains a summary of studies performed related to flow patterns for two-phase flow in corrugated channels. I also covers a number of studies for two-phase flow in tubes, to assist in comparing and discussing results.

B.1. CORRUGATED CHANNELS

In Tribbe and Müller-Steinhagen (2001)[52, 63] it is stated that little information is available on two-phase flows and phase changes occurring in plate heat exchangers. Even though it is clear that flow patterns and their transitions are influenced by the channel geometry, little is still known about two-phase phenomena in narrow channels. From relations between pressure gradient and flow patterns derived for pipe flow, it is evident that phenomenological models lead to better understanding and prediction for a larger number of cases. The identification of flow patterns in plate heat exchangers is therefore considered as an important step towards understanding their behavior. Experiments were performed to calculate pressure drops and visualize flow paths and flow patterns for gas-liquid two phase flows in plate heat exchangers. Transition boundaries between the flow patterns are found to be a function of the chevron angle. In total five different flow patterns are observed:

1. Regular bubbly flow (Figure B.1): at low gas flow rates individual bubbles of different sizes flow along the channels, their flow paths and possible breakage influenced by the flow conditions, geometric conditions (surfaces) and the presence of other bubbles.

2. Irregular bubbly flow (Figure B.2): at higher gas flow rates, large irregularly shaped regions of gas bubbles form.

3. Churn flow (Figure B.3): the appearance of high velocity liquid slugs indicates the transition to the churn flow regime, under which conditions are usually highly turbulent.

4. Film flow (Figure B.4): a thin liquid film flows along the furrows over which a fast-moving gas stream flows throughout the plate, demonstrating several similarities to annular flow in pipes in terms of surface phenomena such as ripples, roll waves and disturbance waves.
5. Partial film flow (Figure B.5): at the highest gas flow rates, the liquid no longer wets the entire surface of the plates and dry surfaces start to appear. Shear forces cause secondary flows in the gas phase.

![Regular bubbly flow](image1.png)

**Figure B.1:** Regular bubbly flow, air-water downflow in a plate heat exchanger (Tribbe and Müller-Steinhagen (2001) [63]).

![Irregular bubbly flow](image2.png)

**Figure B.2:** Irregular bubbly flow, air-water downflow in a plate heat exchanger (Tribbe and Müller-Steinhagen (2001) [63]).

The observed flow patterns have certain similarities to two-phase flow in circular and rectangular channels of comparable hydraulic diameter. At the central spine of the plates, special flow phenomena occur. Depending on the chevron angle and observed flow pattern in the plate, flows follow different paths and may be reflected into a different direction, unable to cross the central spine. The flow patterns observed at the central spine under regular bubbly flow are presented in Figure B.6.

Flow pattern maps are presented in terms of superficial velocities, since no physical property effects were investigated. The maps present the transitions between the previously described flow patterns, which are different for plates of different combinations of chevron angles on either side of the central spine. Flow pattern maps are presented for plates of different chevron angles in figures B.7 and B.8. Apparently, the flow pattern map is the same for unsymmetrical plates with a high and low chevron angle as for symmetrical plates of two low chevron angles, and the transition boundaries are nonlinear functions of the chevron angles.
Figure B.3: Churn flow, air-water downflow in a plate heat exchanger (Tribbe and Müller-Steinhagen (2001) [63]).

Figure B.4: Film flow, air-water downflow in a plate heat exchanger (Tribbe and Müller-Steinhagen (2001) [63]).

Figure B.5: Partial film flow, air-water downflow in a plate heat exchanger (Tribbe and Müller-Steinhagen (2001) [63]).
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Figure B.6: Regular bubbly flow at the herringbone spine in a plate heat exchanger (Tribbe and Müller-Steinhagen (2001) [63]).

Figure B.7: Flow pattern map for the 30°/30° and 30°/60° chevron angle configurations (Tribbe and Müller-Steinhagen (2001) [63]).
Vlasogiannis et al. (2002) [59] performed visualization experiments for air-water two-phase flow in a plate heat exchanger. They made visualization possible by replacing the movable end cover of the exchanger by a plexiglass plate. The corrugation pattern of a steel plate was embossed on the inner surface using lithographic techniques. The plate geometry is found in Figure B.9.

Based on observations of the flow recorded by a high speed camera, four major flow regimes are identified: A, B, C, AB and BA. These are presented in a flow map in Figure B.10. The regimes are defined for different combinations of gas and liquid superficial velocities.

