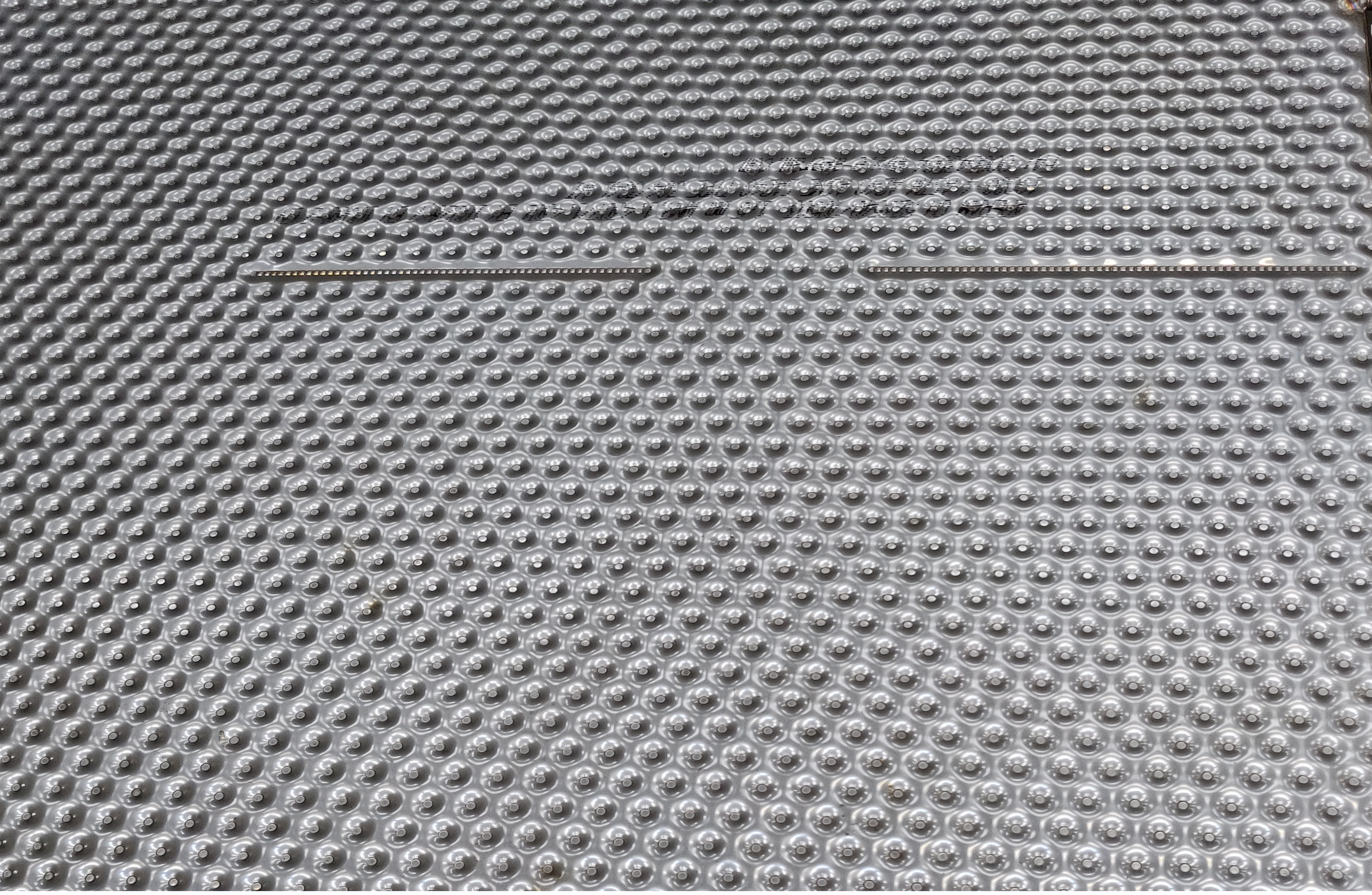

Parametric Analysis of a Falling Film Pillow Plate Evaporator

Lasse Olsen

Energy Technology, TEPE4-1009, 2021-05

Master's Project

STUDENT REPORT

AALBORG UNIVERSITY

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Abstract:

The purpose of this project was to create a parametric model, which can analyse the heat transfer for a flooded falling film pillow plate evaporator. The first step of modelling the heat transfer was to determine the thermal resistance. This was done by modelling the three types of heat transfer and accounting for whether dry out happens during the evaporation process and how the boiling heat transfer changes as a function of the quality of the refrigerant. This was then implemented using the cell method to model the heat exchanger. To validate the model it was compared to data provided by ATHCO, a company that produces pillow plates. Here it showed to be able to predict the overall heat transfer coefficient with an error margin of 14% mostly under predicting the heat transfer. This is an acceptable accuracy, so there was made a parametric analysis. Here the results showed that the best method for improving the heat transfer is to either increase the area by increasing the height, increase the temperature difference by increasing the inlet temperature or flow rate of the falling film or increase the flow rates of either the refrigerant or the falling film. Especially increasing the refrigerant flow rate is beneficial as this reduces the risk of dry out happening.

Summary

This project concerns the modelling of a falling film pillow plate evaporator and a following parametric analysis of this model. This model is made to assist in the dimensioning process of pillow plates evaporators, which can function as an component in a heat pump system. Heat pumps are an important part of the danish governments plan for reducing the CO2 emission in the the danish district heating system. The purpose of a pillow plate evaporator is to extract heat from a source and use the heat to evaporate a refrigerant as part of a heat pump cycle.

ATHCO is a producer of pillow plates and have been doing it since 1947, but the dimensioning process is currently based on older technology and experiments, which means that there is a lack of a good design tool. This is also due to pillow plates not being a well examined geometry for heat exchangers, which results in the theory for creating a design tool is limited.

The first step in modelling the heat transfer process is determining the types of heat transfer there is present and what parameters are important for modelling this process. For the geometry of a single pillow plates it is the height, width, welding spot pattern and expansion height. For the two fluids it is the type of fluid, pressure, temperature and flow rate. From this, the thermal resistance is modelled through Nusselt correlation for falling film, conduction through a plate and flow boiling. Here it is necessary to make some assumptions as there was found no correlation for falling film or boiling in a pillow plate during the literature study. Instead it was assumed that the flow behaviour of a pillow plate is similar to a vertical tube, which means a method developed for tubes can be used. For this method there was used a Nusselt correlation for pillow plates regarding the single phase heat transfer. This was then implemented in a cell method where there was accounted for how the change in quality affected the heat transfer. There was also investigated were there would be dry out in the evaporator and how that would affect the heat transfer.

This model was validated against data provided by ATHCO for how they usually design pillow plate evaporators. Here the model was compared to six different configurations of pillow plates and it was found that it is generally able to produced similar results, with a error margin of about $\pm 14\%$ where it tends to under

predict the heat transfer.

Following this there was made a parametric analysis were several of the input parameters were varied. Here it was found that there are several ways that the design of the pillow plates can be improved. The area can be increased, where especially increasing the height is effective, through the width can be increased instead if the pressure losses should be lowered. The heat transfer coefficient can be increased by increasing the flow rate of either the falling film or refrigerant. Increasing the flow rate of the refrigerant is especially beneficial as it also reduces the risk of dry out which severely decreases the heat transfer. Lastly the average temperature difference can be increased by having a higher inlet temperature of the falling film or increasing the flow rate. All of these changes will improve the heat transfer, but they needs to be considered together to achieve the optimal heat transfer.

The overall conclusion for this project is that there has been created a model that is capable of predicting the heat transfer of a falling film with a acceptable accuracy which can be used to design and improve the design of pillow plates. The model can be improved further by conducting experiments to further increase the validity of the model and adjust some of the assumptions and equations that has been used to create the model.

Contents

Preface

This project has been written by group *TEPE4-1009* at the Department of Energy Technology at Aalborg University as the master project during 4^{th} semester
The following software have been utilised during this project:

- **Matlab** - Modelling and data processing
- **Overleaf** - Writing and formatting the project
- **CoolProp** - Table values for fluid properties
- **Inkscape** - Creating figures

Readers' Guide

This project will present the work which has been done during on the master project concerning the modelling of a falling film pillow plate evaporator.

On page v the Table of Content can be found, where the tittles for the chapters, sections and subsections can be seen. These tittles will function as hyperlinks if viewed as a PDF.

The nomenclature can be found on page ix where all the abbreviations, subscripts, notation and symbols which will be used in the report is present with corresponding units.

The bibliography can be found on page 59 where all the literature used in this project is presented in the Harvard format.

All units used in this project will be SI units. Decimal numbers will be noted with a dot while thousand separators will use comma.

Chapter Structure

Chapter 1 introduces the project and the reason for doing it.
Chapter 2 further investigate the back ground for the project.
Chapter 3 states the overall problem statement for the project with corresponding objectives.
Chapter 4 presents the proposed method for modelling the falling film pillow plate evaporator and the used equations.

Chapter 5 examines the validity of the model by comparing it to data for designed pillow plate evaporators.
Chapter 6 examines the effect that different input parameters for the model have on the overall heat transfer of the heat exchanger.
Chapter 7 discusses the model which has been made in the project and the results it have showed.
Chapter 8 gives an conclusion to the problem statement made in Chapter 3 and the objectives.
Chapter 9 examines the potential future work that can be done on this project to improve the results.

Aalborg University, May 27, 2021

Signatures of project participants

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Nomenclature

Symbols	Explanation	Unit
A	Area	m^2
a	Dimensionless parameter for pillow plate	-
a_{Vn}	Volume fitting parameter	-
a_{wn}	Wet area fitting parameter	-
b	Dimensionless parameter for pillow plate	-
c	Dimensionless parameter for pillow plate	-
C	Dimensionless length parameter	-
C_F	Liquid property parameter	-
C_∞	Falling film coefficient	-
c_p	Specific heat	J/(kg K)
d	Diameter	m
E	Friedel equation factor	-
f	Fitting factor	-
F	Factor for parameter	-
g	Gravitational acceleration	m/s^2
H	Height	m
H_F	Friedel equation factor	-
h_V	Specific enthalpy	J/kg
h	Convective heat transfer coefficient	$W/(m^2\ K)$
k	Thermal conductivity	W/(m K)
K	Sub cooled parameter	-
L	Length	m
m	Mass	kg
n	Fitting factor	-
NTU	Number of transfer units	-
Nu	Nusselt number	-
Nr	Number of elements	-
P	Pressure	Pa
Pr	Prandtl number	-
Q	Heat energy	J
q	Heat flux	W/m^2
$\dot{Q}$	Heat energy rate	W

Re	Reynolds number	-
R	Thermal resistance	K/W
R_1	Capacity rate	-
R_a	Arithmetic mean roughness height	μm
s	Spacing	m
$s^\star$	Dimensionless number for Reynolds correction	-
S	Slip factor	-
T	Temperature	K
u	Velocity	m/s
V	Volume	m^3
w	Width	m
W	Capacity rate	W/K
x	Quality of mixture	-
Y	Multiplication factor	-
α	Void fraction	-
δ	Distance between points	m
μ	Dynamic viscosity	Pa s
ν	Kinematic viscosity	m^2/s
ψ	Two zone parameter	-
ϕ_A	Welding area parameter	-
ϕ^2	Pressure loss multiplier	-
ρ	Density	kg/m^3
σ	Surface tension	N/m
θ	Angle	rad
ζ_f	Fanning friction factor	-
$\zeta_{\Delta P}$	Darcy friction factor	-

Subscripts	
A	Area
acc	Acceleration
c	Characteristic
cs	Cross sectional
cr	Critical
conv	Convection
film	Falling film
g	Gas
go	Gas only
grav	Gravitational
h	Hydraulic
hom	Homogeneous

i	Inner
L	Longitudinal
l	Liquid
lo	Liquid only
lam	Laminar
mid	Middle
nuc	Nucleate
out	Outer
PE	Periodic element
PP	Pillow plate
Q	Heat
r	Reduced
ref	Refrigerant
sp	Welding spot
T	Transverse
tot	Total
tp	Two phase
tran	Transitional
turb	Turbulent
w	Wall
W	Wet
z1	Zone one
0	Reference

Abbreviations	
PPHE	Pillow plate heat exchanger
COP	Coefficient of performance

Chapter 1

Introduction

Denmark is a large user of district heating with 64.5% of homes using it [State of Green, 2018]. Currently most of this heat comes from combined heat and power plants, with some coming from dedicated heat plants. These plants will usually run on either gas, coal, oil or biomass. However, most of these fuels produce CO_2 which causes global warming. Biomass does not have that issue, but there is a limited supply, so therefore other options for heat production need to be considered. One of the more promising options is to utilise waste heat that inevitable will be produced by the industry. Currently the estimated potential of the waste heat is 12.5 PJ which is enough to provide heat for 128,000 houses [Jørgensen, 2018]. The are however some issues with utilising this heat, because while some of it might be at suitable high temperatures of about 60 °C with noticeable mentions being Aalborg Portland and Skjern Papir, most of the potential is at temperatures around 20 to 55 °C [Fjernvarme, 2017].

This temperature is usually too low to be utilised directly in the district heating network. Therefore the usual solution is to utilise the waste heat as a heat source for a heat pump, which can then produce high temperature heat[Fjernvarme, 2017]. This is well known technology that is already widely used and is only planned to be further implemented in the danish district heating network. One of the core components of a heat pump is the evaporator, which transfers heat from the heat source to evaporate the refrigerant. There are several different types of evaporators, the difference mostly depending on the available heat source and the capacity. A promising solution in regards to utilising waste heat in the form of warm water is a pillow plate heat exchanger(PPHE). The PPHE is a promising solution as it has a high heat transfer to volume ratio and is easy to clean which is important as the waste heat water often will include particles. However, this is a new technology which has not been fully developed yet and there is a lack of understanding of the exact capacity of a given design and the optimal design parameters. One of the few suppliers of this type of heat exchanger is ATHCO. ATHCO has been producing heat exchangers since 1947, and have produced many PPHE. However, their knowledge in regards to how their design can be improved is limited. Therefore the goal of this project is to investigate and model a PPHE used as a evaporator. [ATHCO Enginering, 2017]

Chapter 2

Problem analysis

To be able to model the PPHE, different parts of the system needs to be examined. This means examining what type of waste heat is available, how does the PPHE fit into the overall heat recovery and the principle for how a PPHE works.

2.1 Waste heat

When considering how to utilise waste heat, the first step is to investigate what kind of waste heat is available. There are a few things that characterise waste heat, those being the temperature, amount, fluid and amount of pollution in the fluid. An example of waste heat could be the waste heat that is produced at Skjern Paper. Here the waste heat comes in two forms where one part is from the air that is used to dry the paper, while the other part is the heat in flue gas from the furnace. Both of these heat sources will have air, mixed with some moisture, be the heat source at a relatively high temperature. [Skjern Paper, 2019].

Another example of waste heat could be the heat is available from data centres. Data centres produces waste heat through the cooling of the servers. This waste heat will generally be on liquid form, either in the form of water or another cooling fluid. Furthermore it will generally be a lower temperature compared to high energy waste as what is seen at Skjern paper. This changes how the waste heat can be utilised, as this temperature is not high enough to be utilised directly to heat up district heating water. However, it has the advantage that it is easier to utilise the heat in a liquid as the energy density is higher. This project will mainly concern the usage of waste heat in liquid form with low temperatures. [Fjernvarme, 2017]

2.2 Heat Recovery

Now that the available waste heat have been investigated there needs to be made considerations for how the heat can be recovered. One of the most common method for utilising waste heat is through the use of heat pumps. Heat pumps are a technology that utilises electric power to move energy from a low temperature to a high temperature. A heat pump will consist of four main components, those being an evaporator, a compressor, a condenser and an expansion valve. These four components will be connected as seen in Figure 2.1.

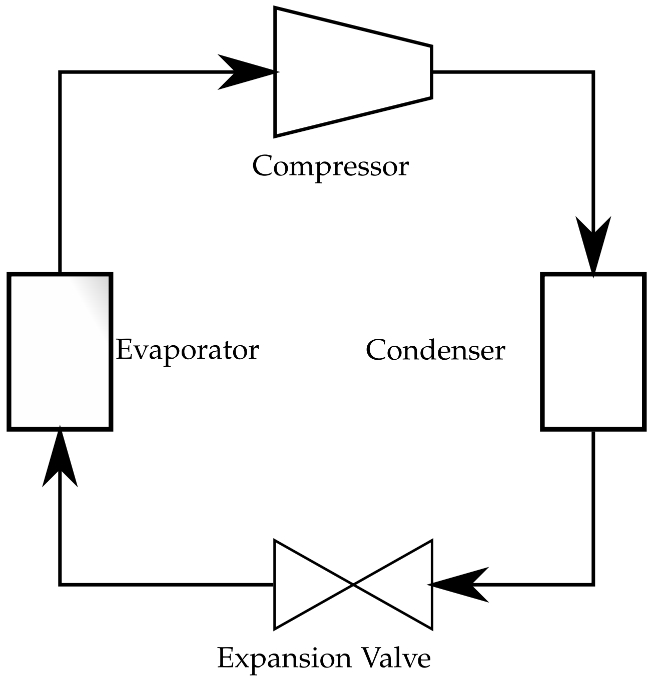

Figure 2.1: Overview diagram of the components of a heat pump.

The fluid before the evaporator in Figure 2.1 will be a mixture of gas and liquid. It will then enter the evaporator whereafter it will be completely on gas form. It will then enter the compressor where the pressure and temperature will be increased resulting in a super heated gas. This will then be cooled down and condensed in the condenser. Thereafter the pressure will be reduced in the expansion valve, which also results in some evaporation, thereby returning to the starting point forming a loop that moves heat from the evaporator to the condenser[Turner et al., 2012]. This process can be represented by a log P-h diagram as seen in Figure 2.2.

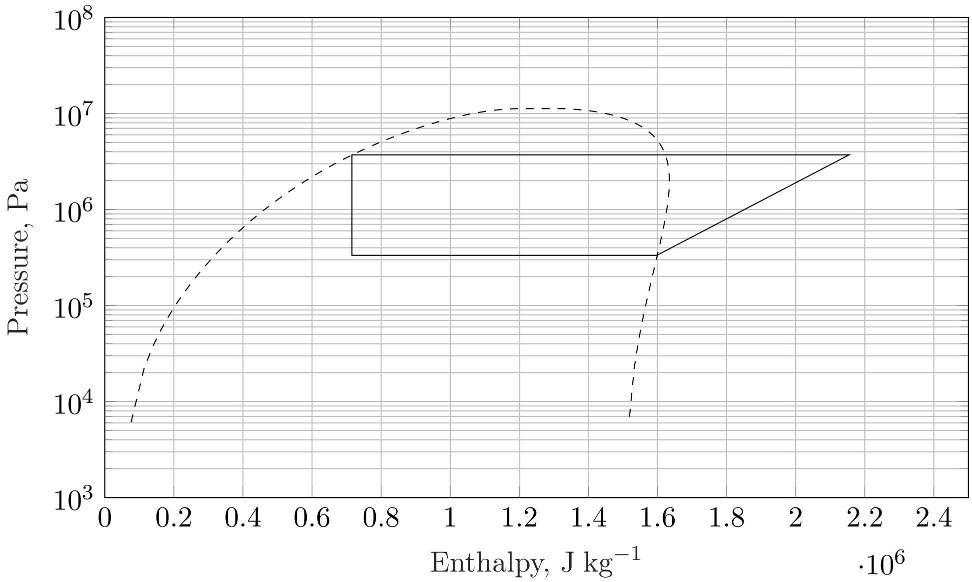

Figure 2.2: log P-h curve of Ammonia, with a heat pump cycle marked for a given evaporation and condensation pressure.

The important things to read from this diagram is the length of the two horizontal lines as the bottom one represents the heat absorbed in the evaporator while the top one represent the heat released from the condenser. When the difference in length between these two are small the heat pump will have a high coefficient of performance (COP). Another thing to note is the dotted line which shows the region where the refrigerant will be a vapour/liquid mixture. This line is fully dependent on the chosen refrigerant, in this case ammonia. Therefore the refrigerant is very important in regards to the amount of heat the refrigerant can contain and at which temperatures it can operate. Ammonia is one of the more commonly used refrigerants for large scale heat pumps as it have a high heat potential and can function at high temperatures, however ammonia is also toxic and moderately flammable. Therefore ammonia has to be handled carefully. An alternative to ammonia that is also used quite a lot is carbon-dioxide (CO_2). This is non toxic and has a similar performance to ammonia, but it has to work under high pressure which makes it more difficult to utilise for large scale heat pumps. Besides these two refrigerant which is classified as natural refrigerant, there also exist synthetic refrigerant an example being R-404a. These all also have there advantages and disadvantages that needs to be considered. [David et al., 2017]

This project will not be limited to a single refrigerant, so several refrigerants will be examined. Besides the refrigerant there are specific components that needs to be considered. Generally all these components are well researched so there is plenty of knowledge available in regards to designing these. However, as the heat source that will be utilised is waste heat from industrial production there needs to be made special consideration for the evaporator. The reason that the evaporator needs special consideration is that the waste heat can contain particles and there are therefore special requirements regarding how cleanable it is. A potential solution for this is a PPHE.

2.3 Pillow Plate Heat Exchanger

Pillow plates are heat exchangers that are made by welding a spot pattern on two plates whereafter the edges are sealed. Then a fluid is inserted between the plates at high pressure so the distance between the two plates expand, resulting in a pillow plate as seen in Figure 2.3.

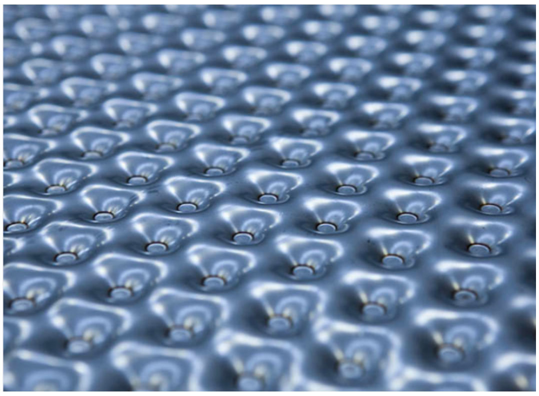

Figure 2.3: Pillow plate that have been welded and expanded[ATHCO Enginering, 2017].

From the figure it can be seen how the plate consist of the welding spots and the expanded plate. The advantage of having these welding spots is that it will induce turbulence in the flow within the plate, which enhances heat transfer. While this also increases the pressure loss, it is overall seen as a better solution.

Pillow plates is a not a commonly used type of heat exchanger as they generally are more expensive that normal plate and fin tube heat exchangers. One of the companies that produce this type of pillow plates are ATCHO Engineering. ATCHO is one of the few manufactures of pillow plates and have been doing this since 1947. Therefore they are very experienced in design solutions for customers. However, most of their design knowledge is based on either experiments or old design equations. This means that the design of the pillow plates have not evolved and improved much during recent years which is needed to obtain a larger market share for pillow plates. This problem is mostly in regards to designing pillow plates as evaporators. Therefore a model to estimate the performance of a given design will improve the design process, which will make pillow plates a better option for heat recovery. [ATHCO Enginering, 2017]

Chapter 3
Problem Statement

Following the problems discussed in the previous chapter there can be made a problem statement which the project will aim to solve.

3.1 Problem Statement

Create a parametric model of the heat transfer within a falling film pillow plate evaporator.

3.2 Objectives

To be able to fulfil the problem statement the following objectives needs to be resolved.

- Determine the thermal resistance of the heat transfer.
- Investigate how the evaporation process enhances the heat transfer.
- Investigate the accuracy of the proposed model.
- Determine the important parameters for the performance of a pillow plate evaporator.

Chapter 4

Modelling

In this chapter the methodology for modelling the PPHE will be examined. This will consist of modelling the convective heat transfer from the falling film, the conduction through the plate and the mixed convection and boiling heat transfer from the pillow plate to the refrigerant. This will also include accounting for how the two fluids will flow along or inside the heat exchanger and how they interact. There will also be accounted for pressure losses through the heat exchanger, while accounting for the changing quality of the refrigerant through the pillow plate.

4.1 System description

The first step in regards to modelling the PPHE is to understand how it functions. The purpose of the PPHE is to evaporate the refrigerant by transferring heat from the heat source. This is achieved by having the refrigerant flow within the plate while the water will flow along the plate. Then as there are a temperature difference between the refrigerant and the water, there will be transferred heat through the plate. To be able to model this heat transfer it is important to consider how the refrigerant flow within the plate and how the water flows on the outside of the plate. Firstly the flow within the plate needs to be understood. All heat exchanger consist on an inlet and an outlet where the area between is used to exchange heat. In the case of pillow plate heat exchangers the will usually be configured with the inlet in the bottom and the outlet in the top on the same side of the plate. An example of this can be seen in Figure 4.1.

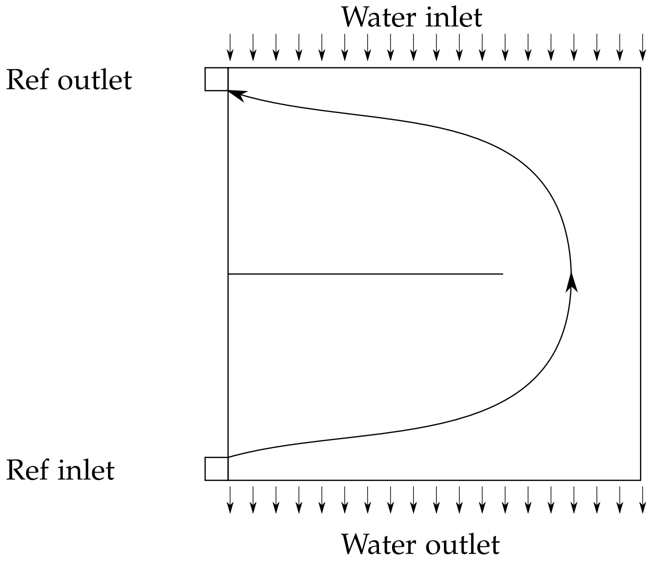

Figure 4.1: Flow of refrigerant and falling film from inlet to outlet.

This can potentially create issues where a large amount of the area of the heat exchanger is not used so for application where there is not a significant difference between the height and width there can be installed baffles to ensure that the full area of the heat exchanger is utilised as seen in Figure 4.1. The placement of these baffles will have a noticeable effect on the flow and are therefore important to consider when modelling the flow. However, a baffle is not always used and sometimes there is also added several smaller baffles on a line instead of a bigger baffle depending on the application.

Other than the flow of the refrigerant there is also the water flow that needs to be considered. The water flow will flow down the plate in a thin layer which is called falling film. This film is made by having a distribution system placed above the plate. Then based on how this distribution system is made and the flow rate of the water, there will form a film of a certain thickness. This will also depend on the design of the plate as the surface geometry will influence how the film layer develops. An example of this liquid film and how it flows compared to the refrigerant can be seen in Figure 4.2 when a baffle is present.

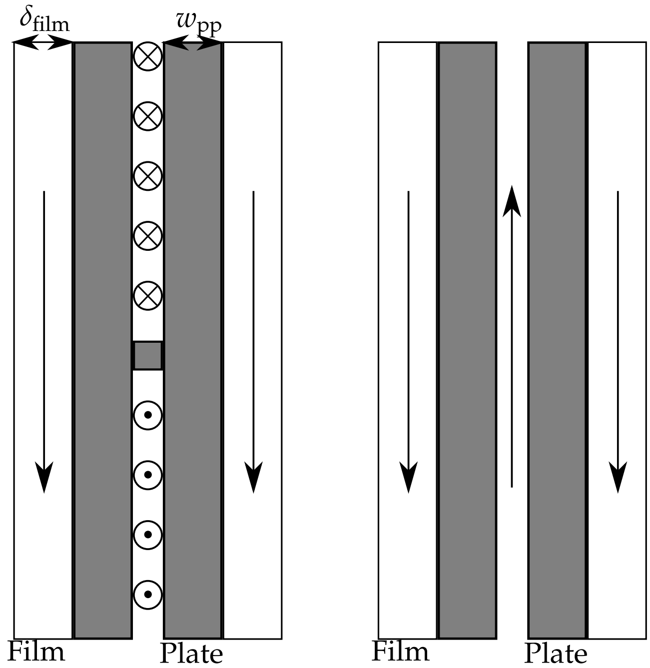

Figure 4.2: Cross section of falling film and flow within plate, with and witout a baffle.

From the figure it can be seen how the water flows down along the plate as a thin layer. It can also be seen that there are two different flow arrangements between the falling film and the refrigerant. Either it will be cross flow as what is seen in the left figure which represent the area where there is an baffle or it will be counter flow as what is seen in the right figure. It should be noted that this is simplified to a flat plate as the pillow plate geometry will result in more movement perpendicular to the flow direction for both the film flow and the flow within the pillow plate.

4.1.1 Pillow Plate Geometry

Besides understanding the flow arrangement it is also important to understand how a given geometry of a PPHE is defined, and which parameters of those that needs to be accounted for in the modelling process. In regards to the pillow plate the important parameters can be seen in Figure 4.3.

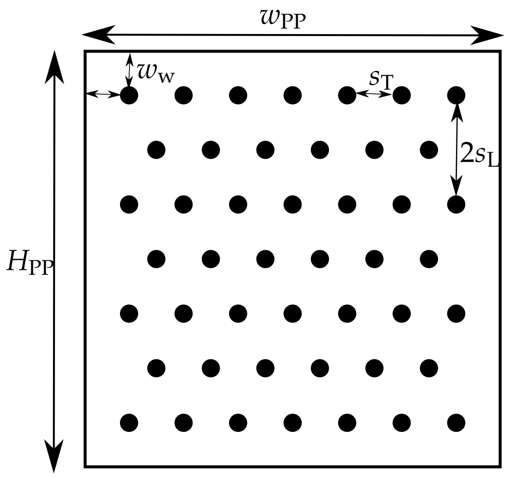

Figure 4.3: Overview of important parameters for the pillow plate geometry.

From Figure 4.3 it can be seen that firstly there is the overall size of the plate which is defined by the height, H_{PP} and width of the plate, w_{PP}. Besides that there is the offset to the first welding spot row in both directions, w_W. Then there is the spacing of the welding spots, which is defined by the longitudinal distance, S_L and the transversal distance, S_T. Then there is also the diameter of the welding spots, d_{sp} and the height of the inner channel, δ_i which can be seen in Figure 4.4.

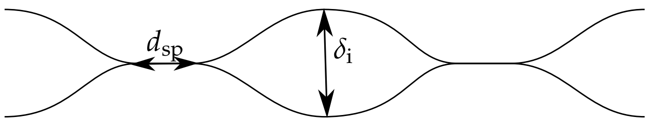

Figure 4.4: Definition of inner channel height for a pillow plate.

Lastly for the geometry there is parameters which describe the overall structure such as the number of plats and the distance between them. From these geometric parameters it is important to be able to calculate the cross sectional area, hydraulic diameter, volume and surface area. Due to the complexity of the geometry there have been made correlations for these, based on the geometric parameter in a study by Piper et al. [2015]. Here they utilised formation simulations to create relation to that can calculate the cross sectional area and the other useful areas, lengths and volumes. All these simulations were made for the minimum repeating area which can be seen in Figure 4.5.

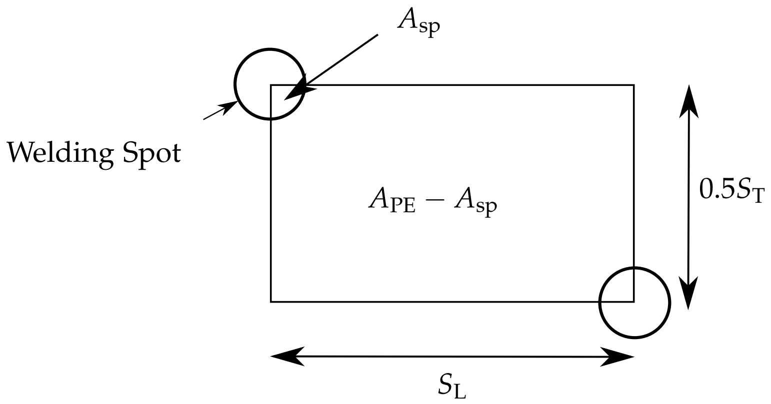

Figure 4.5: Overview of important parameters for the pillow plate geometry.

The part of the pillow plate which can be seen in Figure 4.5 is the part where all the important parameters will be calculated for which can then be extended to the whole pillow plate. Firstly there is the volume of this part of the pillow plate which can be calculated from eq. 4.1.

$$V_i = a_{Vn,ref} \cdot \delta_i \cdot s_D^2 \cdot f_{sp} \tag{4.1}$$

Here S_D [m] is the diagonal distance between the two welding spots, $a_{Vn,ref}$ is a parameter that was fitted to the geometric parameters based on the simulations and can be calculated as eq. 4.2

$$a_{Vn,ref} = 0.1 \cdot \left(\frac{s_T}{2 \cdot s_L}\right)^2 - 0.18 \left(\frac{s_T}{2 \cdot s_L}\right) + 0.19 \tag{4.2}$$

Then the f_{sp} factor in eq. 4.1 is to account for the effect the welding spots have and can be calculated as eq. 4.3.

$$f_{sp} = 1.37 \cdot \phi_A^{2.58} \tag{4.3}$$

Where ϕ_A represent the influence of the welding spot diameter can can be calculated as eq. 4.4.

$$\phi_A = 1 - \frac{\pi \cdot d_{sp}^2}{4 \cdot s_T \cdot s_L} \tag{4.4}$$

Besides the volume which can be calculated with the previous equations it is also important to be able to calculate the inner surface area of the pillow plate which can be done by eq. 4.5.

$$A_{w,i} = A_0 + A_0 \cdot a_{wn,ref} \cdot \frac{\delta_i^2}{s_D^2} \tag{4.5}$$

Where A_0 [m^2] is the area of the flat square in Figure 4.5 and $a_{\text{wn,ref}}$ is a fitting factor which can be calculated as eq. 4.6.

$$a_{\text{wn,ref}} = 3.12 \cdot \left(\frac{s_{\text{T}}}{2 \cdot s_{\text{L}}}\right)^2 - 5.74 \left(\frac{s_{\text{T}}}{2 \cdot s_{\text{L}}}\right) + 3.08 \tag{4.6}$$

The next parameter which needs to be defined is the hydraulic diameter. This is normally defined as the cross sectional area divided with the perimeter, however for a pillow plate these parameters vary with the length so it will be integrated along the length resulting in the hydraulic diameter being calculated as eq. 4.7.

$$d_{\text{hi}} = \frac{4 \cdot V_{\text{i}}}{A_{\text{w,i}}} \tag{4.7}$$

Lastly there is the average cross sectional area of the small piece which can be calculated as eq. 4.8.

$$A_{\text{cs,i}} = \frac{V_i}{s_{\text{L}}} \tag{4.8}$$

This can then be extended to give the total cross sectional area which can be calculated as eq. 4.9.

$$A_{\text{cs,tot}} = A_{\text{cs,i}} \cdot 4 \cdot \left(\frac{w_{PP} - 2 \cdot w_{\text{w}}}{s_T}\right) \tag{4.9}$$

By using the presented equations it is possible to calculate all the relevant geometric parameters of the pillow plate for the input parameters. Thereby all the necessary parameters for determining the heat transfer can be calculated for a given pillow plate geometry which will be done in the following section, with the other forms of heat transfer.

4.2 Heat Transfer

The first step in modelling the heat transfer, after understanding the geometry, is to understand which type of heat transfer there will be present during the process. In this case there are three separate forms of heat transfer present. Firstly there will be convection from the water film to the plate, then the heat will travel through the plate by conduction, whereafter the heat will be transferred to the refrigerant though a mixture of boiling and convection. This is modelled by calculating a sum of serial connected thermal resistances. This means that the local heat transfer can be formulated as eq. 4.10.

$$\dot{Q} = \frac{\Delta T}{R} \tag{4.10}$$

Where ΔT [K] is the temperature difference between the two media and R [W/K] is the thermal resistance. The thermal resistance in this case is defined as eq. 4.11.

$$R = \frac{1}{h_{PP} \cdot A_i} + \frac{\delta_w}{k_{\text{PP}} \cdot A_{\text{mid}}} + \frac{1}{h_{film} \cdot A_{out}} \tag{4.11}$$

Where h_{PP} [W/(m^2 K)] is the boiling heat transfer coefficient between the refrigerant and the pillow plate, A_{inner} [m^2] is the inner area of the pillow plate, δ_w [m] is the thickness of the pillow plate, k_{PP} [W/(m K)] is the thermal conductivity of the plate, h_{film} [W/(m^2 K)] is the convective heat transfer coefficient between the falling film and the pillow plate and A_{outer} [m^2] is the outer area of the pillow plate. All these areas are assumed to be same as the area as a flat plate as the added area from the inflation is about 2-7% with the welding spot also making up 3-10% of the surface area. From this it can be seen that the area added from the inflation is similar to the area added with the inflation, so the error involved in assuming the area to be similar to a flat plate is minimal. [Piper et al., 2015]

To utilise this equation for determining the heat transfer in any part of the pillow plate, the different parts of the thermal resistances needs to be modelled. The conduction through the plate can be estimated by using the thermal conductivity of the given material. However, the two convection terms are more complicated and will therefore be based on Nusselt relations developed in studies made on the subject. Therefore there needs to be found a fitting relation for all the different forms of heat transfer starting with the heat transfer from the falling film.

4.2.1 Falling Film Heat Transfer

The Nusselt relation for the falling film heat transfer will be examined as described by Schnabel [2010] where the first step is to define the Reynolds number which is done using eq. 4.12. This Reynolds number is calculated from mass flow rather than the velocity as the thickness and thereby velocity is very dependent on the Reynolds number.

$$Re_{\text{film}} = \frac{\dot{m}}{\mu \cdot 2 \cdot W_{PP}} \tag{4.12}$$

Where $\dot{m}$ [kg/s] is the mass flow of the fluid forming the film and μ [Pa s] is the dynamic viscosity. The Nusselt number is very similar to the normal Nusselt number equation where the characteristic length is replaced by a relation between the kinematic viscosity and the gravitational acceleration as seen in eq. 4.13.

$$Nu_{\text{film}} = \frac{h_{\text{film}}}{k_{film}} \cdot \left(\frac{\nu^2}{g}\right)^{1/3} \tag{4.13}$$

Where g [m/s^2] is the gravitational acceleration which is defined as $9.81\,\mathrm{m/s^2}$ and ν[m^2/s] is the kinematic viscosity. The Nusselt number for the falling film is dependent on the degree of turbulence. It will either be fully laminar, in the transition to turbulent or be fully turbulent. Therefore there exist different methods for calculating the Nusselt number in the different situations. The Nusselt relation for the laminar flow is defined as eq.4.14.

$$Nu_{\mathrm{film,lam}} = C_\infty \cdot \mathrm{Re}_{\mathrm{film}}^{-1/3} \tag{4.14}$$

Where C_∞ is parameter that depends on whether there is a constant wall temperature or a constant heat flux and Pr is the Prandtl number. For the case of a constant temperature the value will be 1.3 and for constant heat flux it will be 1.43.

The Nusselt relation for transitional flow is defined as eq. 4.15

$$Nu_{\mathrm{film,tran}} = 0.0425 \cdot \mathrm{Re}_{\mathrm{film}}^{0.2} \cdot \mathrm{Pr}^{0.344} \tag{4.15}$$

The Nusselt relation for fully turbulent flow is defined as eq. 4.16

$$Nu_{\mathrm{film,turb}} = 0.0136 \cdot \mathrm{Re}_{\mathrm{film}}^{0.4} \cdot \mathrm{Pr}^{0.344} \tag{4.16}$$

In regards to determining which of the equation that should be used it depends on the fluid, whether it is fixed heat flux or temperature, but the most important parameter is the Reynolds number. With water as the fluid and a fixed wall temperature eq. 4.14 is valid up to a Reynolds number of 170 whereafter eq. 4.15 is used until the Reynolds number is about 300, after which the flow is turbulent and eq. 4.16 is used. It should be noted that this method is based on a vertical pipe that does not have the unevenness that are present for the pillow plate. Therefore it is expected that there will be some deviation from the actual heat transfer of the falling film.

4.2.2 One Phase Pillow Plate Heat Transfer

After the heat from the falling film is transferred to the plate it will be transferred to the refrigerant. The basis for modelling the heat transfer within the pillow plate is to understand the one phase heat transfer. This heat transfer will be based on a study by M.Piper et al. [2017]. The main basis for this heat transfer correlation is understanding the flow within the pillow plate. Therefore M.Piper et al. [2017] had previously made Computional Fluid Dynamic (CFD) simulation of the flow. The result of the simulations can be seen in Figure 4.6.

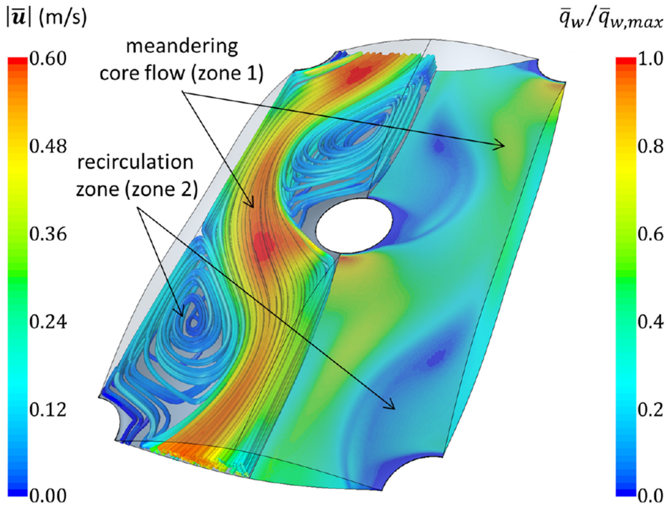

Figure 4.6: Flow within the pillow plate for the liquid phase based on CFD results [M.Piper et al., 2017].

From the figure it can be seen that there mainly are two flow zones within the plate. Those zones are the meandering core flow and the recirculating zone. The meandering core flow is the part that is moving the fluid through the pillow plate, while the recirculating zone is whirling between the core flow and the welding spots. This is also linked to the heat transfer as the areas that have the highest heat flux is within the meandering core flow. Therefore the modelling of the heat transfer is based on how this core flow is for a given set of pillow plate parameters.