A and B form the two extremes of these flow patterns, where regime A is present for all low gas flow rates. In this regime the liquid mainly moves in the form of rivulets at the bottom of the corrugation furrows, leaving most of the channel space for a continuous gas phase. In regime B, air is dispersed in the form of small bubbles inside a liquid-continuous stream that covers the entire channel space. Visualizations of regimes A and B are found respectively in Figure B.11 and Figure B.12. The detailed characteristics of the gas-sheared rivulet flow occurring in regime A are to a large extent unknown, for example if the exchanger plates in this regime are only partially wetted or whether a thin liquid layer is actually sustained everywhere. This may depend on the wetting characteristics of the plate material. Regimes AB and BA can both be described as churn flow, and regime C as slug flow.

Winkelmann (2010) [35] discusses visualization of flow patterns for adiabatic, vertical, downward, co-current, air-water two-phase flow in corrugated channels. A flow pattern map is presented for air-water flow. It is stated that flow pattern maps exist in literature that show variation with the corrugation angle. For certain flow patterns the phase distribution varies not only on the level of one corrugation, but also across the channel as the liquid flows along the edges and the spine of the channel. After identifying the flow patterns, models can be developed for pressure drop and heat transfer corresponding to the flow pattern found.

The combined mass flux of air and water is kept constant at two values, and the vapor quality is constant along the channel since no phase change occurs. The resulting flow
Figure B.9: A steel plate of the heat exchanger and the plexiglass cover with the embossed corrugation pattern (Vlasogiannis et al. (2002) [59]).

Figure B.10: The flow regime map for air/water downflow in a channel of the plate heat exchanger (Vlasogiannis et al. (2002) [59]).
pattern map is presented in Figure B.13. Five different flow patterns are defined:

- Bubble flow \((b)\): For low air quality, not representative of the conditions inside a condenser (Figure B.14).

- Churn flow \((c)\): Characterized by swarms of tiny bubbles rushing in successive waves of sweeping motion through the channel (Figure B.15).

- Wavy flow \((w)\): Described by liquid flowing along the corrugations and by an agitated air–water interface (Figure B.16).

- Film flow \((f)\): Characterized by liquid that follows the orientation of the corrugations (Figure B.17).

- Partial wetting flow \((p)\): The plates are only partially wetted by the liquid present.

The observed irregular bubbly flow is comparable to slug flow found in tubes, but an air bubble can not take the form of a slug in a corrugated channel. It is found that, when looking at the channel as a whole, maldistribution of the phases occurs, especially at low flow rates. The liquid does not reach the top left corner, and further down the channel the liquid concentrates along the middle spine and edges of the channel.

Nilpueng and Wongwises (2010) [64] discuss the observed flow patterns for two-phase vertical, single pass, countercurrent flow in stainless steel commercial plate heat exchangers with corrugated sinusoidal shapes of unsymmetrical chevron angles, for both upflow and downflow. In both flow directions the annular-liquid bridge flow pattern is observed. The bubbly flow pattern is only found in upward flow, and the slug flow pattern only in downward flow. The paper emphasizes that a clear understanding of two-phase flow patterns is required to understand the hydrodynamics of the two-phase flow. The resulting flow pattern in the heat exchanger will influence many aspects related to mass, heat and momentum transfer, and to a large extent determine the pressure drop. The identification of the two-phase flow patterns is reported in terms of superficial gas and liquid velocities.
Figure B.13: Flow-pattern map for adiabatic air–water flow in a corrugated plate channel (Winkelmann (2010) [35]).

Figure B.14: Bubble flow of air in water, in a plate heat exchanger with four contact points circled. A bubble is dividing upstream of one of the contact points. (Winkelmann (2010) [35]).

Figure B.15: Churn flow of colored water and air, in a plate heat exchanger with four contact points circled (Winkelmann (2010) [35]).
A distinction is made between the cases for the upward and for the downward flows. For upward flow the following patterns were observed:

1. Bubbly flow (Figure B.18): bubbles move both through the troughs of the channel and directly upwards through the spacing between two plates. The reason for its direction is most probably due to encountered flow resistances.