This is done by calculating the properties for the core flow and then extending them to also cover the recirculation zones. Firstly the core flow is seen to be similar to the flow within a pipe and therefore it is possible to use the well examined correlations for pipe flow developed by Petukhov [1963]. The correlation for the Nusselt number is given in eq. 4.17 and is valid for Reynolds from 1,000 to 8,000 and Prandtl numbers from 1 to 150.

$$Nu = \frac{(\zeta_f/8) \cdot \mathrm{Re}_{z1} \cdot \mathrm{Pr}}{1.07 + 12.7\sqrt{(\zeta_f/8)} \cdot (Pr^{2/3} - 1)} \tag{4.17}$$

Where Re_{z1} is the Reynolds number for the core flow. This equation can then be used to model the heat transfer from the core flow. To able to utilise the equation it is necessary to calculate the fanning friction factor ζ_f. This is done through a power law equation which is defined as eq. 4.18.

$$\zeta_f = n_6 \cdot \mathrm{Re}^{n_7} \tag{4.18}$$

Where n_6 and n_7 are coefficient that depend on the pillow plate geometry. Besides the heat transfer from the core flow, there will also be heat transfer from the recirculation zones, therefore they also need to be accounted for. This was done by calculating two correlation factors ψ_A and ψ_Q which corresponds to dimensionless variables that describe the size and contribution to heat transfer from the recirculating zones respectively. This can then be used to calculate the convective heat transfer coefficient for the entire flow using eq. 4.19.

$$h_{\text{tot}} = h_{z1} \cdot \left(\frac{1-\psi_A}{1-\psi_Q} \right) \tag{4.19}$$

Where h_{z1} [W/(m^2 K)] is the convection heat transfer of the core flow. These coefficient depend on the geometry of the plate, especially the placement of the welding spots. Other factors that are dependent on the geometry are the hydraulic diameter of the core flow and a coefficient $s^\star$ which correlates the total Reynolds number to the Reynolds number of the core flow. All these coefficient has been modelled by M.Piper et al. [2017] based on three dimensionless parameters a, b and c which is defined in eq. 4.20, 4.21 and 4.22.

$$a = \frac{2 \cdot s_L}{s_T} \tag{4.20}$$

$$b = \frac{d_{\text{sp}}}{s_T} \tag{4.21}$$

$$c = \frac{\delta_i}{s_T} \tag{4.22}$$

With these coefficient there was made a table to calculate the different coefficients which can be seen in Table 4.1.

Table 4.1: Table with coefficient for the important parameters for three type of pillow plates.

	$a \approx 0.58$ $0.1 < b < 0.14$ $0.042 < c < 0.083$	$a \approx 1$ $0.17 < b < 0.24$ $0.071 < c < 0.143$	$a \approx 1.71$ $0.17 < b < 0.24$ $0.071 < c < 0.143$
n_6	$4.36c + 1.14$	$2.52c + 0.24$	$4.62c + 0.6$
n_7	-0.44	-0.3	-0.34
ψ_A	$0.94b + 0.4$	$0.75b + 0.46$	$0.81b + 0.263$
ψ_Q	$2.16b + (4.23c - 0.352)$	$0.75b + (1.54c - 0.014)$	$0.46b + (1.17c - 0.042)$
$d_{h,z1}$	$-11.22b + (113c + 1.82)$	$-18.31b + (35.42c + 4.8)$	$-8.1b + (60c - 2.1)$
$s^\star$	1	1	1.0761

As long as the examined pillow plate is within the given limits the study showed that it will have an accuracy of $\pm 15\%$. Thereby the convection part of the heat transfer can be modelled.

4.2.3 Boiling Heat Transfer

Boiling Flow Regimes

To be able to model the heat transfer during the two phase boiling process it is first necessary to examine how the flow changes due to the flow being a two phase flow. However, the research on two phase flow within a pillow plate has not yet been examined in depth. Therefore, the flow within pipes will be examined instead as it is a well researched subject and it has earlier been shown in the one phase flow that the flow within a pillow plate is not that different. The different flow regimes in a vertical pipe can be seen in Figure 4.7.

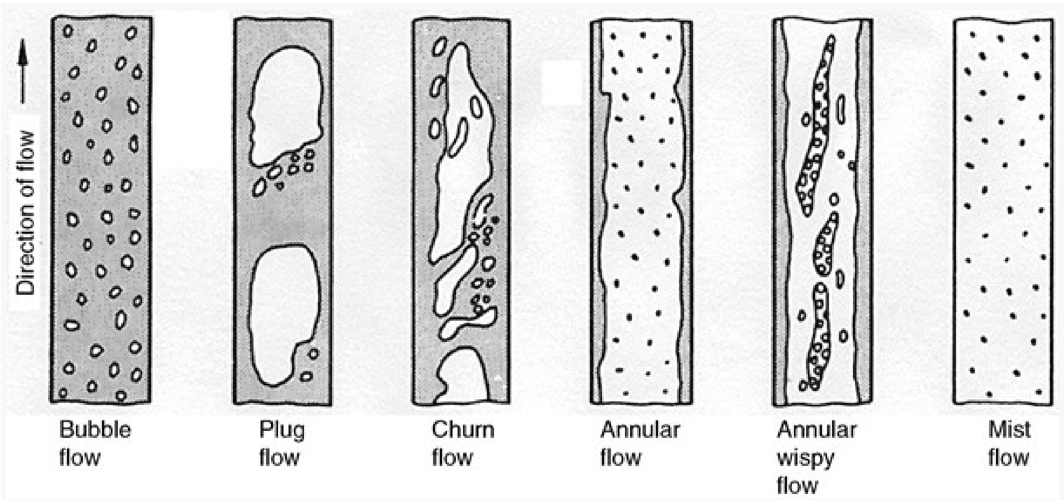

Figure 4.7: Different flow regimes of boiling within a vertical tube [Kind et al., 2010].

From the figure it can seen that there are many different types of flows that are mostly dependent on the quality of the flow. At the start there will be a very low quality in the flow and there will only be a few bubbles which is called the bubble flow. As the quality increase these bubbles will start to lump together eventually forming the plug flow. Then when the plug flow becomes unstable it will result in churn flow which eventually results in annular flow where the edges are fully wetted while there is a gas flow in the middle with some droplets. Eventually as more of the flow is evaporated the edges will no longer be wetted and the flow will become mist flow. Then when all the droplets in the flow has been evaporated it again becomes a fully one phase flow this time just as an gas.

Now that the different flow regimes has been examined it is important to understand how these different flows affect the heat transfer. Firstly the two heat transfer mechanics for boiling needs to be understood. Heat transfer from flow boiling will partly be from convection and partly from nucleate boiling. In Figure 4.8 it can be seen how the wall temperature changes due to the different flow regimes and which heat transfer method is dominant for the given flow regime.

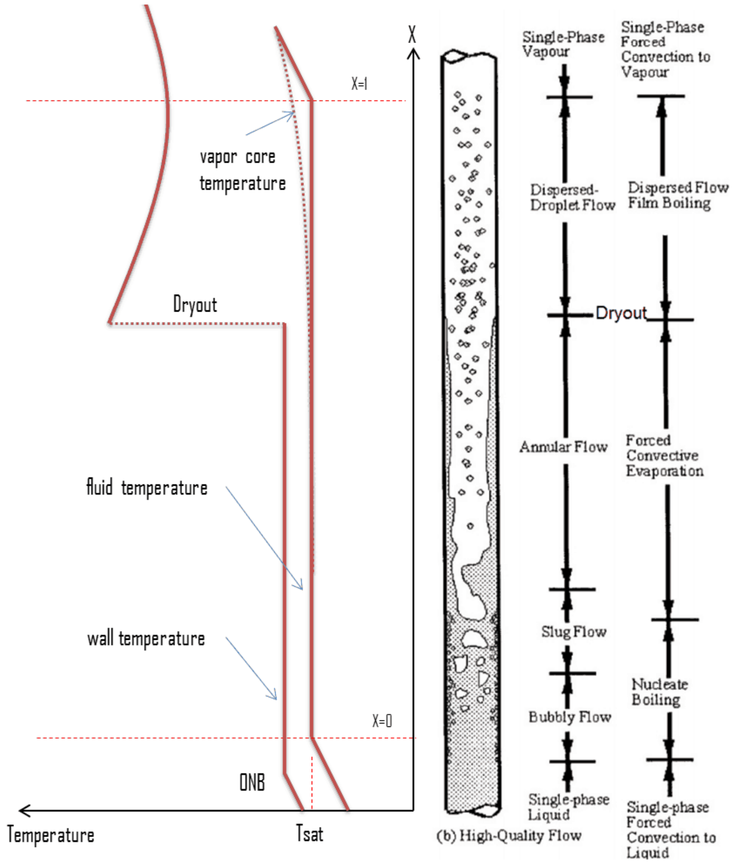

Figure 4.8: Flow regimes of boiling in vertical pipe correlated to the wall temperature [nuclear power.net, 2021].

From Figure 4.8 it can be seen that initially from the bubbly flow to the end of the annular flow, where the dry out happens, that the wall temperature is constant and close to the fluid temperature. This mean that the heat transfer is also more or less constant and is also high. It can also be seen that starting from the bubbly flow and for parts of the slug flow it will be heat transfer from nucleate boiling that is the main factor and afterwards it will be from convection as the quality increases and the flow becomes annular. Then when the annular phase ends and the mist flow begins it can be seen that there is a noticeable increase in the wall temperature. This is a phenomena called dry out which is due to the walls no longer being wet. This significantly reduced the heat transfer as the heat transfer from the wall now can not be used to directly evaporate the fluid, but instead has to transfer the heat to the gas which then can evaporate the remaining droplets. It can lastly be seen that the wall temperature slowly is reduced a bit again which is due to more of the fluid evaporating which increases the velocity of the fluid. From this it can be seen that there are two situation for which the heat transfer needs to be modelled which is the mixture of nucleate and convection boiling before the dry out and the heat transfer within the mist flow after the dry out. The first step is however to establish a method to determine where the dry out happens which will be done with the critical heat flux.

Critical Boiling

This critical heat flux is very dependent on the given fluid and the geometry of the pipe. The method which will be utilised in the project is only valid for subcooled or saturated flows in vertical pipes which is applicable as it is a flooded evaporator that is utilised for the pillow plate. However, it is expected that there will be a noticeable difference between the wetting observed in a tube compared to a pillow plate so the critical heat flux might not necessarily be the same, due to the turbulence created by the welding spots. However, as there is not enough research made for pillow plates the theory for vertical pipes will be examined. This method is described by Kind et al. [2010] where the method is dependent on three dimensionless numbers which describe the geometry of the pipe, density of the liquid and the surface tension. The dimensionless number for the density is calculated as eq. 4.23.

$$\rho^\star = \frac{\rho_g}{\rho_l} \tag{4.23}$$

Where ρ_g[kg/m^3] is the density on gas form and ρ_l[kg/m^3] is the density in liquid form. The pipe geometry is described by a dimensionless length which is calculated as eq. 4.24.

$$l^\star = \frac{l}{d_\mathrm{h}} \tag{4.24}$$

Where l [m] is the distance from inlet to calculation point. Then finally there is the dimensionless surface tension which is defined as eq. 4.25.

$$\sigma^\star = \frac{\sigma \cdot \rho_g}{G \cdot l} \tag{4.25}$$

Where G [kg/m^2] is the mass flux and σ [N/m] is the surface tension. All of these dimensionless numbers are important in regards to calculating the dimensionless heat flux which is calculated as eq. 4.26.

$$q_1^\star = \frac{q_\mathrm{cr,o}}{\dot{m} \cdot \Delta h_\mathrm{V}} \tag{4.26}$$

Where $q_\mathrm{cr,o}$ [W/m^2] is the critical heat flux and h_V [J/kg] is the latent heat. There are five different equations for calculating this which depend on the dimensionless numbers. These equations are defined as eq. 4.27 to 4.31.

$$q_1^\star = C \cdot (\sigma^\star)^{0.043} \cdot \frac{1}{l^\star} \tag{4.27}$$

$$q_2^\star = 0.1 \cdot (\rho^\star)^{0.133} \cdot (\sigma^\star)^{1/3} \cdot \frac{1}{1 + 0.0031 \cdot l^\star} \tag{4.28}$$

$$q_3^\star = 0.098 \cdot (\rho^\star)^{0.133} \cdot (\sigma^\star)^{0.433} \cdot \frac{(l^\star)^{0.27}}{1 + 0.0031 \cdot l^\star} \tag{4.29}$$

$$q_4^\star = 0.234 \cdot (\rho^\star)^{0.513} \cdot (\sigma^\star)^{0.433} \cdot \frac{(l^\star)^{0.27}}{1 + 0.0031 \cdot l^\star} \tag{4.30}$$

$$q_5^\star = 0.0384 \cdot (\rho^\star)^{0.6} \cdot (\sigma^\star)^{0.173} \cdot \frac{(l^\star)^{0.27}}{1 + 0.28 \cdot (\sigma^\star)^{0.233} \cdot l^\star} \tag{4.31}$$

The C coefficient in the equations is dependent on the dimensionless length. If it is above 50 it will be equal to 0.25, while if it is above 150 it will be 0.34. If it is between it can be calculated as eq. 4.32 which is a linear interpolation between the values.

$$C = 0.25 + 9 \cdot 10^{-4} \cdot (l^\star - 50) \tag{4.32}$$

Now that all the different equations for calculating the dimensionless heat flux can be calculated the method to determine which equation to use will be examined. The first thing to examine is the size of the dimensionless density. If it is below 0.15 the following method will be utilised. Then the size of of the dimensionless heat flux be determined from the following decision tree.

$$\begin{aligned} q_1^\star \leq q_2^\star \longrightarrow & q^\star = q_1^\star \\ q_1^\star > q_2^\star \longrightarrow & q_2^\star \leq q_3^\star \longrightarrow q^\star = q_1^\star \\ & q_2^\star > q_3^\star \longrightarrow q^\star = q_3^\star \end{aligned} \tag{4.33}$$

From this it can be seen that the dimensionless heat flux can be calculated as eq. 4.27 if $q_1^\star$ is less than $q_2^\star$ or that $q_2^\star$ is less than $q_3^\star$. If either of these are not the case it can be calculated from eq. 4.29. If the dimensionless density is instead higher than 0.15 the dimensionless heat flux can be determined using the following methodology.

$$\begin{aligned} q_1^\star \leq q_4^\star \longrightarrow & q^\star = q_1^\star \\ q_1^\star > q_4^\star \longrightarrow & q_4^\star \geq q_5^\star \longrightarrow q^\star = q_4^\star \\ & q_4^\star < q_5^\star \longrightarrow q^\star = q_5^\star \end{aligned} \tag{4.34}$$

Thereby it is possible to calculate the critical heat flux. However the critical heat flux is higher if the fluid that is entering is sub-cooled, which can be accounted for using eq. 4.35

$$\dot{q}_{cr} = \dot{q}_{cr,n}(1 - K \cdot \dot{x}_{inl}) \tag{4.35}$$

Where $\dot{x}_{inl}$ is the quality of the fluid at the inlet and K is a coefficient to account for the amount of sub-cooling which can be calculated by eq. 4.36 to 4.38.

$$K_1 = \frac{1.043}{4 \cdot C \cdot (\sigma^\star)^{0.043}} \tag{4.36}$$

$$K_2 = \frac{5}{6} \cdot \frac{0.0124 + 1/l^\star}{(\rho^\star)^{0.13} \cdot (\sigma^\star)^{1/3}} \tag{4.37}$$

$$K_3 = 1.12 \cdot \frac{1.52 \cdot \sigma^\star)^{0.233} + 1/l^\star}{(\rho^\star)^{0.6} \cdot (\sigma^\star)^{0.173}} \tag{4.38}$$

Similar to the critical heat flux the method for determining which K equation to use is also dependent on whether the dimensionless density is above or below 0.15. If it is below then the highest value of eq. 4.36 and 4.37 will be chose while it will be eq. 4.36 if they are equal. If it is above 0.15 then it will again be eq. 4.36 if it is higher or equal to eq. 4.37, otherwise it will be the smallest number of eq. 4.37 and 4.38.

Now that sub-cooling is accounted for it can be examined how the critical heat flux influence where the dry out happens. If the heat flux from the wall is higher than the critical heat flux there will be film boiling from the start which is similar to the dry out. If that is not the case the dry out will happen when the critical quality is reached which can be calculated as eq. 4.39

$$\dot{x}_{cr} = \frac{4 \cdot \dot{q}_{cr}}{\dot{m} \cdot \Delta h_V} \cdot \frac{l}{d_\mathrm{h}} + \dot{x}_{inl} \tag{4.39}$$

This approach is very versatile and is valid for a wide range of applications. The only limitations are that the fluid has to be sub-cooled or saturated at the inlet, the diameter and length ratio must not be above 600 and the outlet quality needs to be above 0. Therefore this approach should be fully valid as long as the evaporator considered is flooded. It should be noted that this method is developed for a uniform heat flux, so the results obtained from this equation may have differences compared to the varying heat flux there is seen in a PPHE.

Nucleate and Convective Boiling

Before dry out the heat transfer will be from convection and nucleate boiling, so a equation for both types of heat transfers will be examined. Firstly, it will be examined how the convection part can be modelled whereafter the nucleate boiling and how these two types of heat transfer can be combined. Both convective and nucleate boiling will be based on the approach described by Kind et al. [2010].

The convective boiling can be calculated as an enhancement of the convective heat transfer in an one phase flow. The enhancement from the two phase flow has been developed for fully wet tubes where the flow is either horizontal or vertical. The enhancement of the heat transfer depends on the density of the two phases, the quality and the convective heat transfer for the two phases. For the case of vertical tubes the enhancement ratio can be calculated as eq. 4.40.

$$\begin{aligned}\frac{h_{conv}}{h_l} = &\left[(1-\dot{x})^{0.01} \cdot \left((1-\dot{x})^{1.5} + 1.9 \cdot \dot{x}^{0.6} \cdot \left(\frac{\rho_l}{\rho_g}\right)^{0.35}\right)^{-2.2} + \right. \\ &\left. \dot{x}^{0.01} \cdot \left(\frac{h_{go}}{h_{lo}} \cdot \left[1 + 8 \cdot (1-\dot{x})^{0.7} \cdot \left(\frac{\rho_l}{\rho_g}\right)^{0.67}\right]\right)^{-2}\right]^{-0.5}\end{aligned} \tag{4.40}$$

When the flow is horizontal, the flow is more uneven which decreases the heat transfer rate. Therefore the enhancement ratio can not be calculated using eq. 4.40 and is instead calculated using 4.41

$$\begin{aligned}\frac{h_{conv}}{h_l} = &\left[(1-\dot{x})^{0.01} \cdot \left((1-\dot{x})^{1.5} + 1.2 \cdot \dot{x}^{0.4} \cdot \left(\frac{\rho_l}{\rho_g}\right)^{0.37}\right)^{-2.2} + \right. \\ &\left. \dot{x}^{0.01} \cdot \left(\frac{h_{go}}{h_{lo}} \cdot \left[1 + 8 \cdot (1-\dot{x})^{0.7} \cdot \left(\frac{\rho_l}{\rho_g}\right)^{0.67}\right]\right)^{-2}\right]^{-0.5}\end{aligned} \tag{4.41}$$

By using one of the presented equations the convective heat transfer can be accounted for. Then the nucleate heat transfer needs to be calculated. The nucleate heat transfer is calculated by multiplying a reference value with some given factor. This can be formulated as eq. 4.42.

$$\frac{h(z)_{\text{nuc}}}{h_0} = C_F \cdot \left(\frac{\dot{q}}{\dot{q}_0}\right)^n F(p^\star) \cdot F(d) \cdot F(W) \cdot F(\dot{m}, \dot{x}) \tag{4.42}$$

Where C_F is a factor dependent on the properties of the liquid, $\dot{q}_0$ is the reference heat flux, $F(p^\star)$ is the factor accounting for the pressure, $F(d)$ is accounting for the diameter, $F(W)$ account for properties of the wall and $F(\dot{m}, \dot{x})$ account for the mass flux and quality. All these factors have an corresponding equation while the reference values depends on the given fluid. The pressure factor can be calculated as eq. 4.43.

$$F(p^\star) = 2.816 \cdot p^{\star 0.45} + \left(3.4 + \frac{1.7}{1 - p^{\star 7}}\right) \cdot p^{\star 3.7} \tag{4.43}$$

Where $p^\star$ is the reduced pressure. The factor for the diameter can be calculated as eq. 4.44

$$F(d) = \left(\frac{d_0}{d}\right)^{0.4} \tag{4.44}$$

Where d_0 is the reference diameter which is 0.01 m. The wall factor can be calculated as eq. 4.45.

$$F(W) = \left(\frac{R_a}{R_{a0}}\right)^{0.133} \tag{4.45}$$

Where R_{a0} is the reference value which is given as 1 μm and R_a [μm] is the arithmetic mean roughness height. The last factor which depends on the quality and the mass flux have shown that there generally are no correlation between either of those parameters so it will simply be put to 1. Lastly there is the factor, n, which the heat flux is taken to the power of. This is calculated differently for inorganic fluid and hydrocarbons compared to cryogenic fluids. For the first case, which contains most of the commonly used refrigerants, it can be calculated as 4.46.

$$n = 0.8 - 0.1 \cdot 10^{0.76 \cdot p^{\star}} \tag{4.46}$$

The cryogenic fluids are lest dependent on the reduced pressure and can be calculated as eq. 4.47.

$$n = 0.7 - 0.13 \cdot 10^{0.48 \cdot p^{\star}} \tag{4.47}$$

Regarding the reference values it will be determined from looking up the values in tables. The exception for this is C_F which can be calculated from eq. 4.48

$$C_F = 0.435 \left(\frac{Mol_{mass}}{Mol_{mass,H2}}\right)^{0.27} \tag{4.48}$$

This approach is very general through the value should not exceed 2.5. This approach for nucleate boiling is valid for the following conditions.

$$\begin{gathered} 0.01 \leq p^{\star} \leq 0.985 \\ 1 \leq d \leq 32 \\ 0.05 \leq Ra \leq 5 \end{gathered} \tag{4.49}$$

Where d is in [mm] and R_a is in [μm]. The combined heat transfer coefficient for the convective and nucleate boiling can be calculated by using eq. 4.50.

$$h_{\text{tp}} = \sqrt{h_{\text{conv}}^2 + h_{\text{nuc}}^2} \tag{4.50}$$

However this approach is still limited by the amount of fluids that there are reference values available for. For the case where it is not available a relation for pool boiling can be utilised. This approach was examined by Cooper [1984] where the relation was reduced to only depend on the molecular weight, reduced pressure and the heating surface roughness. This can be formulated as eq. 4.51.

$$h_{pool} = 55 \cdot P_r^{0.12-0.2 \cdot log(R_p)} \cdot (-log(P_r))^{-0.55} \cdot Mol_{\text{mass}}^{-0.5} \cdot q^{0.67} \tag{4.51}$$

While this relation is developed for pool boiling it has been found to also be useful for predicting flow boiling which has been shown in studies such as the one made by Zhao et al. [2020]. By using this correlation or the one described in eq. 4.42 the heat transfer before the critical heat flux can be modelled.

Mist Boiling

Mist boiling is the heat transfer method which is present after the dry out happens. Here the liquid that needs to be evaporated are present as small droplets within a core flow of gas. This means that the heat transfer is mainly going to be from the wall to the gas which is significantly lower than the heat transfer that was present when the liquid was boiling on the surface. The way this is modelled is by calculating the heat transfer as it would be if it was just a one phase gas flow where there is added a multiplication factor and the Reynolds and Prandtl number is calculated as being for a homogeneous mixture as the method was presented by Wojtan et al. [2005]. It should be noted that this flow is not really suitable for being modelled as a homogeneous mixture, but the alternative is to model it as a pure gas flow, which completely neglects the effect the droplets has. The best option would be to use a method developed for a bubbly flow in a pillow plate, but that was not found due to a lack of literature on the subject. The Reynolds number is calculated using eq. 4.52.

$$Re_H = \frac{G \cdot d}{\mu_V} \left(x + \frac{\rho_g}{\rho_l} \cdot (1 - x) \right) \tag{4.52}$$

Then there is the multiplication factor which is calculated as eq. 4.53

$$Y = 1 - 0.1 \cdot \left[(\frac{\rho_l}{\rho_g} - 1) \cdot (1 - x) \right]^{0.4} \tag{4.53}$$

This can then be multiplied on eq. 4.19 instead of the correlation for pipe that was given in the study. This is then implemented in the model such that before the critical heat flux is reached the approach in the previous section is utilised and after the critical heat flux this approach will be utilised. This is a simplification because before the critical heat flux there should still be dry spots which has a lower heat transfer and after the critical heat flux there will probably still be some spots where the fluid will boil on the surface with a higher heat transfer. Besides that there is also the overall simplification that the flow will behave similarly to the flow within a pipe for which there is expected to be noticeable differences.

4.3 Heat exchanger model

With the method for calculating the thermal resistances described, the local heat transfer of the pillow plate can be calculated. However, to calculate the total heat transfer there needs to be accounted for the change in temperature and other properties through the process. There exist several method for doing this depending on what needs to be calculated during the process. The goal of this project is to predict the performance of a given pillow plate so therefore the Number of Transfer Units(NTU) method will be utilised. The specific way it will be utilised is as part of the cell method. Here the main idea is that the dimensionless temperatures can be calculated as a function of the area, thermal resistance and flow configuration. This can then be split into smaller parts to account for differences in one of those parameters. In the case of a pillow plate this will allow to account for the potential mixture of cross and counter flow that will be present in the pillow plate and the changing quality which affects the thermal resistance. This method will be implemented as described by Roetzel and Spang [2010]. This method will result in the pillow plate being split into three zones as seen in Figure 4.9 if there is a baffle, otherwise it will just be a single zone.

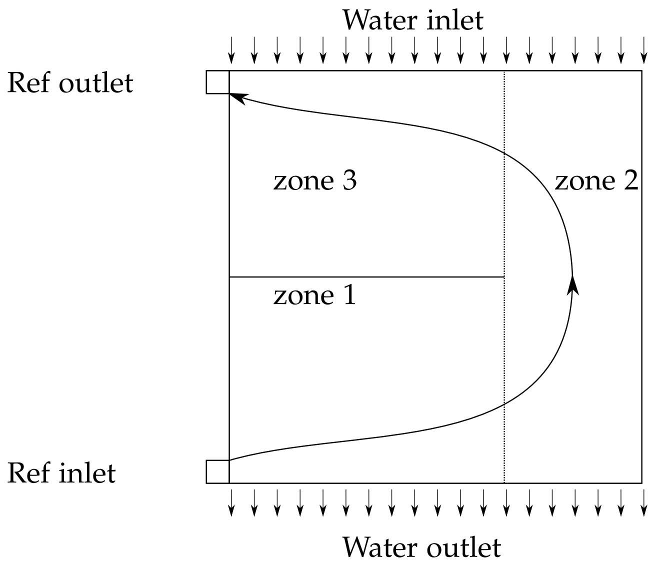

Figure 4.9: Zones of the pillow plate with different flow configuration.

From the figure it can be seen that in zone 1 and 3 there will be cross flow while there will be counter flow in zone 2, which is the only zone present without baffles. It can also be seen that there could be a difference in the flow rate for the heating media in zone 2 depending on the size of the baffle. All of this needs to be accounted for when calculating the heat transfer for the individual cell. Firstly there are a couple of parameters that needs to be defined for each cell. Firstly there are the NTU for each flow which is defined as eq. 4.54.

$$NTU_x = \frac{k \cdot A}{W_x} \tag{4.54}$$

Where A [m^2] is the surface area and W_x [J/K] is the heat capacity rate for the specific flow which can be calculated as eq. 4.55.

$$W_x = \dot{m}_x \cdot cp_x \tag{4.55}$$

Where cp_x [J/(kg K)] is the specific heat of the fluid. It should be noted that in the case of a two phase flow the heat capacity is defined to be infinite [Roetzel and Spang, 2010]. The heat capacity rate is another important parameter that describe the ratio of the heat capacity between the two streams, which is defined as eq. 4.56.

$$R_1 = \frac{W_1}{W_2} \tag{4.56}$$

All these dimensionless values are needed to calculate the dimensionless temperature which is defined as eq. 4.57

$$P_x = \frac{T_{x,in} - T_{x,out}}{T_{1,in} - T_{2,in}} = \frac{1 - exp((R_x - 1) \cdot NTU_x \cdot F)}{1 - R_x \cdot exp((R_x - 1) \cdot NTU_x \cdot F} \tag{4.57}$$

Where the factor F is to account for the flow configuration. For pure counter flow it will be unity, while for other configuration it can be calculated using eq. 4.58.

$$F = \frac{1}{(1 + a \cdot R_1^{d \cdot b} \cdot NTU_1^b)^c} \tag{4.58}$$

Where a, b, c and d depend on the given flow. For pure cross flow the following values can be used:

- $a = 0.433$
- $b = 1.6$
- $c = 0.267$
- $d = 0.5$

Using this method it is possible to calculate the temperature changes in each single cell and this can then be calculated as a complete system of equations. Then by evaluating the total change in temperature, the total heat transfer can be estimated.

4.4 Pressure loss

When considering a heat exchanger application, it is also necessary to examine the associated pressure loss. Generally high heat transfer coefficients will result in high pressure losses so it is important to be able to model these to account for the required pumping power. When considering a single phase flow the most important pressure loss will generally be due to friction while gravitational losses also can be influential. This is also the case for two phase flow where there is added another pressure loss due to acceleration. To account for the gravitational and acceleration pressure losses it is important to precisely estimate the fraction of the volume that is occupied by gas and the fraction that is occupied by liquid [Whalley, 1996]. This is represented by the void fraction, α which represent the volume fraction occupied by the gas, the definition of this can be seen in eq. 4.59.

$$\alpha = \frac{1}{1 + \left(\frac{u_g}{u_l} \cdot \frac{1-x}{x} \cdot \frac{\rho_g}{\rho_l}\right)} \tag{4.59}$$

From the equation it can be seen that this void fraction depend on the quality of the mixture, the density of the two phases and the velocity of the two phases. The first two of these are easily calculated for a given mixture, but the ratio between the velocities are more difficult to calculate. This ratio is also called the slip factor and there are several relations which can be used to calculate it. One of these are a correlation which calculates the slip factor as defined in eq. 4.60 [Whalley, 1996].

$$S = \left[1 - x \cdot \left(1 - \frac{\rho_l}{\rho_g}\right)\right] \tag{4.60}$$

Using this equation it is possible to calculate the void fraction as a function of the quality. The first pressure loss that will be examined will be the frictional pressure loss. The frictional pressure loss for a two phase flow is closely related to the pressure loss for a one phase flow. Therefore the correlation for pressure loss in a pillow plate in the study by M.Piper et al. [2017] will be used. Similarly to the method used for modelling the heat transfer, there was made a power law correlation to calculate the Darcy friction factor which can be seen in eq. 4.61.

$$\zeta_{\Delta P} = n_1 \cdot Re^{n_2} \tag{4.61}$$

Where n_1 and n_2 are the power law coefficients. The value of these coefficients can be seen in Table 4.2 for the three pillow plate types.

Table 4.2: Table with coefficient for the Darcy friction factor power law for three types of pillow plates.

	$a \approx 0.58$ $0.1 < b < 0.14$ $0.042 < c < 0.083$	$a \approx 1$ $0.17 < b < 0.24$ $0.071 < c < 0.143$	$a \approx 1.71$ $0.17 < b < 0.24$ $0.071 < c < 0.143$
n_1	$8.74b + 17c + 0.73$	$-15.3b + 1.4c + 5.4$	$1.35b + 2.8c + 0.92$
n_2	-0.38	$1.725b + 1.11c + 0.66$	$0.3b + 0.53c - 0.29$

Using the values given in the table it is possible to calculate the one phase pressure loss using the standard pressure loss equation with the Darcy friction factor which can be seen in eq. 4.62.

$$\Delta P_{fric} = \frac{\zeta_{\Delta P} \cdot \rho \cdot u_m^2 \cdot L}{2 \cdot d_h} \tag{4.62}$$

However, this equation is only capable of accounting for the pressure loss for a single phase. Therefore the end result will have to be modified to account for the effects of the different phases in the flow. The simplest method for this is with a two phase multiplier, the simplest of which is when the two phase flow is homogeneous, which mean that the slip factor will be equal to unity. This is only the case if the ratio between the densities is less than 10 and that the mass flux, G, is larger than $2000\,\mathrm{kg/(m^2s)}$. However, the flow within pillow plates are generally not that large, so assuming the flow to be homogeneous will lead to noticeable errors. Therefore the more complex Friedel two phase multiplier model will be used to determine the two phase flow multiplier [Vassallo and Keller, 2006]. Firstly the two phase multiplier is defined as eq. 4.63.

$$\phi_{lo}^2 = \frac{\Delta P_{2\phi}}{\Delta P_{lo}} \tag{4.63}$$

The model Friedel developed for this calculates the two phase multiplier as eq. 4.64.

$$\phi_{lo} = E + \frac{3.24 \cdot F \cdot H_F}{Fr_h^{0.045} \cdot We_l^{0.035}} \tag{4.64}$$

To be able to calculate this parameter there are a many coefficient that needs to be calculated. Firstly there is the E parameter which is dependent on the quality, Darcy friction factor and density for the two phases which is defined as eq. 4.65.

$$E = (1-x)^2 + x^2 \cdot \frac{\rho_l \cdot \zeta_l}{\rho_g \cdot \zeta_g} \tag{4.65}$$

Then there is the F parameter which solely depends on the quality of the mixture and is defined as eq. 4.66.

$$F = x^{0.78} \cdot (1-x)^{0.224}; \tag{4.66}$$

The next parameter is H which is a power law relation of the viscosity and density of the two different phases and is defined as eq. 4.67.

$$H = \left(\frac{\rho_l}{\rho_g}\right)^{0.91} \cdot \left(\frac{\mu_g}{\mu_l}\right)^{0.19} \cdot \left(1 - \frac{\mu_g}{\mu_l}\right)^{0.7} \tag{4.67}$$

Then there are two parameters in the denominator. Both of these parameters depend on the homogeneous density which is defined as eq. 4.68.

$$\rho_{hom} = \left(\frac{x}{\rho_g} + \frac{1-x}{\rho_l}\right)^{-1} \tag{4.68}$$

The first parameter in the denominator is Fr_h which is defined as eq. 4.69.

$$Fr_h = \frac{G^2}{g \cdot d_h \cdot \rho_{hom}^2} \tag{4.69}$$

Then lastly in the denominator is the liquid Weber number which is defined as eq. 4.70.

$$We_l = \frac{G^2 \cdot d_h}{\sigma \cdot \rho_{hom}}; \tag{4.70}$$

Thereby it has been defined how to calculate all the parameter in eq. 4.64 which together with eq. 4.62 makes it possible to calculate the frictional pressure loss for the pillow plate while accounting for there being two different phases. Besides the pressure loss from friction there is also the pressure loss from acceleration and gravity. The gravitational loss can be calculated from eq. 4.71 for a constant density and void fraction [Whalley, 1996].

$$\Delta P_{grav,2ph} = \left(\alpha \cdot \rho_g + (1-\alpha) \cdot \rho_l\right) \cdot g \cdot sin(\theta) \cdot l \tag{4.71}$$

The changing void fraction and density will be accounted for using numerical integration methods. The acceleration pressure loss can be calculated from eq. 4.72.

$$\Delta P_{acc,2ph} = G^2 \cdot \int \frac{d}{dz} \left(\frac{x^2}{\alpha \cdot \rho_g} + \frac{(1-x)^2}{(1-\alpha) \cdot \rho_l}\right) \tag{4.72}$$

For this equation the result would be zero if there is no change in quality, but as there is the differentiation and integration will be solved using numerical methods.

Chapter 5

Validation

After creating the model it needs to be validated. To valuate the model it needs to be compared to experimental data or other models that have a proven accuracy. This model will be validated by comparing it to data provided by ATHCO, which is a producer of pillow plates. This data is based their current method for dimensioning their pillow plates. This data contain different configurations of the hot and cold fluid for the same plate. The plate have the measurements in table 5.1.

Table 5.1: Parameters of pillow plate used for validation study.

Parameter	Measurement	Unit
H_{PP}	1,000	mm
w_{PP}	2,000	mm
δ_{in}	5	mm
S_l	30	mm
S_t	35	mm
Thickness	1.5	mm
d_{sp}	7.2	mm

Using the values presented in the table, the validity of the model can be compared to the data. There was six different configurations in the data which can be seen in table 5.2.

Table 5.2: Input parameters for the six different configurations.

Number	Nr_{plates}	Refrigerant	T_{evap}	$\dot{m}_{ref}$	$\dot{m}_{film}$	T_{in}
1	12	R404A	-3	1.2	8	7
2	4	R404A	-3	0.4	2.9	6
3	6	R404A	-3	0.6	4.8	6
4	28	R507A	-3	2.9	18.7	6.5
5	13	R507A	-2	1.2	9.2	7
6	7	R404A	-6	1.4	5.6	12

From the table it can be seen that the refrigerant used is either R404A or R507A, neither of which have table values for eq. 4.42 on page 24. Therefore for the validation it will be replaced with eq. 4.51 on page 26. Besides the refrigerant, the number of plates is also varied which is followed by changes to both the refrigerant and film flow rate. The last parameter is the inlet temperature of the falling film. It should be noted that these configurations are quite similar, but there are some variation between them. The result from these configurations can be seen in table 5.3 for both the measurements and the model.

Table 5.3: Result for comparison of provided data and model result.

Number	Expected		Calculated	
-	U	T_{out}	U	T_{out}
1	636	1	590	1.31
2	608	1	611	1.04
3	676	1	607	1.37
4	698	0.5	605	1.01
5	598	2	564	2.21
6	673	2	722	1.61

From the table it can be seen that most of the result from the model matches the data, with it slightly under predicting the heat transfer for all cases except number 6. The reason that number 6 is the only one that over predicts the heat transfer is that it is the only one which will have super heat. Super heating is not properly accounted for in the model as the temperature of the refrigerant will not be increasing even after the refrigerant has completely evaporated. This results in a higher heat transfer than what would be possible if the refrigerant temperature increased. The under prediction can be caused by several of the assumptions made regarding it acting as a pipe, the most likely being to which degree the dry out will happen. However, the margin of error is acceptable with the largest variation for the overall heat transfer coefficient being 13.4% and it being 8.5% for the temperature change. Therefore the model is generally able to accurately predict the heat transfer. It should be noted that the Reynolds number from the model generally was around 500-600 for the refrigerant meaning that they are below the lower limit of the Nusselt relation for the pillow plate. To understand how that affects the result it will be compared to another relation which was developed for Reynolds number from 500-10,000. the result of which can be seen in Figure 5.1.

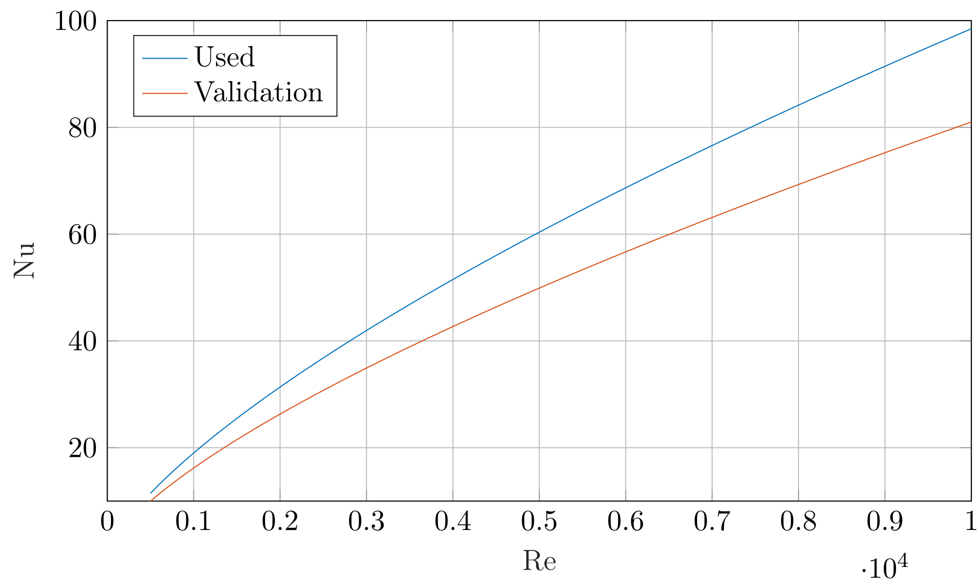

Figure 5.1: Comparison of result from the heat transfer correlations proposed by M.Piper et al. [2017] and Shirzad et al. [2019] for different Reynolds numbers

From the figure it can be seen that the two correlations are similar with the difference increasing for higher Reynolds number. Therefore while the accuracy that was found in the original study of ±15% is not applicable outside the examined range, there is no development that will lead to large errors. Following the comparison to the data and the other validation it can be seen that the model has a margin of error of about 14% for the overall heat transfer coefficient and 9% for the temperature change. These differences tends to underestimate the heat transfer, unless there is super heat.

Chapter 6

Parametric Study

After examining the validity of the model, it can be used to make a parametric analysis. This is both useful for understanding how the pillow plate design potentially can be improved, but it can also be used to further examine the validity of the model. This can be done as there is an expected effect of changing some parameters, which the model is expected to follow. The first step in making an parametric analysis is to determine which parameters that will be examined and which parameters that will be kept fixed.

6.1 Examined Parameters

The main parameters that affects the heat transfer can be split into three groups. There are parameters relating to the refrigerant flow, to the film flow and to the plate geometry. All three groups will be examined to some extent. In regards to the refrigerant flow the parameters which will be varied is the refrigerant used and the flow rate. For the plate geometry it will be the overall size and the inner height. For the film flow the parameters will be the flow rate and temperature of the fluid. All of these parameters are the ones that will be varied. This means that several parameters will be fixed, the values of which can be seen in Table 6.1.

Table 6.1: Table of the fixed parameters in the parametric study.

Parameter	Value	Unit
St	35	mm
Sl	30	mm
d_{sp}	7.5	mm
Nr_{plates}	20	-
w_w	15	mm
thickness	1.5	mm
k_{plate}	14.2	W/m
T_{ref}	-5	$^{\circ}C$

Besides the fixed parameters there also needs to be given a range for the variation of the examined parameters and a reference value. The value of these can be seen in Table 6.2.

Table 6.2: Table of the varying parameters in the parametric study.

Parameter	Reference Value	Variation	Unit
δ_{in}	5	2.5-5	mm
$\dot{m}_{ref}$	5	3-15	kg/s
$\dot{m}_{film}$	30	20-50	kg/s
T_{film}	5	3-10	K
Height	1	0.5-2.5	m
Width	2	1-3	m

By varying these parameters it will be examined which of these parameters are important for improving the performance of the pillow plate. When varying the mass flow there will also be examined three refrigerants, those being ammonia(NH3), R507A and R404A, while the rest will be for R404A.

6.2 Results

Before the parameter study is made, it will firstly be examined how some of the important parameters change from the inlet to the outlet of evaporator scaled by the height. This will also form a base line for the values that will be examined in the parameter study. Firstly it will be examined how the quality of the refrigerant changes through the pillow plate, the result of which can be seen in Figure 6.1.

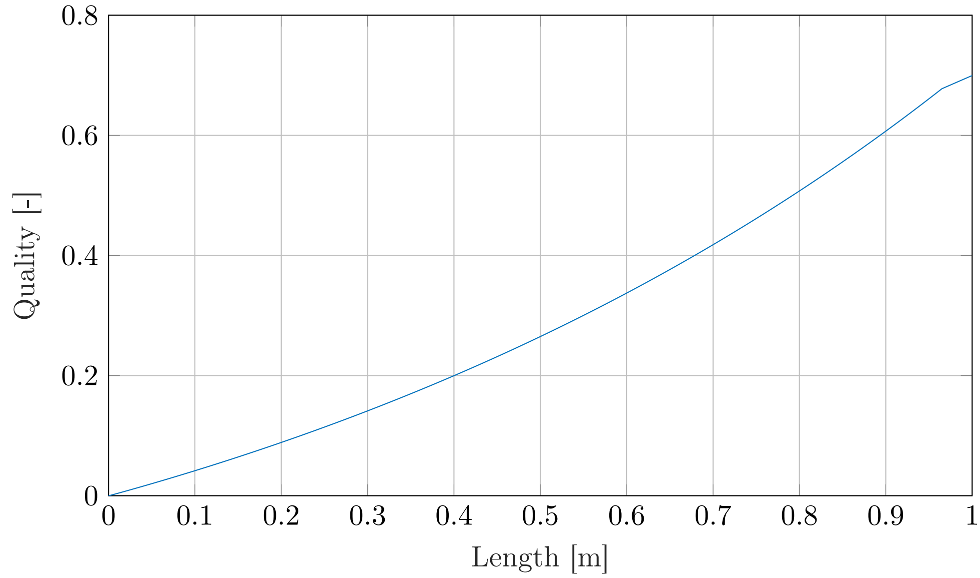

Figure 6.1: Development of the quality from the from the bottom of the plate to the top.