2. Bubble recirculation flow (Figure B.19): when the air flow rate increases in bubbly flow, interfacial shear between bubbles flowing in the troughs and those in the middle of the channel results in recirculation of bubbles in the trough. The air bubble size decreases due to collisions.

3. Annular-liquid bridge flow (Figure B.20): a thin water film is present on the plate surface, and some water forms liquid bridges spanning the corrugation crests of the heat exchanger plates.

Patterns observed for downward flow are:

1. Slug flow (Figure B.21): alternating full water flow and air flow are observed in the flow channel.

2. Annular-liquid bridge flow/air alone flow (Figure B.22): different patterns are observed on either side of the unsymmetrical plate heat exchanger, where on one side the flow consists solely of air.

3. Annular-liquid bridge flow (Figure B.23): a water film covers the surface of the whole heat exchanger plate.

Broad transition bands exists for the flow patterns due to the uncertainty at the vicinity of the transitions between flow patterns. The flow regime maps are presented in Figure B.24.

Grabenstein and Kabelac (2012) [61] describe observed flow patterns for a single gap of a plate heat exchanger, during the two-phase flow of air-water and boiling flow of refrigerant R-365mfc. Flow pattern maps were made for two different corrugation angles of the plates. Three main clearly identifiable flow patterns were observed:
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Figure B.18: Bubbly flow for upward air-water flow in a plate heat exchanger (Nilpueng and Wongwises (2010) [64]).

Figure B.19: Bubble recirculation flow for upward air-water flow in a plate heat exchanger (Nilpueng and Wongwises (2010) [64]).

Figure B.20: Annular-liquid bridge flow for upward air-water flow in a plate heat exchanger (Nilpueng and Wongwises (2010) [64]).
1. Film flow: occurs at high fractions of the liquid phase.

2. Bubbly flow: occurs at low rates of liquid flow and high rates of vapor.

3. Slug flow: occurs at high flow rates of the liquid and vapor phases.

Transition regimes are observed to occur between the different flow patterns, and the flow pattern map is presented in Figure B.25. Models for the heat transfer and pressure drop mechanism were developed individually for all three flow patterns. This was done using the performed experiments, but it was also stated that more experiments should be done to get a better understanding of the phenomena.

**B.2. OTHER GEOMETRIES**

Hewitt and Roberts (1969) [65] discuss two-phase gas liquid flow in vertical tubes. The following flow patterns were identified:

1. Plug flow
2. Churn flow
3. Annular flow
4. Wispy annular flow
Figure B.22: Annular-liquid bridge/air-alone flow for downward air-water flow in a plate heat exchanger (Nilpueng and Wongwises (2010) [64]).

Figure B.23: Annular-liquid bridge flow for downward air-water flow in a plate heat exchanger (Nilpueng and Wongwises (2010) [64]).
Figure B.24: Flow pattern map for upward air-water flow (left) and downward air-water flow (right) in a plate heat exchanger (Nilpueng and Wongwises (2010) [64]).

Figure B.25: Flow pattern map of the two-phase flows of air/water (blue dotted line) and liquid/gaseous R365mfc (red continuous line) in a corrugated gap (Grabenstein and Kabelac (2012) [61]).
5. Bubble flow with developing structure

Several observations are made about the studied flow patterns. At high liquid flow rates a gradual transition from bubble flow to wispy annular flow was observed, without the intermediate stage characteristic of plug flow. Flow pattern transitions occur at very different superficial phase velocities for different studied cases, implying that a simple scaling on the basis of phase velocity is not valid. Effective description of flow patterns based on other simple parameters such as momentum flux is also unlikely, and physical properties of the fluids and characteristics of the channels will have to be taken into account. Different methods for scaling of flow patterns have already been proposed in literature. These take into account the superficial mass velocities of the gas and liquid phases, and certain parameters that are functions of the physical properties of the fluids. The physical properties include the density of both phases, the surface tension and the liquid viscosity. The obtained flow pattern maps using $\rho_L \dot{V}_L^2$ and $\rho_G \dot{V}_G^2$, as parameters on the x- and y-axis respectively, are presented in Figure B.26.