From the figure it can be seen that the quality increase at a steady rate until around 0.95 m where the rate decreases. It can also be seen that the outlet quality is about 0.7 meaning that there was evaporated about 3.5 kg/s refrigerant. Besides the amount evaporated, the local heat flux is also examined which can be seen in Figure 6.2.

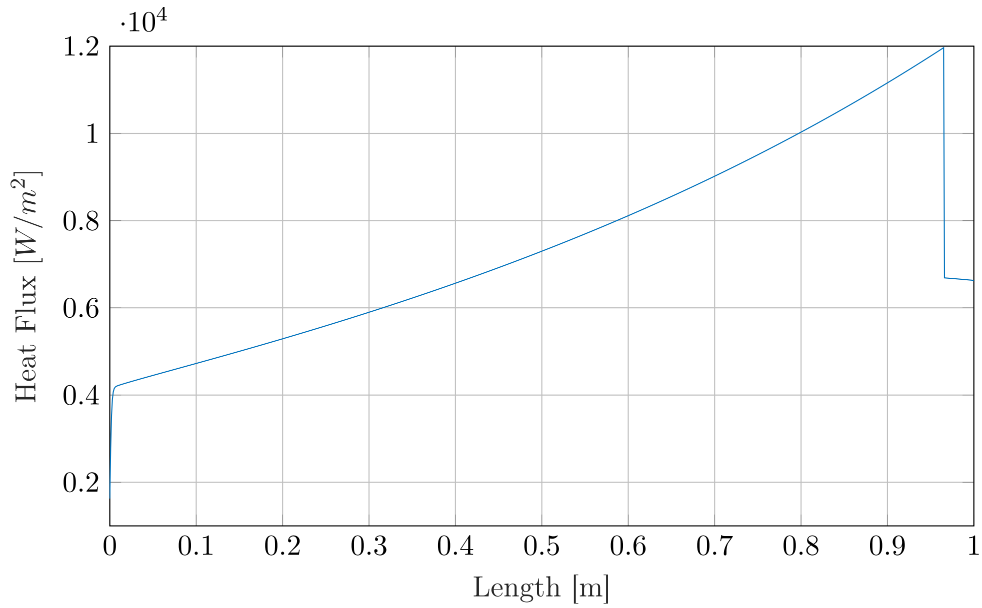

Figure 6.2: Development of the heat flux from the from the bottom of the plate to the top.

From the figure it can be seen that the heat flux follows a similar trajectory to the slope of the quality which is expected at they are directly correlated. The decrease in the slope for the quality can also be seen on this figure, where the local heat flux has a sharp drop. This point is where the dry out happens which is known to sharply decrease the heat transfer. This is due to fluid that touches the plate will now be gas and not liquid, which changes the heat transfer from boiling to forced convection. The last thing that will be examined is how the water temperature changes from inlet to outlet which can be seen in Figure 6.3.

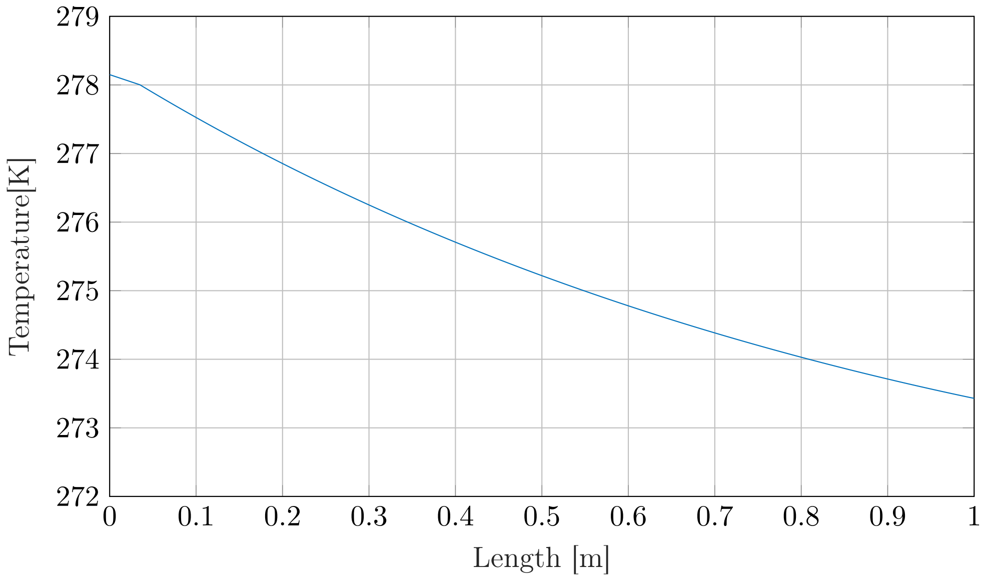

Figure 6.3: Development of the water film temperature from the from the top of the plate to the bottom.

It can be seen from the figure that like the other parameters there are two different rates at which the temperature of the film decreases. At the start the decrease is slower which is because the dry out reduces the heat transfer at the top of the pillow plate whereafter the decrease is sharper. From these plots it can be seen that the heat transfer is generally increasing with the quality through the pillow plate until dry out happens. Thereafter there will be a sharp decrease in the heat transfer which is followed by a slowly decreasing heat transfer. Now that the standard measurements have been examined the parameter study can be made, where the first parameter that will be examined is the inner height.

Inner Height

The first parameter that is examined is the inner height. The inner height is determined during the production process by the pressure which the plate is expanded by. Increasing the inner height of the pillow plate will result in a larger cross sectional area and hydraulic diameter. The larger cross sectional area leads to an decrease in velocity of the refrigerant, which should result in a decreased heat transfer, however it should also reduce the pressure losses. Firstly it will be examined how the heat transfer is affected, which can be seen in Figure 6.4.

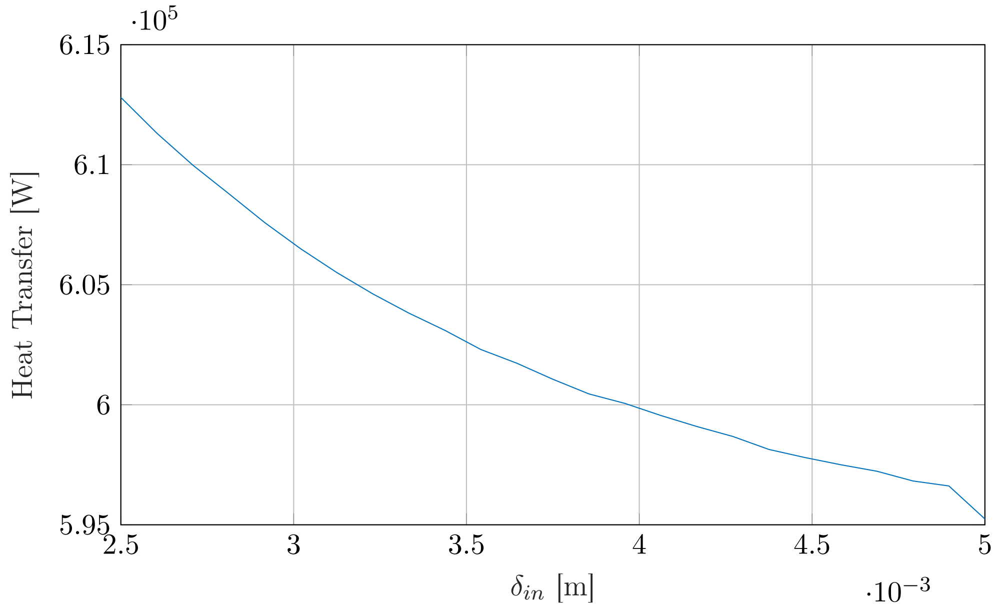

Figure 6.4: Effect of changing the inner height of the pillow plate on the overall heat transfer.

From Figure 6.4 it can be seen that the heat transfer decreases with the increasing inner height which is expected. However, it can also be seen that the decrease is less than 3% by doubling the inner height so this parameter is not very important for the overall heat transfer.

Mass Flow Rate Refrigerant

The next parameter that will be examined is the mass flow rate of the refrigerant. This is a parameter that can be adjusted both by increasing the compressor power or by adjusting the number of plates. Also, as this is a flooded evaporator, it can also be done by increasing the recirculation rate which is the flow rate of the evaporator compared to the flow rate of the whole system. It is expected that increasing the mass flow will increase the heat transfer due to a higher velocity, but the outlet quality will be lower as more refrigerant needs to be evaporated. While examining the mass flow rate, it will also be examined how different refrigerant affect the system. The result regarding the heat transfer for different refrigerants and flow rates can be seen in Figure 6.5.

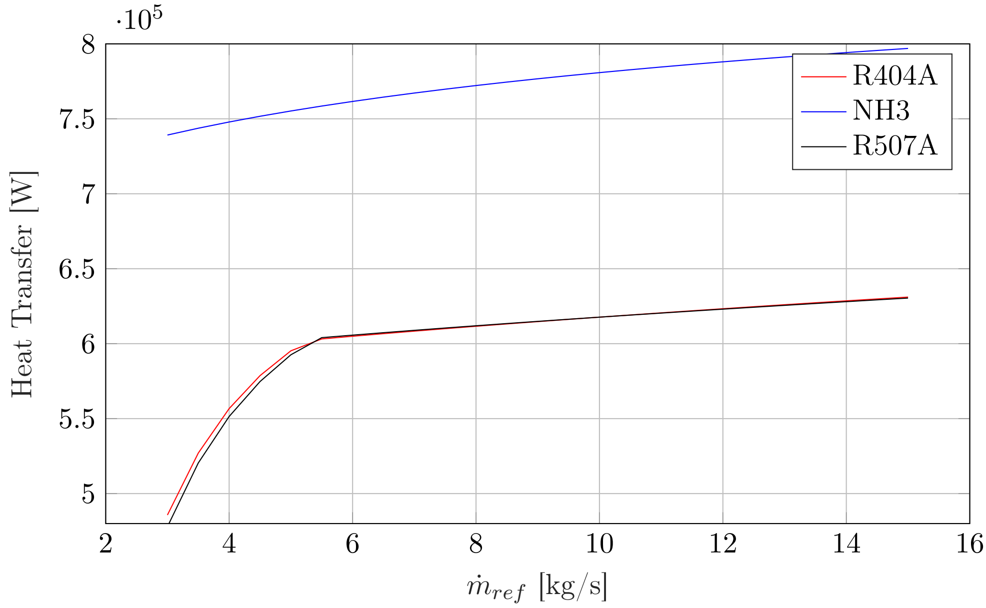

Figure 6.5: Effect of changing the refrigerant mass flow on the overall heat transfer.

From the figure it can be seen that from a mass flow rate of 3 kg/s to 6 kg/s there is a noticeable increase in heat transfer for both R404A and R507A. Afterwards the increase is mostly linear. Meanwhile the heat transfer for NH3 is significantly higher. This is due to it having both a higher one phase heat transfer coefficient and a higher nucleate boiling coefficient. It can also be seen that after a flow rate of 6 kg/s the increase in heat transfer is slightly higher for NH3, which suggest it benefits more from a higher refrigerant velocity compared to the other refrigerants. To have a further understanding of the phenomena that was observed for the heat transfer it is examined how the quality is affected by the mass flow of the refrigerant. The result of this investigation can be seen in Figure 6.6.

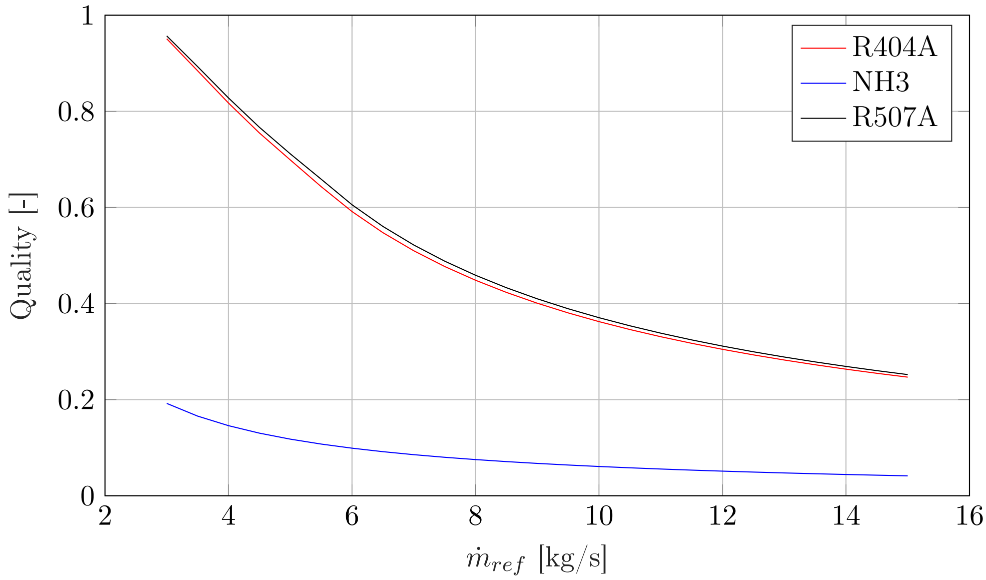

Figure 6.6: Effect of changing the refrigerant mass flow on the outlet quality.

From the figure it can be seen that the outlet quality decreases with the flow rate which is expected. In regards to the quality of R507A and R404A where the sudden increase happens it can be seen that it is around 0.6. Looking at Figure 6.2 and 6.1 on page 39 it can be seen that the dry out happens at a quality around 0.6-0.7 which is where the noticeable increase also seems to stop. Therefore the noticeable increase should be due to dry out not happening early in the pillow plate or at all. It can also be seen that while NH3 had a significantly higher heat transfer, the outlet quality is much lower which is due to NH3 having a higher latent heat. From this parametric analysis it can be seen that adjusting the flow rate is an effective method for avoiding dry out and thereby increasing the heat transfer. Furthermore it can also be seen that the choice of refrigerant can have a significant effect on the heat transfer and the outlet quality which needs to be accounted for when designing the system. Of the three refrigerants examined it was found that NH3 has a significantly better heat transfer, but due to its high latent heat the quality of the outlet will be quite low.

Mass Flow Rate Falling Film

Another parameter that is examined is the flow rate of the falling film. The flow rate of the falling film is mostly dependent on the heat source which is available, but in the specific case where the water needs to be cooled a certain amount this can be a parameter that is directly controlled. Firstly it will be examined how the heat transfer is affected which can be seen in Figure 6.7.

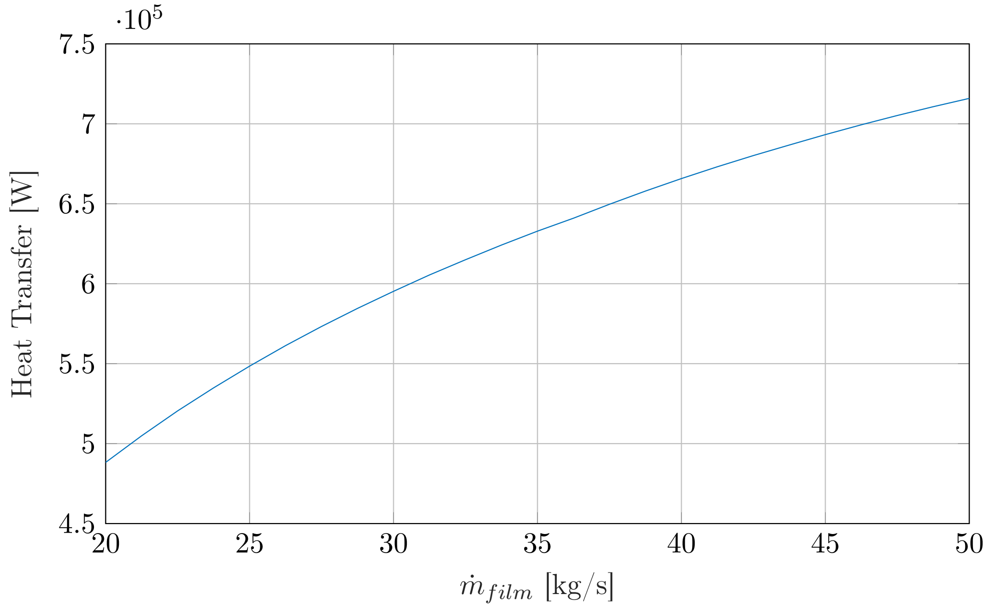

Figure 6.7: Effect of changing the falling film mass flow on the overall heat transfer.

From the figure it can be seen that increasing the mass flow of the falling film is also an effective way of increasing the heat transfer. By increasing the mass flow of the falling film from 30 kg/s to 50 kg/s the heat transfer is increased by more than 100 kW. The reason for this increase in heat transfer is due to two factors. Firstly it is due to the convective heat transfer from the falling film increasing with the flow and the second reason is the average temperature of the falling film will be higher with a higher flow rate due to a higher heat capacity. Therefore this is further examined by investigating the outlet temperature of the falling film which can be seen in Figure 6.8.

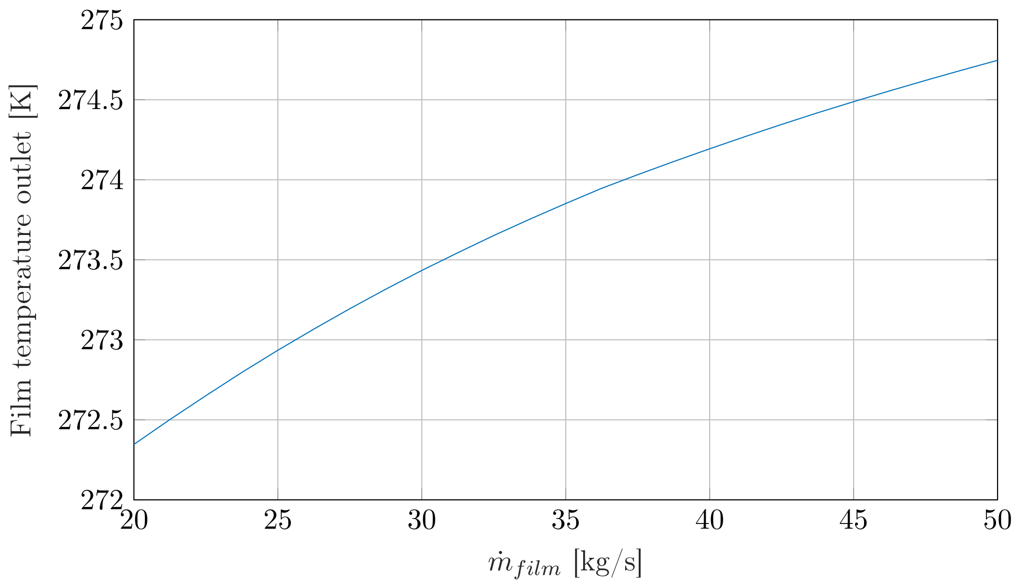

Figure 6.8: Effect of changing the falling film mass flow on outlet temperature of the falling film.

From the figure it can be seen that the temperature follows a very similar development as the heat transfer which is expected as they are directly correlated. It can also be seen that if the flow rate is reduced just a bit below 30 kg/s the water temperature will approach the freezing point of water which in reality will have a noticeable effect on the heat transfer which the model does not account for. It can also be seen that the increase in temperature is about 1.2 K which changes the logarithmic mean temperature difference to 8.18 K from 7.39 K, which is an increase of 9.7 % which is more than half of the 16.9 % increase there was seen in the heat transfer. Therefore the temperature increase is a large part of the increase in heat transfer, but the increase in heat transfer coefficient is also important. It should also be noted that as there is an increase in heat transfer the quality will also increase which result in a larger amount of the pillow plate will experience dry out. Therefore it could be beneficial to also increase the flow rate of the refrigerant. From this and the previous parameter study it is also clear that it is possible to increase the heat transfer per plate by having the same conditions and reducing the number of plates.

Temperature Falling Film

Besides the flow rate of the falling film it is also examined how the inlet temperature of the water affects the heat transfer. The temperature of the water is mostly dependent on the heat source, so this study will examine how the same setup can perform under different conditions. The results for the heat transfer can be seen in Figure 6.9.

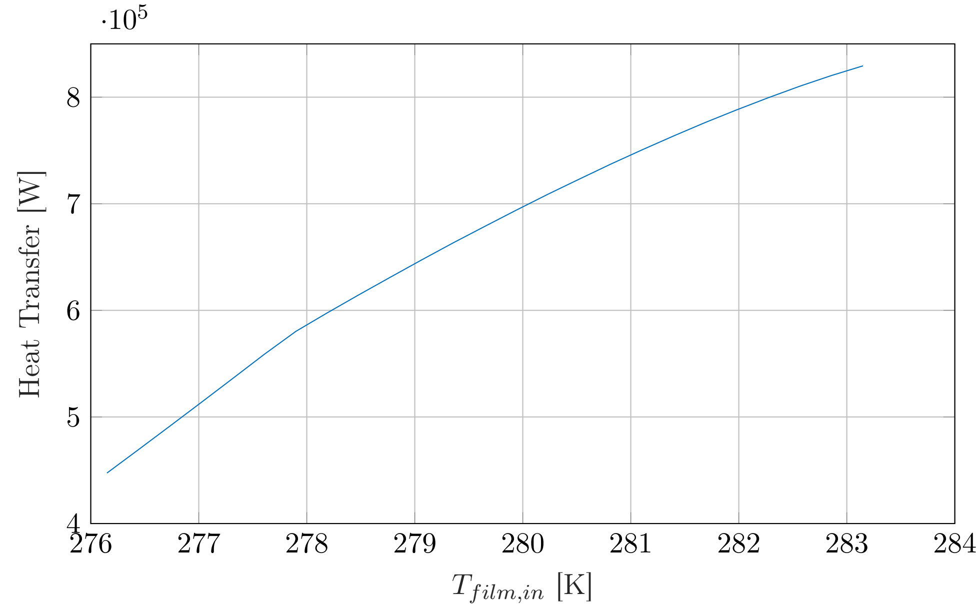

Figure 6.9: Effect of changing the inlet temperature of the falling film on the overall heat transfer.