![Flow pattern map](image)

Figure B.26: Flow pattern map plotted from different experimental data for flow in vertical tubes (Hewitt and Roberts (1969) [65]).

are distinguished for downflow, and are presented here in the order of increasing gas flow rate:

1. Coring-bubble
2. Bubbly-slug
3. Falling film
4. Falling bubbly-film
5. Froth
6. Annular

Schematic representations of each of the flow patterns are presented in Figure B.27. In this study, air-water was used as mixture for defining the different observed flow regimes. During the experiments it was observed that transitions of one flow regime to another were gradual with respect to the fluid flow rates. Transitions between bubble and slug flow and from slug to froth flow were previously derived analytically. The proposed flow pattern maps for both vertical upflow and downflow are given in figures B.28 and B.29. Broad bands are given to approximate the representations of the zones in which gradual transition occurs of one flow patterns to another. Governing parameters and correlations for the transitions between patterns are discussed further ahead.
Figure B.27: Flow patterns observed in vertical downflow in vertical tubes (Oshinowo and Charles (1974) [66]).
Figure B.28: Proposed general flow pattern map for two-phase gas-liquid vertical upflow in vertical tubes (Oshinowo and Charles (1974) [66]).

Figure B.29: Proposed general flow pattern map for two-phase gas-liquid vertical downflow in vertical tubes (Oshinowo and Charles (1974) [66]).
MODELLING RESULTS

In this appendix the modelling results of the base case not included in the main text are presented. Inlet and outlet values in the control volume are usually very close and not distinguishable in the figures, due to the large amount of control volumes. In certain figures, the same parameters are plotted based on the calculations of both the working fluid and cooling water loop, and the profiles should overlap if the simulation has converged.

Figure C.1: Specific enthalpy of the cooling water flow in the bulk at the inlet and outlet of the control volume, as function of the position inside the heat exchanger.

Figure C.2: Specific enthalpy of the working fluid flow in the bulk of either phase and the mean two-phase bulk enthalpy at the inlet and outlet of the control volume, and the specific enthalpy of vaporization, as function of the position inside the heat exchanger.
Figure C.3: Specific enthalpy of the working fluid vapor and liquid phase flow in the bulk and at the interface, as function of the position inside the heat exchanger.

Figure C.4: Specific enthalpy, species specific partial enthalpy and partial enthalpy differences of the working fluid vapor and liquid phase flow at the interface, as function of the position inside the heat exchanger.

Figure C.5: Density of the cooling water and working fluid vapor and liquid phase, as function of the position inside the heat exchanger.

Figure C.6: Specific heat of the cooling water and working fluid vapor and liquid phase at the control volume inlet and outlet, as function of the position inside the heat exchanger.
Figure C.7: Thermal conductivity of the wall, cooling water and working fluid vapor and liquid phase, as function of the position inside the heat exchanger.

Figure C.8: Thermal diffusivity of the cooling water and working fluid vapor and liquid phase, as function of the position inside the heat exchanger.

Figure C.9: Dynamic viscosity of the cooling water and working fluid vapor and liquid phase, as function of the position inside the heat exchanger.

Figure C.10: Kinematic viscosity of the cooling water and working fluid vapor and liquid phase, as function of the position inside the heat exchanger.

Figure C.11: Surface tension of the working fluid vapor and liquid phase, as function of the position inside the heat exchanger.

Figure C.12: Diffusion coefficient of the working fluid vapor and liquid phase, as function of the position inside the heat exchanger.
Figure C.13: Volumetric expansion coefficient of the working fluid vapor phase, as function of the position inside the heat exchanger.

Figure C.14: Molar concentration of the working fluid vapor and liquid phase, as function of the position inside the heat exchanger.

Figure C.15: Molar fractions of the total working fluid and the vapor and liquid phase, as function of the position inside the heat exchanger.

Figure C.16: Mass fractions of the total working fluid and the vapor and liquid phase, as function of the position inside the heat exchanger.

Figure C.17: Schmidt number of the working fluid vapor and liquid phase, as function of the position inside the heat exchanger.

Figure C.18: Lewis number of the working fluid vapor and liquid phase, as function of the position inside the heat exchanger.
Figure C.19: Prandtl number of the cooling water and working fluid vapor and liquid phase, as function of the position inside the heat exchanger.

Figure C.20: Prandtl number function of the working fluid vapor phase, as function of the position inside the heat exchanger.

Figure C.21: Rayleigh number of the working fluid vapor phase, as function of the position inside the heat exchanger.

Figure C.22: Graetz number of the working fluid vapor phase, as function of the position inside the heat exchanger.

Figure C.23: Lockhart-Martinelli parameter of the working fluid, as function of the position inside the heat exchanger.