From the figure it can be seen that the increase in heat transfer increases mostly linearly with the temperature which is expected since the changes to the overall heat transfer should not be that large when changing the temperature. Besides affecting the heat transfer, the outlet temperature of the falling film is also affected which can be seen in Figure 6.10.

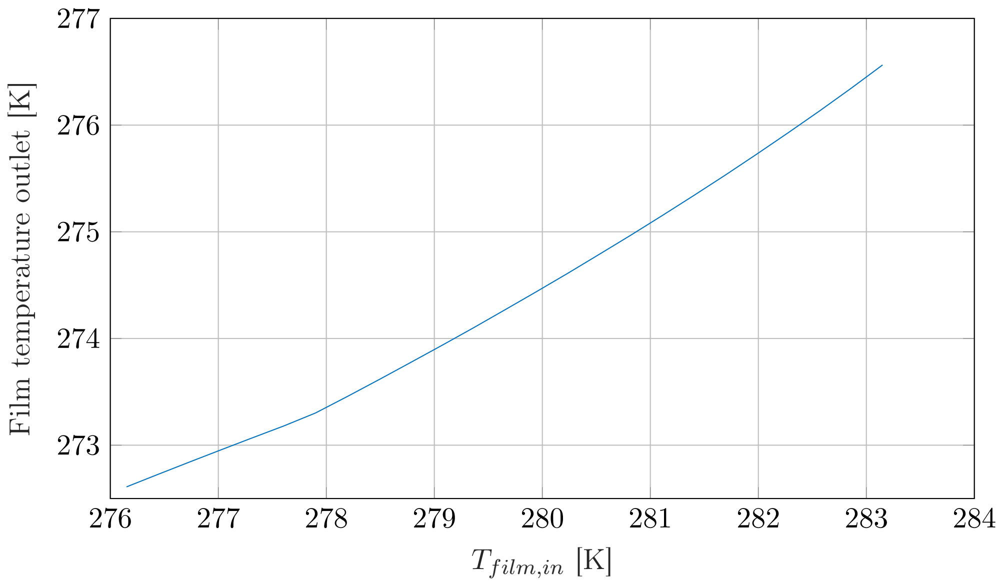

Figure 6.10: Effect of changing the inlet temperature of the falling film on the on the outlet temperature of the falling film.

From the figure it can be seen that the water temperature steadily increases with a higher inlet temperature which is expected, and it can also see that there is a small change in the slope at 278 K which was also observed for the heat transfer. It can also be seen that for the current configuration the inlet temperature should not be more than a bit below 278 K, because it will the result in ice formation on the pillow plate which will reduce heat transfer.

Height

Lastly there are two parameters that will be examined in regards to the size of the plate those being the height and the width. The result for the height in regards to the heat transfer can be seen in Figure 6.11.

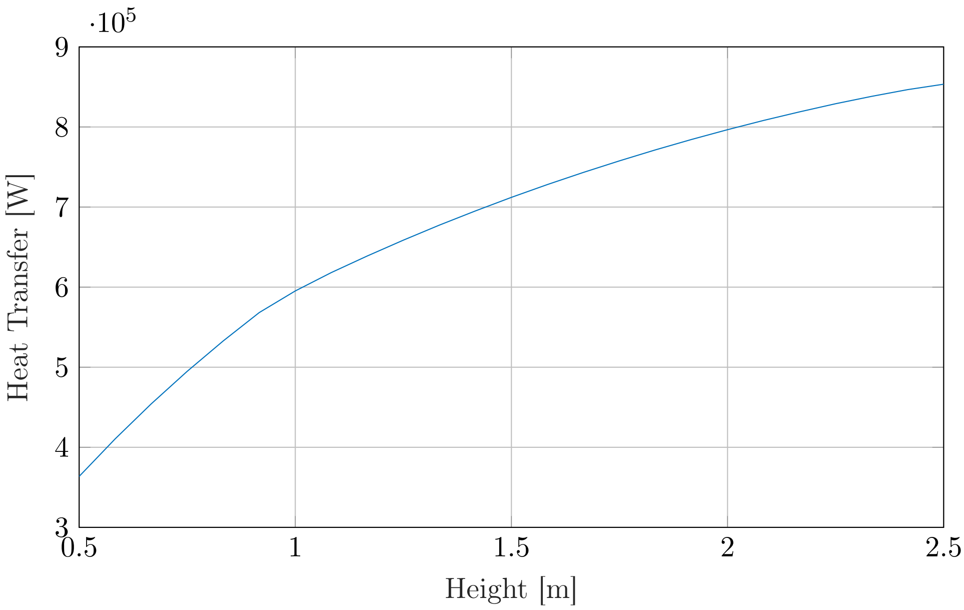

Figure 6.11: Effect of changing the height of the pillow plate on the overall heat transfer.

From the figure it can be seen that the increase in heat transfer is more significant below 1 m which matches where the dry out happens in Figure 6.2 on page 39. This means that the heat transfer gained from increasing the height from 0.5 m to 1 m is significantly more effective than increasing from 1 m to 1.5 m. It can also be seen that after 1 m the slope of the increase is decreasing which is due to the water temperature falling lower. Besides the heat transfer the height can also be examined for what it should be if there is a desired quality at the outlet. The results for the quality can be seen in Figure 6.12.

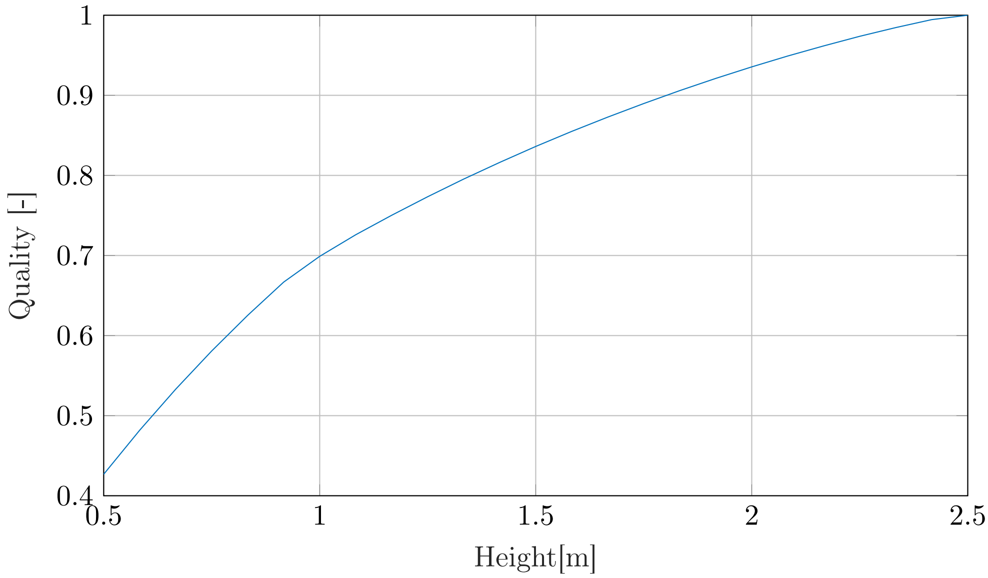

Figure 6.12: Effect of changing the height of the pillow plate on the outlet quality.

From the figure it can be seen that the quality follows a very similar development as the heat transfer, which is expected. It can also again be seen that the dry out seems to happen at a quality of about 0.68. It can also be seen that by increasing the height to about 2.5 m it is possible to reach a quality of about unity. This also clearly highlights the advantage of having a flooded evaporator as it can take advantage of the significantly better heat transfer at a lower quality while still delivering a quality of unity which is needed for the compressor.

Width

The last parameter which will be examined is the width of the evaporator. While changing the height and width will both increase the area of the pillow plate, increasing the width will also increase the cross sectional area and decrease the wetting rate, both which should decrease the heat transfer. The result for the heat transfer can be seen in Figure 6.13.

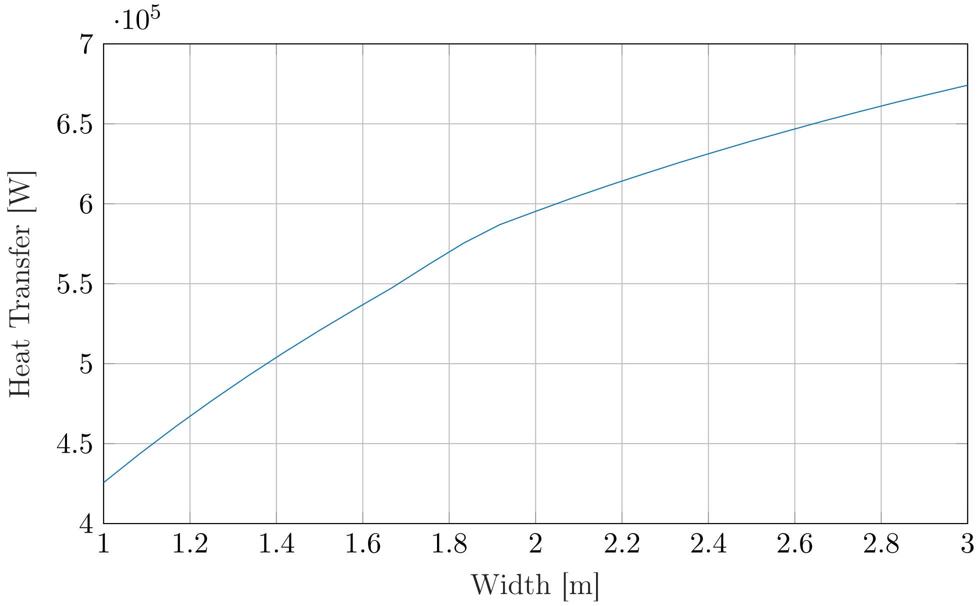

Figure 6.13: Effect of changing the width of the pillow plate on the overall heat transfer.

From the figure it can be seen that the heat transfer overall will increase as the area is increased, however if the increase is compared to the increase obtained by increasing the height the difference can be noticed. For a height of 1.5 m the total heat transfer is about 712 kW whereas it is 674 kW with a width of 3 m which gives the same total surface area. In general this parameter study has showed that there are many ways to increase the heat transfer, but the most effective method is to ensure an average large temperature difference that does not result in dry out. It was also found that if the area should be increased it is more beneficial to increase the height rather than increasing the width.

Chapter 7

Discussion

After examining some of the parameters for the pillow plate in the parametric analysis, the results and overall model will be discussed. This will be done starting with the model.

7.1 Model

The first part of the model which will be discussed is the Nusselt correlations used. When working with heat transfer models it is often necessary to rely on Nusselt correlations as very few heat transfer cases are able to be solved analytically. This is also the case for this study, where several Nusselt correlations were used. The first one of these is the one used for the one phase flow within the pillow plate. This Nusselt correlation has an accuracy of $\pm 15\%$, but it is only within the narrow range that it covers. The range that it covers is rather small as it is only for three specific types of plates and for Reynolds numbers between 1,000 and 8,000. During this study there was several times where it was observed that the parameters was out of the range an example being during the validation where the Reynolds number was around 500 and therefore below the lower limit. This was also something that could be observed when calculating the heat transfer for the gas phase as it would often exceed the upper limit and sometimes even reach Reynolds numbers up towards 20,000. This results in the accuracy for the study not being accurate for the gas flow, which especially affects the accuracy of the heat transfer post dry out. Therefore there is an increased uncertainty for the heat transfer post dry out. However, the heat transfer outside the range of the Nusselt correlation was examined and there was no sudden variation so the margin of error should not be significantly larger than the one described in the study.

Besides the Nusselt correlation for the one phase flow there was also used correlations for the two phase heat transfer within the plate and the falling film heat transfer on the outside of the plate. The issue with both these correlation are that they are not developed for a pillow plate. The Nusselt correlation for the falling film are made for a vertical smooth pipe, which means that the effect the unevenness that is present on the pillow plat is not accounted for. Also as the is a plate and not a pipe there will be an end to the plate which should effect the heat transfer locally. However considering the plates width, the effect of this should be quite minimal. The correlation for a vertical pipes was used as there was not found a correlation for falling film for a pillow plate during the literature study. This was also the reason why the boiling heat transfer was modelled based on a approach for vertical tubes. However here there was accounted for the pillow plate to some extend as the one phase heat transfer used in the equation was calculated for a pillow plate. The correlation for the nucleate heat transfer is not really dependent on the flow geometry so the expression used for that should also be suitable.

The flow within the pillow plate is also assumed to follow the same flow pattern as a vertical tube. Normally the flow pattern within a pipe follows a fixed pattern where the walls are mostly wet until dry out happens. However, the flow pattern within a pillow plate will be more turbulent which will form a different flow pattern. Therefore the method of calculating dry out for a pipe and assuming it can be applied to a pillow plate might not be accurate. It should also be noted that the method for calculating dry out is also based on a uniform heat flux which is not the case for the pillow plate, which could also change where the dry out happens. An alternative method that could be closer to a uniform heat flux would be to compare the average heat flux to the critical heat flux, instead of the local heat flux. This should result in the dry out happening later which would increase the heat transfer. However the effect of this was found to reduce the accuracy. Lastly there is the method for calculating the heat transfer after the dry out, which made the assumption of a homogeneous flow and thermal equilibrium between the gas and the droplets. The assumption of thermal equilibrium is mostly just a simplification and will result in a slight increase in heat transfer. The assumption of homogeneous is more troublesome as it is usually used for flows with significantly higher flow rates, which means using this method will result in errors. An alternative method would be to simply model it as a one phase gas flow, but then it would not be accounted for how the droplets affect the flow. This would result in a lower heat transfer right after dry out which will slowly get closer to the same value as the homogeneous approach. Optimally a method made specifically for this type of flow would be used, but none was found during the literature study.

Other than the correlations which are important there are also some assumptions that was made during the modelling that needs to be discussed. The most important of these is that there was assumed a uniform flow within the pillow plate and the falling film. This means that the heat transfer is the same for all the plates and therefore no difference between the first, middle and last plates. Also for the plates the heat transfer coefficient for the falling film is uniform, which assumes the plate is completely wet with no dry spots. It was also assumed that there are no difference for the heat transfer coefficient within the plate in the horizontal direction. This overall functions as a simplification to make it easier to model. The alternative would be to to make a CFD simulation or test the specific pillow plate. This would reduce the reliance on correlations that are made for pipes, but that is both a complicated process and requires significantly more time compared to the current model. Therefor this assumption is made, but it will result in some uncertainty. It should also be noted that the error resulting from a uniform flow should be more noticeable for plates that are much wider than high.

Following the correlations and assumptions, there was also the overall implementation of the method that needs to be considered. Here there are three main points which can affect the result. These parameters should be selected such that the model is independent of the specific value. These three are the start guesses for the iteration regarding the quality and the film temperature, the maximum allowable error and the number of cells the pillow plate is split into. The way this was examined was to first examine the amount that was evaporated with the standard conditions and then see the difference when either reducing the allowable error, increasing the number of cells or by changing the start guess. For the reference values there will be evaporated 3.4979 kg/s which is the value the other results will be compared to. By reducing the allowable error from 10^{-8} to 10^{-9} the amount evaporated changed to 3.4979 kg/s which was the same as the reference value until the 6^{th} decimal where it was slightly lower so the difference there is completely negligible. For the start guesses the were adjusted so the quality would be linear from 0 to 0.5 compared to the standard 0 to 1 and the reduction in film temperature was also halved. This changed the amount evaporated to 3.4980 kg/s which is an increase of 0.0045% which again is negligible. Lastly the number of cells was changed from 1000 to 1500, which changed the amount evaporated to 3.4990 kg/s which is an increase of 0.03% which is negligible. Therefore it can be seen that the numerical methods used for the modelling has converged. From this it can overall be seen that the model is robust on are capable of accounting for most of the important parameters which some assumptions while still having a acceptable accuracy.

7.2 Results

Besides the model, the results obtained in the parametric analysis will also be discussed. In the parametric analysis there was examined the effect of several of the input parameters from the model. These results showed that most of the parameters have a significant effect on the heat transfer. Therefore there are several different parameters that can be changed to improve the performance. However, it still follows the conventional theory for a high heat transfer that a large temperature difference, area and heat transfer coefficient is needed. A large temperature difference can either be achieved by increasing the temperature of water film inlet which will raise the overall temperature difference or by increasing the mass flow of the falling film which will increase the average temperature at the film temperature as the outlet temperature will be higher. There is also another method which was not investigated in this study, which is to reduce the saturation temperature of the refrigerant by reducing the pressure. This method will have a similar effect to increasing the film temperature, but the lower pressure will require an increase to the compressor power which result in a more inefficient system. When enhancing the heat transfer, it will often be beneficial to also increase the mass flow of the refrigerant if dry out is approached. It can also be necessary to examine the falling film to ensure there is still a good temperature difference and that the temperature does not drop below the freezing point.

In regards to improving the heat transfer coefficient there are a few methods for how that can be done. The first thing that can be improved is to avoid dry out, as it results in a noticeable decrease in heat transfer when present. Besides avoiding dry out, the most noticeable increase in heat transfer was seen by increasing the mass flow and by having ammonia as the refrigerant, which will increase the heat transfer coefficient from the wall to the refrigerant. The heat transfer coefficient for the falling film can also be improved by increasing the flow rate. However, besides changing the refrigerant the effect is not that large after dry out has been avoided.

In regards to improving the area there are three main methods for doing this. It will either be to increase the width, height or number of plates. Increasing the width increases the heat transfer as seen in the parametric analysis, but it is not as effective as increasing the height. While increasing the number of plates has not been investigated it will have a similar effect to increasing the width, assuming the total flow rates are not changed. So if the focus is just to increase the heat transfer the results shows that it is the most optimal to increase the height. However, if the pressure loss is also to be considered it is more complicated because increasing the width and number of plates will reduce the pressure loss while increasing the height will increase the pressure loss. Therefore when designing a pillow plate that fits to a given application there needs to considered several of the investigated parameters to achieve the required heat transfer without using an excessive amount of plate area. Also when optimising the design of a pillow plate, it is important to consider how the parameters affect each other. An example could be that if the height is increased, it could be beneficial to also increase the refrigerant flow rate to avoid dry out and increase the flow rate or temperature of the falling film to ensure the temperature difference does not become too small or that the water freezes. Therefore all the parameters needs to be considered when designing a pillow plate configuration to ensure a good design.

Chapter 8

Conclusion

Following the discussion made in the previous chapter, the model and its results have been examined. Thereby it is possible to evaluate the problem statement that was defined earlier in the project which is:

Create a parametric model of the heat transfer within a falling film pillow plate evaporator.

To be able to create this parametric model the first step was to determine the heat transfer from the falling film to the plate. This was done by evaluating the heat transfer through the total thermal resistance of the system. This was done by examining the convective heat transfer from the falling film to the plate, conduction through the plate and the two phase heat transfer from the plate to the refrigerant. The heat transfer from the falling film to the plate was modelled using a Nusselt correlation for vertical pipes and the conduction was modelled as conduction through a plate. The two phase was more complicated where the first step was to model the one phase heat transfer for pillow plates with a Nusselt correlation. Then this was enhanced with a correlation for boiling in vertical pipes and then there was added a coefficient to account for the nucleate boiling. There was also examined where in the pillow plate there would be dry out which meant the flow would be modelled as a one phase homogeneous flow. Thereby it was possible to account for all the heat transfer methods that will occur in the pillow plate by using relations developed for similar configuration.

The model this created is capable of evaluating the heat transfer for a given pillow plate and a given set of flow conditions. To examine the accuracy of the model it was compared to data provided by ATHCO who is a manufacture of pillow plates which shows that the model has an accuracy of $\pm 14\%$ where it tends to underestimate the heat transfer. This accuracy is completely acceptable considering the complexity of the system and the assumptions made. The general response of the model was also evaluated during the parametric study. Here it was found that it followed the expected behaviour where the heat transfer increased when there was added area, the average temperature difference increased or parameters was changed that should increase the heat transfer coefficients. From this it can be seen that there has been created a robust model that are able to model a flooded falling film evaporators with acceptable accuracy.

During the parametric study the effect of several of the input parameters was investigated. From this it could be concluded that to enhance the heat transfer of the pillow plate there are several parameters that needs to be considered. To have a large heat transfer the three important parameters are area, temperature difference and heat transfer coefficient. The area should be increased by increasing the height, but the width and number of plates can be considered if the pressure loss becomes to large. The temperature difference can either be ensured by a high inlet temperature or a large flow rate so the temperature reduction is small. An alternative is to reduce the saturation pressure, but that requires extra work for the compressor. To increase the heat transfer coefficient the most important thing is to avoid dry out which can mainly be done by increasing the mass flow of the refrigerant. Besides that it depends on the geometric parameters of the plate and the flow rate of the refrigerant and falling film. From the parametric analysis it could also be concluded that the model is versatile and able to realistically model several different configurations of pillow plates.

Chapter 9

Future Work

The work that has been done during this project has resulted in a model that are capable of determining the heat transfer of a pillow plate with comparable results to the method used by ATHCO. However this model relies on several assumption and have not been compared to experimental which reduces the applicability of the model. Therefore there can be done further work on this project which will be discussed chapter.

9.1 Experimental Work

Currently all the data the model has been compared to is theoretical which limit the certainty of the accuracy. However, improving the validity of the model is not the only reason to examine the model using experimental data. By conducting experiments it will be possible to make adjustment to the model so it can be examined if the dry out happens when it is predicted to happen, is the temperature distributed uniformly as assumed and so on. Then from these results the model can be adjusted and if there is enough experimental data there can even be developed new correlation for the falling film and two phase flow that was assumed to behave similar to pipes. The overall goal of doing this experimental study would be to improve both the validity of the model and the accuracy of its results. This will make the model more suitable for being used to actually design and optimise actual pillow plates.

9.2 Extend Versatility of Model

Besides doing the experimental study to improve the trust and accuracy it can also be used to improve the versatility of the model as it can add adjustment factor for when there is extrapolated for the currently used correlations. However this is not the only method for improving the versatility of the model. In this project there was selected several different correlations which was mostly selected based on covering a wide range and have a well documented accuracy. However, this still resulted in some of the correlations, especially the one phase correlation, not covering what was being examined. This could be improved by doing an extensive literature study and select several correlations to be implemented with a proper method for selecting the most suitable correlation. This can be combined with the experimental study, where the most accurate correlations can be selected. This will make the model more versatile which makes it possible to examine a large variety of pillow plate geometries and Reynolds numbers for the falling film and refrigerant.

Bibliography

ATHCO Enginering. Pillow plates, 2017. URL https://www.athco-engineering.dk/products/pillow-plates/.

M. Cooper. Heat flow rates in saturated nucleate pool boiling-a wide-ranging examination using reduced properties. *Advances in Heat Transfer*, 16:157–239, 1984. ISSN 0065-2717. doi:10.1016/S0065-2717(08)70205-3. URL https://www.sciencedirect.com/science/article/pii/S0065271708702053.

A. David, B. V. Mathiesen, H. Averfalk, S. Werner, and H. Lund. Heat roadmap europe: Large-scale electric heat pumps in district heating systems, 2017. URL https://vbn.aau.dk/ws/portalfiles/portal/281265126/energies_10_00578_v2.pdf.

D. Fjernvarme. Skal vi have glæde af overskudsvarme i danmark?, 2017. URL https://datacenterindustrien.dk/wp-content/uploads/Overskudsvarme-i-DK-overblik-og-fakta.pdf.

K. Jørgensen. Spildvarme fra industrien kan dække forbruget i 128.000 huse, 2018. URL https://ing.dk/artikel/spildvarme-industrien-kan-daekke-forbruget-128000-huse-211414.

M. Kind, D. Steiner, J. M. Chawla, J.-J. Schröder, Y. Saito, H. Auracher, O. Herbst, and A. Katsaounis. *H3 Flow Boiling*, pages 793–902. Springer Berlin Heidelberg, Berlin, Heidelberg, 2010. ISBN 978-3-540-77877-6. doi:10.1007/978-3-540-77877-6_124. URL https://doi.org/10.1007/978-3-540-77877-6_124.

M.Piper, A. Zibarta, and E. Kenig. New design equations for turbulent forced convection heat transfer and pressure loss in pillow-plate channels. *International Journal of Thermal Sciences*, 120:459–468, 2017. doi:10.1016/j.ijthermalsci.2017.06.012.

nuclear power.net. Flow boiling – forced convection boiling, 2021. URL https://www.nuclear-power.net/nuclear-engineering/heat-transfer/boiling-and-condensation/flow-boiling-forced-convection-boiling/. Last accessed: 23-05-2021.

B. S. Petukhov. Theoretical calculation of heat exchange and frictional resistance in turbulent flow in tubes of an incompressible fluid with variable physical properties. *Teplofiz. Vysok. Temperatur (High Temperature Heat Physics)*, 1, 1963. URL https://ci.nii.ac.jp/naid/10011325838/en/.

M. Piper, A. Olenberg, J. Tran, and E. Kenig. Determination of the geometric design parameters of pillow-plate heat exchangers. *Applied Thermal Engineering*, 91:1168–1175, 2015. ISSN 1359-4311. doi:10.1016/j.applthermaleng.2015.08.097. URL https://www.sciencedirect.com/science/article/pii/S1359431115008947.

W. Roetzel and B. Spang. *C1 Thermal Design of Heat Exchangers*, pages 31–66. Springer Berlin Heidelberg, Berlin, Heidelberg, 2010. ISBN 978-3-540-77877-6. doi:10.1007/978-3-540-77877-6_4. URL https://doi.org/10.1007/978-3-540-77877-6_4.

G. Schnabel. *M3 Heat Transfer to Falling Films at Vertical Surfaces*, pages 1287–1294. Springer Berlin Heidelberg, Berlin, Heidelberg, 2010. ISBN 978-3-540-77877-6. doi:10.1007/978-3-540-77877-6_96. URL https://doi.org/10.1007/978-3-540-77877-6_96.

M. Shirzad, M. A. Delavar, S. S. M. Ajarostaghi, and K. Sedighi. Evaluation the effects of geometrical parameters on the performance of pillow plate heat exchanger. *Chemical Engineering Research and Design*, 150:74–83, 2019. ISSN 0263-8762. doi:https://doi.org/10.1016/j.cherd.2019.06.032. URL https://www.sciencedirect.com/science/article/pii/S0263876219303235.

Skjern Paper. Bæredygtighedsrapport 2019, 2019. URL https://issuu.com/oerskovweb/docs/26103_-b_redygtighedsrapport_2019_dk_ipaper?fr=sZjM1MzkwNDEzNw.

State of Green. DISTRICT ENERGY, 2018. URL https://stateofgreen.com/en/uploads/2018/08/SoG_WhitePaper_DistrictEnergy_210x297_V22_WEB.pdf?time=1613140703.

R. H. Turner, Y. A. Cengel, J. M. Cimbala, and M. Kanoglu. *Fundamentals of Thermal-Fluid Sciences*, volume 1. Mc Graw Hill, 4 edition, 2012. ISBN 978-007-132511-0.

P. Vassallo and K. Keller. Two-phase frictional pressure drop multipliers for suva r-134a flowing in a rectangular duct. *International Journal of Multiphase Flow*, 32(4): 466–482, 2006. ISSN 0301-9322. doi:10.1016/j.ijmultiphaseflow.2006.01.004. URL https://doi.org/10.1016/j.ijmultiphaseflow.2006.01.004.

P. B. Whalley. *M3 Heat Transfer to Falling Films at Vertical Surfaces*, pages 16–26. Oxford University Press, 1996. ISBN 0-19-856444-9. doi:10.1002/apj.5500050213. URL https://doi.org/10.1002/apj.5500050213.

L. Wojtan, T. Ursenbacher, and J. R. Thome. Investigation of flow boiling in horizontal tubes: Part ii—development of a new heat transfer model for stratified-wavy, dryout and mist flow regimes. *International Journal of Heat and Mass Transfer*, 48(14):2970–2985, 2005. ISSN 0017-9310. doi:10.1016/j.ijheatmasstransfer.2004.12.013. URL `https://www.sciencedirect.com/science/article/pii/S001793100500027X`.

A. Zhao, Y. Fan, Y. Suzuki, and K. Morimoto. Flow boiling heat transfer of low-gwp working fluids at low mass and heat fluxes in a 4 mm diameter tube. *International Journal of Heat and Mass Transfer*, 162:120332, 2020. ISSN 0017-9310. doi:https://doi.org/10.1016/j.ijheatmasstransfer.2020.120332. URL `https://www.sciencedirect.com/science/article/pii/S0017931020332683`.

Appendix A

Matlab Code

Main Program

```
%Model is made by Lasse Olsen, based on:
%https://www.sciencedirect.com/science/article/pii/S1290072917302363(Mainly this for pillow plate)
%https://link-springer-com.zorac.aub.aau.dk/referenceworkentry/10.1007/978-3-540-77877-6_96(Mainly this for falling film)
%https://www.sciencedirect.com/science/article/pii/S1359431115008947(Characteristic dimmensions of pillow plates)
%https://www.sciencedirect.com/science/article/pii/S001793101530627X
%https://www.sciencedirect.com/science/article/pii/S135943111832492X

%This is overall model that contains the input parameters for the pillow
%plate and the fluids and then calls the sub functions as needed.
clc
clear
close

%% Parameters Plates:
Height = 1;                                  %Height of plate
Width = 2;                                    %Width of plate
Nr_plates = 20;                              %Number of plate
```

```
Sl = 30*10^-3;                    %Longtitude distance between welding spots
St = 35*10^-3;                    %Transverse distance between welding spots
Sd = sqrt((0.5*St)^2+Sl^2);       %Direct distance between centre of tubes
d_sp = 7.5*10^-3;                 %Diameter of welding spots (not in data sheet)
delta_in = 5*10^-3;               %Inner expansion heigt
delta_out = 45*10^-3;             %Distance between plates (centre to centre)
thickness = 1.5*10^(-3);          %Thickness of pillow plate
w_e=15*10^(-3);                   %Width of edge
k_plate= 14.2;                    %Thermal conductivity of pillow plate
A_tot=Height*Width*Nr_plates*2;   %Area of pillow plates, is simplified to be the same as flat plate
H_baffle=Height/2;                %Heigth from bottom that the baffle is placed at
Baffle_extension=0;               %Percentage of the total width the baffle covers
W_baffle=Width*Baffle_extension;  %Width of baffle

%% Parameter Refrigerant
Refrigerant_fluid = 'NH3';        %Refrigerant fluid used
T_ref_in = -5+273.15;             %Temperature of refrigerant in
T_ref_out = -5+273.15;            %Temperature of refrigerant out
%P_ref_in = 33.91*10^5;           %Pressure of refrigerant in
m_dot_ref = 5;                    %Mass flow of refrigerant
m_dot_ref_evap = 1.2;             %Mass flow of refrigerant that evaporated (Thus will later be calculated, when two phase is added)

%% Parameter Waste heat
```

```
Heat_fluid = 'water';                %Fluid used for heating
T_film_in = 5+273.15;                %Temperature of heating fluid in
T_film_out = 1+273.15;               %Temperature of heating fluid out
P_heat_in = 101325;                  %Pressure of heating fluid in
m_dot_film = 30;                     %Mass flow of heating media
g=9.81;                              %Gravitational acceleration

%Here the sub program to calculate the convective heat transfer coefficient for the pillow plate. This is done for both the refrigerant on gas form and on liquid form. 1 represent gas, while 2 represent liquid
[h_pp(1),Re_tot(1),u_m(1),d_hi,a,b,c,A_csitot]=HeatTransferPP(Sl,St,Sd,delta_in,Width,Height,Nr_plates,thickness,d_sp,1,T_ref_in,Refrigerant_fluid,m_dot_ref,w_e);
[h_pp(2),Re_tot(2),u_m(2),d_hi,a,b,c,A_csitot]=HeatTransferPP(Sl,St,Sd,delta_in,Width,Height,Nr_plates,thickness,d_sp,0,T_ref_in,Refrigerant_fluid,m_dot_ref,w_e);

%Here the sub program to calculate the convective heat transfer coefficient for the falling film.
[h_film,delta_pp_eq1,delta_pp_eq2]=HeatTransferFF(P_heat_in,T_film_in,Heat_fluid,m_dot_film,Width,Nr_plates,g,T_ref_in);

Nr_cell_vertical=1000; %Number of cells the heat exchanger is split into on the vertical axis
%Call the subprogram that calculates the heat transfer, the amount
%evaporated, the temperature of the film and the gradual change in quality.
[m_dot_evap_ref,x_dot_ref,Q_calc_NTU,T_film,k,err,h_boil,alpha_nuc,x_crit,alpha_void,x_critical,q_crit,q_dot]=HeatExchangerNoBaffleVert(m_dot_film,m_dot_ref,h_pp,h_film,Refrigerant_fluid,P_heat_in,T_film_in,Heat_fluid,Nr_cell_vertical,Nr_plates,Height,Width,T_ref_in,thickness,k_plate,d_hi,A_csitot,a,b,c);
```

```
m_dot_evap_ref_tot=sum(m_dot_evap_ref); %Total amount evaporated in the pillow plate.
Q_tot=sum(Q_calc_NTU); %Total amount of heat transfer.

%Loop for calculating the pressure loss, this is calculated on a cell basis
%as it is dependent on the quality of the mixture.
for i=1:Nr_cell_vertical
[Delta_P_grav_2ph(i),Delta_P_acc_2ph(i),Delta_P_Friedel(i)]=PressureLossPP(Width,Height,a,b,c,Re_tot,u_m,d_hi,T_ref_in,Refrigerant_fluid,m_dot_ref,x_dot_ref(i),Nr_plates,A_csitot,Nr_cell_vertical);
if i>2
    Delta_P_acc_2ph_actual(i-1)=Delta_P_acc_2ph(i)-Delta_P_acc_2ph(i-1); %Account differentation in equation
end
end
Delta_P_tot=sum(Delta_P_Friedel+Delta_P_grav_2ph)+sum(Delta_P_acc_2ph_actual); %Total presure loss through the evaporator


length_plot=linspace(0,Height,Nr_cell_vertical);
length_plot_film=linspace(0,Height,Nr_cell_vertical+1);

figure(1)
plot(length_plot_film,x_dot_ref)
xlabel('Lenght [m]')
ylabel('Quality [-]')
grid on
%matlab2tikz('QualityStandard.tikz')
%title('Quality of Refrigerant from Inlet to Outlet')

% figure(2)
% plot(x_dot_ref(2:Nr_cell_vertical+1),h_boil)
% xlabel('Quality [-]')
% ylabel('Convective heat transfer coefficient [W/(m^2 K)]')
% title('Convective/Boiling Heat Transfer Coeffient on Inside of PP')

```

```
% figure(3)
% plot(length_plot,Q_calc_NTU)
% xlabel('Lenght [m]')
% ylabel('Heat Transfer [W]')
% title('Heat Transfer Rate of Each Cell From Inlet to Outlet
   ')

figure(2)
plot(length_plot_film,T_film)
xlabel('Lenght [m]')
ylabel('Temperature[K]')
grid on
%matlab2tikz('TFilmStandard.tikz')
%title('Temperature of Falling Film from Top to Bottom')

figure(3)
plot(length_plot_film,q_dot)
xlabel('Lenght [m]')
ylabel('Heat Flux [W/m^2]')
grid on
%matlab2tikz('qdotStandard.tikz')
%title('Amount Evaporated within Each Cell From Inlet to
   Outlet')
```

Pillow Plate Convective Heat Transfer

```
function [h_pp,Re_tot,u_m,d_hi,a,b,c,A_csitot]=HeatTransferPP
   (Sl,St,Sd,delta_in,Width,Height,Nr_plates,thickness,d_sp,
   phase,T_ref_in,Refrigerant_fluid,m_dot_ref,w_e)

%Calculates the convective heat transfer within the pillow
   plate based on
%https://www.sciencedirect.com/science/article/pii/
   S1290072917302363 and
%https://www.sciencedirect.com/science/article/pii/
   S1359431115008947

h_ref(1)=py.CoolProp.CoolProp.PropsSI('H','T',T_ref_in,'Q',
   phase,Refrigerant_fluid);          %Enthalpy of refrigerant
    at inlet, currently unused
Pr_ref(1)=py.CoolProp.CoolProp.PropsSI('PRANDTL','T',T_ref_in
```

```
    ,'Q',phase,Refrigerant_fluid);      %Prandtl number of refrigerant
rho_ref(1)=py.CoolProp.CoolProp.PropsSI('D','T',T_ref_in,'Q',phase,Refrigerant_fluid);         %Density of refrigerant
mu_ref(1)=py.CoolProp.CoolProp.PropsSI('viscosity','T',T_ref_in,'Q',phase,Refrigerant_fluid);    %Dynamic viscosity of refrigerant
cp_ref(1)=py.CoolProp.CoolProp.PropsSI('C','T',T_ref_in,'Q',phase,Refrigerant_fluid);          %Specific heat capacity of refrigerant
k_ref(1)=py.CoolProp.CoolProp.PropsSI('conductivity','T',T_ref_in,'Q',phase,Refrigerant_fluid); %Thermal conductivity of refrigerant
%T_ref_ss=py.CoolProp.CoolProp.PropsSI('T','T',T_ref_in,'Q',phase,Refrigerant_fluid); %Thermal conductivity of refrigerant
%% One phase pillow plate Nusselt

%Calculated lengths:
a=2*Sl/St;                 %Dimmensionless length for longitudinal spacing
b=d_sp/St;                 %Dimmensionless length for diameter of welding spots
c=delta_in/St;             %Dimmensionless length for inner channel height
A0=Sl*0.5*St;              %Flat area of smallest repeating piece


%Calculating total RE
a_Vnref=0.1*(St/(2*Sl))^2-0.18*(St/(2*Sl))+0.19;     %Fitting factor
a_wnref=3.12*(St/(2*Sl))^2-5.74*(St/(2*Sl))+3.08;    %Fitting factor
phi_A=1-pi*d_sp^2/(4*St*Sl);                          %Parameter that account for welding spot affect on area
f_sp=1.37*phi_A^2.58;                                 %Power law fitting to account for welding spots
V_i=a_Vnref*delta_in*Sd^2*f_sp;                       %Inner volume of the smallest repeating piece in the geometry
A_wi=A0+A0*a_wnref*delta_in^2/Sd^2;                   %Inner
```

```
    wetting area of the smallest repeating piece
d_hi=4*V_i/A_wi;                                    %Hydraulic diameter of the pillow plate
A_csi=V_i/(Sl);                                     %Average cross sectional area of the smallest repeating piece
A_csitot=A_csi*4*((Width-2*w_e)/St);                %Total cross sectional area while accounting for baffle (Calculated for below baffle but are the same the other places)


%If staments to determine which type of pillow plate it is and thereby
%which curve fit should be used
if a>=0.58-0.05 && a<=0.58+0.05 && b<=0.14 && b>=0.1 && c<=0.083 && c>=0.042 %If this is true the pillow plate is of type L
    n6=4.36*c+1.14;                             %Part of power law for fanning factor
    n7=-0.44;                                   %Part of power law for fanning factor
    psi_A=0.94*b+0.4;                           %Dimensionless ratio for area of recirculation zone
    psi_Q=2.16*b+(4.23*c-0.352);                %Dimensionless ratio for heat transfer of recirculation zone
    dh_z1=(-11.22*b+(113*c+1.82))*10^-3;        %Hydraulic diameter of core flow
    s_star=1;                                   %Correction factor for Reynolds number
elseif a>=1-0.05 && a<=1+0.05 && b<=0.24 && b>=0.17 && c<=0.143 && c>=0.071  %If this is true the pillow plate is of type E
    n6=2.52*c+0.24;                             %Part of power law for fanning factor
    n7=-0.3;                                    %Part of power law for fanning factor
    psi_A=0.75*b+0.46;                          %Dimensionless ratio for area of recirculation zone
    psi_Q=0.75*b+(1.54*c-0.014);                %Dimensionless ratio for heat transfer of recirculation zone
    dh_z1=(-18.31*b+(35.42*c+4.8))*10^-3;       %Hydraulic
```

```
        diameter of core flow
    s_star=1;                                    %Correction factor for Reynolds number
elseif a>=1.71-0.05 && a<=1.71+0.05 && b<=0.24 && b>=0.17 && c<=0.143 && c>=0.071 %If this is true the pillow plate is of type T
    n6=4.62*c+0.6;                               %Part of power law for fanning factor
    n7=-0.34;                                    %Part of power law for fanning factor
    psi_A=0.81*b+0.263;                          %Dimensionless ratio for area of recirculation zone
    psi_Q=0.46*b+(1.17*c-0.042);                 %Dimensionless ratio for heat transfer of recirculation zone
    dh_z1=(-8.1*b+(60*c+2.1))*10^-3;             %Hydraulic diameter of core flow
    s_star=1.0761;                               %Correction factor for Reynolds number
else %This specifies that the parameters are not within the range the study examined, this should generally be avoided
    n6=4.62*c+0.6;                               %Part of power law for fanning factor
    n7=-0.34;                                    %Part of power law for fanning factor
    psi_A=0.81*b+0.263;                          %Dimensionless ratio for area of recirculation zone
    psi_Q=0.46*b+(1.17*c-0.042);                 %Dimensionless ratio for heat transfer of recirculation zone
    dh_z1=(-8.1*b+(60*c+2.1))*10^-3;             %Hydraulic diameter of core flow
    s_star=1.0761;                               %Correction factor for Reynolds number
    disp('Pillow plate parameters are out of range')
end


u_m=m_dot_ref/(rho_ref*A_csitot*Nr_plates);          %Average velocity of the refrigerant through the plate
Re_tot=rho_ref*u_m*d_hi/mu_ref;                      %Total Reynolds number through the plate
Re_z1=Re_tot*(s_star/(1-psi_A));                 %Reynolds number
```

```
for coreflow
zeta_f=n6*Re_tot^n7;                          %Fanning factor power law for core flow
if Pr_ref>5
Nu_ref=((zeta_f/8)*Re_z1*Pr_ref)/(1.07+12.7*sqrt(zeta_f/8)*(Pr_ref^(2/3)-1));    %Nusselt number for core flow
else
Nu_ref=((zeta_f/8)*Re_z1*Pr_ref)/(1+3.4*zeta_f+(11.7+1.8*Pr_ref^-(1/3))*sqrt(zeta_f/8)*(Pr_ref^(2/3)-1));
end
h_z1=Nu_ref*k_ref/dh_z1;                      %Convective heat transfer coefficient for core flow
h_pp=h_z1*((1-psi_A)/(1-psi_Q));              %Convective heat transfer coefficient for inner flow of pillow plate
end
```