Figure C.24: Two-phase multiplier of the working fluid, as function of the position inside the heat exchanger.
Figure C.25: Convection number of the working fluid, as function of the position inside the heat exchanger.

Figure C.26: Galileo number of the working fluid, as function of the position inside the heat exchanger.

Figure C.27: Sherwood number of the working fluid vapor and liquid phase, as function of the position inside the heat exchanger.

Figure C.28: Nusselt number of the cooling water and working fluid vapor and liquid phase, as function of the position inside the heat exchanger.

Figure C.29: Nusselt number of the working fluid vapor phase using different methods, as function of the position inside the heat exchanger.

Figure C.30: Nusselt number of the working fluid liquid phase using different methods, as function of the position inside the heat exchanger.
Figure C.31: Heat transfer coefficient of the working fluid vapor phase using different methods, as function of the position inside the heat exchanger.

Figure C.32: Heat transfer coefficient of the working fluid liquid phase using different methods, as function of the position inside the heat exchanger.

Figure C.33: The overall, liquid and vapor phase heat transfer coefficient of the model and two-phase direct heat transfer coefficients using different methods, as function of the heat exchanged between the two streams for the entire heat exchanger.
Figure C.34: The overall, cooling water and working fluid liquid and vapor phase heat transfer coefficient, as function of the heat exchanged between the two streams for the entire heat exchanger.

Figure C.35: Heat transfer coefficients and mass transfer coefficients of the working fluid liquid and vapor phase in heat transfer coefficient units, as function of the position inside the heat exchanger.

Figure C.36: The total, wall, cooling water and working fluid liquid and vapor phase thermal conductance, as function of the position inside the heat exchanger.
Figure C.37: The overall, wall, cooling water and working fluid thermal conductance, as function of the position inside the heat exchanger.

Figure C.38: Temperatures of the working fluid and cooling water in the bulk phase and at the wall based on both the working fluid and cooling water side, as function of the position inside the heat exchanger.
Figure C.39: Temperatures of the working fluid and cooling water in the bulk phase and at the wall based on both the working fluid and cooling water side, as function of the heat exchanged between the two streams for the entire heat exchanger.

Figure C.40: Temperatures related to the working fluid interface, liquid and vapor phase, as function of the position inside the heat exchanger.

Figure C.41: Temperatures related to the cooling water, as function of the position inside the heat exchanger. $T_{lim}$ is the boiling temperature at given pressure and temperature.
Figure C.42: Heat flow changes of the working fluid and cooling water channels (two control volumes), as function of the position inside the heat exchanger.

Figure C.43: Heat flows across the wall, based on local or overall heat transfer and on both the working fluid and cooling water side, as function of the position inside the heat exchanger.

Figure C.44: Absolute difference in heat flux across the wall and interface based on the working fluid side, and the heat fluxes across the wall and at the liquid phase side of the interface, as function of the position inside the heat exchanger.

Figure C.45: Relative difference in heat flux across the wall and interface based on the working fluid side, as function of the position inside the heat exchanger. The positive differences are believed to be a consequence of small numerical errors.
Figure C.46: Working fluid mass flow rates of both species, as function of the position inside the heat exchanger.

Figure C.47: Working fluid and cooling water total mass flow rate and mass flow rates of both working fluid phases at the channel inlet and outlet, as function of the position inside the heat exchanger.

Figure C.48: Working fluid and cooling water total mass flow rate and mass flow rates in a single channel, as function of the position inside the heat exchanger.

Figure C.49: Working fluid and cooling water total volumetric flow rate and volumetric flow rates of both working fluid phases in a single channel, as function of the position inside the heat exchanger.
Figure C.50: Working fluid and cooling water total mass flux and working fluid two-phase equivalent mass flux, as function of the position inside the heat exchanger.

Figure C.51: Working fluid molar total fluxes for either phase across the interface in each control volume, and of both species, as function of the position inside the heat exchanger.

Figure C.52: Working fluid diffusive molar flux in the liquid and vapor phase, as function of the position inside the heat exchanger.

Figure C.53: Working fluid mass transfer coefficient in the liquid and vapor phase, as function of the position inside the heat exchanger.
Figure C.54: Film thickness of the liquid phase film using different methods, as function of the position inside the heat exchanger.
BIBLIOGRAPHY