Falling Film Convective Heat Transfer

```
function [h_film,delta_pp_eq1,delta_pp_eq2]=HeatTransferFF(P_heat_in,T_film_in,Heat_fluid,m_dot_film,Width,Nr_plates,g,T_ref_in)

%Calculates the convective heat transfer for the falling film based on
%https://link-springer-com.zorac.aub.aau.dk/referenceworkentry/10.1007/978-3-540-77877-6_96

%Heating media
if T_ref_in<273.15
   T_ref_in=273.153;
end
mu_film=py.CoolProp.CoolProp.PropsSI('viscosity','P',P_heat_in,'T',T_film_in,Heat_fluid);       %Dynamic viscosity of heating media
Pr_film=py.CoolProp.CoolProp.PropsSI('PRANDTL','P',P_heat_in,'T',T_film_in,Heat_fluid);          %Prandtl number of heating media
k_film=py.CoolProp.CoolProp.PropsSI('conductivity','P',P_heat_in,'T',T_film_in,Heat_fluid);     %Thermal conductivity of heating media
rho_film=py.CoolProp.CoolProp.PropsSI('D','P',P_heat_in,'T',
```

```
T_film_in,Heat_fluid);                %Desity of heating media
nu_film=mu_film/rho_film;                                    %Kinematic viscosity of heating media
mu_wall=py.CoolProp.CoolProp.PropsSI('viscosity','P',P_heat_in,'T',T_ref_in,Heat_fluid);          %Viscosity at wall temperature
sigma_film=74.01*10^(-9);                                    %Surface tension of water

%% Falling Film Nusselt
Re_film=m_dot_film/(Width*Nr_plates*2*mu_film);       %Reynolds number for falling film

C_inf=1.3;                                                    %Factor for Nusselt relation, based on constant wall temperature
wetting_rate=m_dot_film/(Width*Nr_plates*2);
Nu1_film(1)=C_inf*Re_film^(-1/3);                             %Nusselt relation for laminar, hydrodynamically and thermally developed flow
Nu1_film(2)=0.0425*Re_film^(1/5)*Pr_film^(0.344);     %Nusselt relation for transition to turbulent flow
Nu1_film(3)=0.0136*Re_film^(2/5)*Pr_film^(0.344);     %Nusselt relation for turbulent flow

Nu_film=max(Nu1_film)*(mu_film/mu_wall)^0.25;         %Select highest Nusselt number, as the highest will always be based on Re number being in that zone
h_film=Nu_film*k_film*(1/(nu_film^2/g)^(1/3));       %Convective heat transfer coefficient for falling film

h_plus=3.8*10^(-3)*Re_film^(0.4)*Pr_film^(0.65);      %Another relation for falling film, is generally quite a bit lower compared to the other one
h_film1=h_plus/(mu_film^2/(rho_film^2*k_film^3*g))^(1/3)*(mu_film/mu_wall)^0.25;

%% Film thickness (This is about the thickness of the film layer, is currently not important)
```

```
delta_pp_eq1=0.0225*(nu_film^2/g)^(1/3)*(Re_film)+3.97*(
    nu_film^2/g)^(1/3);
delta_pp_eq2=0.573*(nu_film^2/g)^(1/3)*(Re_film)^(0.5175);

delta_flat_flam=(nu_film^2/g)^(1/3)*(3*Re_film)^(1/3);
delta_flat_fturb=0.302*(3*nu_film^2/g)^(1/3)*(Re_film)^(8/15)
    ;

K_f=rho_film*sigma_film^3/(g*mu_film^4);
Ka=1/K_f;

end
```

Heat Exchanger Function

```
function [m_dot_evap_ref,x_dot_ref,Q_calc_NTU,T_film,k,err,
    h_pp_boil,alpha_nuc,x_crit,alpha_void,x_critical,q_crit,
    q_dot]=HeatExchangerNoBaffleVert(m_dot_film,m_dot_ref,h_pp
    ,h_film,Refrigerant_fluid,P_heat_in,T_film_in,Heat_fluid,
    Nr_cell_vertical,Nr_plates,Height,Width,T_ref_in,thickness
    ,k_plate,d_hi,A_csitot,a,b,c)

%Calculates the actual heat transfer of the pillow plate and
    the resulting
%change in quality for the mixture. Heat exchanger modelling
    is based on
%https://link-springer-com.zorac.aub.aau.dk/
    referenceworkentry/10.1007/978-3-540-77877-6_4
%while the boiling enhancement is from
%https://link-springer-com.zorac.aub.aau.dk/
    referenceworkentry/10.1007/978-3-540-77877-6_124#Sec35

%Refrigerant gas
h_ref(1)=py.CoolProp.CoolProp.PropsSI('H','T',T_ref_in,'Q',1,
    Refrigerant_fluid);         %Enthalpy of refrigerant at
    inlet, currently unused
Pr_ref(1)=py.CoolProp.CoolProp.PropsSI('PRANDTL','T',T_ref_in
    ,'Q',1,Refrigerant_fluid);       %Prandtl number of
    refrigerant
rho_ref(1)=py.CoolProp.CoolProp.PropsSI('D','T',T_ref_in,'Q'
    ,1,Refrigerant_fluid);            %Density of refrigerant
```

```
mu_ref(1)=py.CoolProp.CoolProp.PropsSI('viscosity','T',T_ref_in,'Q',1,Refrigerant_fluid);   %Dynamic viscosity of refrigerant
cp_ref(1)=py.CoolProp.CoolProp.PropsSI('C','T',T_ref_in,'Q',1,Refrigerant_fluid);           %Specific heat capacity of refrigerant
k_ref(1)=py.CoolProp.CoolProp.PropsSI('conductivity','T',T_ref_in,'Q',1,Refrigerant_fluid); %Thermal conductivity of refrigerant
P_ref_in=py.CoolProp.CoolProp.PropsSI('P','T',T_ref_in,'Q',1,Refrigerant_fluid); %Thermal conductivity of refrigerant
sigma_ref=py.CoolProp.CoolProp.PropsSI('I','T',T_ref_in,'Q',0,Refrigerant_fluid);
%Refrigerant liquid
h_ref(2)=py.CoolProp.CoolProp.PropsSI('H','T',T_ref_in,'Q',0,Refrigerant_fluid);           %Enthalpy of refrigerant at inlet, currently unused
Pr_ref(2)=py.CoolProp.CoolProp.PropsSI('PRANDTL','T',T_ref_in,'Q',0,Refrigerant_fluid);      %Prandtl number of refrigerant
rho_ref(2)=py.CoolProp.CoolProp.PropsSI('D','T',T_ref_in,'Q',0,Refrigerant_fluid);           %Density of refrigerant
mu_ref(2)=py.CoolProp.CoolProp.PropsSI('viscosity','T',T_ref_in,'Q',0,Refrigerant_fluid);   %Dynamic viscosity of refrigerant
cp_ref(2)=py.CoolProp.CoolProp.PropsSI('C','T',T_ref_in,'Q',0,Refrigerant_fluid);           %Specific heat capacity of refrigerant
k_ref(2)=py.CoolProp.CoolProp.PropsSI('conductivity','T',T_ref_in,'Q',0,Refrigerant_fluid); %Thermal conductivity of refrigerant
%Heating media
mu_film=py.CoolProp.CoolProp.PropsSI('viscosity','P',P_heat_in,'T',T_film_in,Heat_fluid);         %Dynamic viscosity of heating media
Pr_film=py.CoolProp.CoolProp.PropsSI('PRANDTL','P',P_heat_in,'T',T_film_in,Heat_fluid);           %Prandtl number of heating media
k_film=py.CoolProp.CoolProp.PropsSI('conductivity','P',P_heat_in,'T',T_film_in,Heat_fluid);       %Thermal conductivity of heating media
```

```
cp_film=py.CoolProp.CoolProp.PropsSI('C','P',P_heat_in,'T',T_film_in,Heat_fluid);                    %Specific heat capacity of heating media
rho_film=py.CoolProp.CoolProp.PropsSI('D','P',P_heat_in,'T',T_film_in,Heat_fluid);                  %Desity of heating media
nu_film=mu_film/rho_film;                                                                            %Kinematic viscosity of heating media
sigma_film=74.01*10^(-9);%py.CoolProp.CoolProp.PropsSI('I','P',P_heat_in,'T',T_film_in,Heat_fluid);

Mol_mass_ref=py.CoolProp.CoolProp.PropsSI('M','T',T_ref_in,'Q',1,Refrigerant_fluid);                 %
Mol_mass_H2=py.CoolProp.CoolProp.PropsSI('M','T',T_ref_in,'Q',1,'Hydrogen');
p_crit=py.CoolProp.CoolProp.PropsSI('pcrit','T',T_ref_in,'Q',1,Refrigerant_fluid);
G=m_dot_ref/(A_csitot*Nr_plates);

%% Heat exchange cell method
alpha_gas=h_pp(1); %Define the heat transfer for when it would be only gas
alpha_liq=h_pp(2); %Define the heat transfer for when it would be only liquid
W1(1:Nr_cell_vertical)=inf;%m_dot_ref*cp_ref(1)/Nr_plates;                              %Heat capacity rate for refrigerant
W2(1:Nr_cell_vertical)=m_dot_film*cp_film/Nr_plates;      %Heat capacity rate for heating media
A(1:Nr_cell_vertical)=2*Height*Width/Nr_cell_vertical;                              %Area of cell



F_value=1;  %This is set to one as it is pure counter flow
x_dot_ref_guess=linspace(0,1,Nr_cell_vertical+1); %Starting guess for how the refrigerant quality changes through the pillow plate
x_dot_ref=x_dot_ref_guess;  %Define the value to be equal to the guess
```

```
%Initial temperature as vectors
T_ref(1)=T_ref_in;        %Temperature of refrigerant at inlet of zone 1
T_film(1)=T_film_in;      %Temperature of heating media at inlet of zone 3
T_guess=T_film(1)-linspace(0,2,Nr_cell_vertical+1);      %Guess for temperature distripution for falling film
T_film=T_guess;           %Define the guess as the value
err(1)=1;                 %Define error value for iteration
T_ref(1:Nr_cell_vertical+1)=T_ref; %Define the refrigerant temperature to be constant for all cell
iter=1; %Define iteration counter
k(1:Nr_cell_vertical)=(1/(h_pp(2))+1/h_film+thickness/k_plate)^(-1);     %Give starting value for k value
q_dot(1)=k(1)*(T_film(Nr_cell_vertical+1)-T_ref(1));     %Calculate start guess for heat flow
x_critical=0.8;     %Gives guess for when dryout happens, is not important
for i=1:Nr_cell_vertical
[x_crit(i),q_crit(i)]=CritBoil(sigma_ref,rho_ref,Height,m_dot_ref,d_hi,h_ref,Nr_plates,i,Nr_cell_vertical,A_csitot);     %Calculate critical heat flux and quality
end

while err(iter)>1*10^-8 && iter<10000          %Iteration loop to correct the guess for inlet temperature
crit_boil=0;
for i=1:Nr_cell_vertical %The heat transfer is calculated for each

if x_dot_ref(i)<0.995    %Check to see if the fluid can be considered to still be boiling
BoilingFactor(i)=((1-x_dot_ref(i))^0.01*((1-x_dot_ref(i))^1.5+1.9*x_dot_ref(i)^0.6*(rho_ref(2)/rho_ref(1))^0.35)^(-2.2)+x_dot_ref(i)^0.01*((alpha_gas/alpha_liq)*(1+8*(1-x_dot_ref(i))^0.7)*(rho_ref(2)/rho_ref(1))^0.67)^(-2))^(-0.5); %Vertical enchancement of convective heat transfer
%BoilingFactor(i)=((1-x_dot_ref(i))^0.01*((1-x_dot_ref(i))
```

```
    +1.2*x_dot_ref(i)^0.4*(rho_ref(2)/rho_ref(1))^0.37)^(-2.2)+x_dot_ref(i)^0.01*((alpha_gas/alpha_liq)*(1+8*(1-x_dot_ref(i))^0.7)*(rho_ref(2)/rho_ref(1))^0.67)^(-2))^(-0.5); %Horionzontal enchancement of convective heat transfer
if i==1
     alpha_nuc(i)=HeatTransferNucleate(P_ref_in,d_hi,Refrigerant_fluid,T_film(Nr_cell_vertical+1-i),T_ref_in,k(i),Mol_mass_ref,Mol_mass_H2,p_crit);  % Calculate nucleate heat transfer
else
     alpha_nuc(i)=HeatTransferNucleate(P_ref_in,d_hi,Refrigerant_fluid,T_film(Nr_cell_vertical+1-i),T_ref_in,k(i-1),Mol_mass_ref,Mol_mass_H2,p_crit); % Calculate nucleate heat transfer
end
S=(1-x_dot_ref(i)*(1-rho_ref(2)/rho_ref(1)))^(1/2); %Slip factor
alpha_void(i)=1/(1+(S*(1-x_dot_ref(i))/x_dot_ref(i))*rho_ref(1)/rho_ref(2)); %Void fraction
   %Heat transfer coefficient fully wetted surface from VDI heat atlas


mu_V(i)=mu_ref(1)*x_dot_ref(i)+mu_ref(2)*(1-x_dot_ref(i));  % Dynamic viscosity of refrigerant as a homogeneous mixture
k_V(i)=k_ref(1)*x_dot_ref(i)+k_ref(2)*(1-x_dot_ref(i));      % Thermal conductivity of refrigerant as a homogeneous mixture
cp_V(i)=cp_ref(1)*x_dot_ref(i)+cp_ref(2)*(1-x_dot_ref(i));  % Heat capacity of refrigerant as a homogeneous mixture
Pr_V(i)=mu_V(i)*cp_V(i)/k_V(i);                               % Prandtl number of refrigerant as a homogeneous mixture
h_pool(i)=55*(P_ref_in/p_crit)^0.12*(-log10(P_ref_in/p_crit))^(-0.55)*(Mol_mass_ref*1000)^(-0.5)*q_dot(i)^0.67; %Pool boiling/Nucleate boiling heat transfer from Cooper relation
h_tp_test(i)=h_pool(i)+5000*(G*(1-x_dot_ref(i)))^0.2*(Mol_mass_ref*1000)^(-0.5);
   %Heat transfer from relation developed for small flow rates
```

```
h_dryout(i)=DryoutBoilingHeatTransfer(x_dot_ref(i),d_hi,rho_ref,G,Pr_V(i),mu_V(i),x_critical,h_tp_test(i),k_ref,a,b,c,alpha_void(i)); %Heat transfer after dryout has happened
h_boil(i)=sqrt((BoilingFactor(i)*h_pp(2))^2+h_pool(i)^2);

if q_dot(i)>q_crit(i) && crit_boil~=1   %Determine if there is critical boiling
    crit_boil=1;
    x_critical=x_dot_ref(i);
end
if crit_boil~=1 %Determine mode of heat transfer
     h_pp_boil(i)=h_boil(i);     %Select relation to use
     %h_pp_boil(i)=h_tp_test(i);

else
     h_pp_boil(i)=h_dryout(i);
end
else
     h_pp_boil(i)=h_pp(1);    %In the case of high quality a single phase heat transfer mode is used
end

k(i)=(1/(h_pp_boil(i))+1/h_film+thickness/k_plate)^(-1);     %Overall heat transfer koefficient
NTU_ref(i)=k(i)*A(i)/(W1(i));     %Number of transfer units for refrigerant in the given zone
NTU_film(i)=k(i)*A(i)/(W2(i));     %Number of transfer units for heating media in the given zone
R1(i)=W1(i)/W2(i);     %Ratio of heat capacity rate for given zone
R2(i)=1/R1(i);     %Ratio of heat capacity rate for given zone



P1(i)=(1-exp((R1(i)-1)*NTU_ref(i)*F_value))/(1-R1(i)*exp((R1(i)-1)*NTU_ref(i)*F_value));  %Dimensionless temperature 1 for zone 1
P2(i)=(1-exp((R2(i)-1)*NTU_film(i)*F_value))/(1-R2(i)*exp((R2
```

```
(i)-1)*NTU_film(i)*F_value));%Dimensionless temperature 2 for zone 1



T_film(Nr_cell_vertical+2-i)=T_film(Nr_cell_vertical+1-i)+P2(i)*(T_ref(i)-T_film(Nr_cell_vertical+1-i)); %Temperature of heating media at outlet of zone 3/Inlet of zone 1


Q_calc_NTU(i)=m_dot_film*cp_film*(T_film(Nr_cell_vertical+1-i)-T_film(Nr_cell_vertical+2-i)); %Calculate heat transfer as the amount of heat that has been released from the film in that cell
q_dot(i+1)=Q_calc_NTU(i)/(A(i)*Nr_plates);
m_dot_evap_ref(i)=Q_calc_NTU(i)/(h_ref(1)-h_ref(2)); %Calculate the amount of refrigerant that can be evaporated for the amount of heat transfered
x_dot_ref(i+1)=x_dot_ref_guess(i+1)+0.05*(sum(m_dot_evap_ref(1:i))/m_dot_ref-x_dot_ref_guess(i+1));    %Calculate the new quality based on the amount evaporated until now, there is added some dampning for better convergence.
if x_dot_ref(i+1)>1 %Ensures that the quality newer goes above one, will add something that allows for super heating at some point
    x_dot_ref(i+1)=1;
end
end





err(iter+1)=max(abs(x_dot_ref_guess(1:Nr_cell_vertical)-x_dot_ref(1:Nr_cell_vertical)))/mean(x_dot_ref);       %Calculate error between guessed value and calculated value (make to relative)
x_dot_ref_guess=x_dot_ref;                                      %Write calculated value as new guess
iter=iter+1;     %Add one to the iteration counter
end
```

```
end
```

Critical Boiling Function

```
function [x_crit,q_crit]=CritBoil(sigma_ref,rho_ref,Height,
    m_dot_ref,d_hi,h_ref,Nr_plates,i,Nr_cell_vertical,A_csitot
    )

x_in=0; %Inlet quality
m_dot_flux=m_dot_ref/(Nr_plates*A_csitot); %Mass flux of
    refrigerant
%Dimmensionless numbers
sigma_star=sigma_ref*rho_ref(2)/((m_dot_flux)^2*Height/
    Nr_cell_vertical*i);
l_star=Height/Nr_cell_vertical*i/d_hi;
rho_star=rho_ref(1)/rho_ref(2);


C(1)=0.25; %Dimensionless number from length
C(2)=0.25+9*10^(-4)*(l_star-50); %Dimensionless number from
    length
C(3)=0.34; %Dimensionless number from length

%If statement to determine the correct number
if l_star<50
    C_use=C(1);
elseif l_star>=50 && l_star<=150
    C_use=C(2);
elseif l_star>150 && l_star<600
    C_use=C(3);
else
    disp('To high for the hydraulic diameter')
end

%Parameter to account for sub cooling
K(1)=1.043/(4*C_use*sigma_star^0.043);
K(2)=5/6*(0.0124+1/l_star)/(rho_star^0.13*sigma_star^(1/3));
K(3)=1.12*(1.52*sigma_star^0.223+1/l_star)/(rho_star^0.6*
    sigma_star^0.173);

%Dimensionless critical heat flux
```

```
q_star(1)=C_use*(sigma_star)^0.043*1/l_star;
q_star(2)=0.1*(rho_star)^0.133*(sigma_star)^(1/3)*1/(1+0.0031*l_star);
q_star(3)=0.098*(rho_star)^0.133*(sigma_star)^(0.433)*l_star^0.27/(1+0.0031*l_star);
q_star(4)=0.234*(rho_star)^0.513*(sigma_star)^(0.433)*l_star^0.27/(1+0.0031*l_star);
q_star(5)=0.0384*(rho_star)^0.6*(sigma_star)^(0.173)*1/(1+0.28*sigma_star^(0.233)*l_star);

%If statement to determine the critical heat flux
if rho_star<=0.15
    if q_star(1)<=q_star(2)
        q_star_use=q_star(1);
    else
        if q_star(2)<=q_star(3)
            q_star_use=q_star(1);
        else
            q_star_use=q_star(3);
        end
    end
    if K(1)>=K(2)
        K_use=K(1);
    else
        K_use=K(2);
    end
else
    if q_star(1)<=q_star(4)
        q_star_use=q_star(1);
    else
        if q_star(5)<=q_star(4)
            q_star_use=q_star(4);
        else
            q_star_use=q_star(5);
        end
    end
    if K(1)>=K(2)
        K_use=K(1);
    else
        if K(2)<=K(3)
        K_use=K(2);
```

```
        else
        K_use=K(3);
        end
    end
end

q_crit_n=q_star_use*((m_dot_flux)*(h_ref(1)-h_ref(2))); % Critical heat flux without sub cooling
q_crit=q_crit_n*(1-K_use*x_in); %Critical heat flux with sub cooling
x_crit=4*q_crit/((m_dot_flux)*(h_ref(1)-h_ref(2)))*(i*Height/Nr_cell_vertical)/d_hi+x_in; %Critical quality with uniform flux
end
```

Nucleate Heat Transfer Coefficient

```
function [alpha]=HeatTransferNucleate(P_ref_in,d_hi,Refrigerant_fluid,T_film,T_ref_in,k,Mol_mass_CO2,Mol_mass_H2,p_crit)

q0_dot=150*10^3;     %Table value for CO2, see VDI for other fluids
alpha_0=18.89*10^3; %Table value for CO2, see VDI for other fluids
d0=10^-2;            %Reference diamter
R_a=0.78*10^(-6);    %Surface value for plate, needs to be checked
R_a0=0.4*10^(-6);    %Reference surface value


p_red=P_ref_in/p_crit;  %Reduced pressure
n=0.8-0.1*10^(0.76*p_red);  %For inorcanic fluids and hydrocarbons
%n1=0.7-0.13*10^(0.48*p_red); %Coefficient for cryogenic fluids, not that relevant
C_F=0.435*(Mol_mass_CO2/Mol_mass_H2)^0.27; %Factor depending on material, can also be read from table, should not exceed 2.5
F_p=2.816*p_red^0.45+(3.4+1.7/(1-p_red^7))*p_red^3.7;    % Reduced pressure factor
```

```
F_d=(d0/d_hi)^0.4;                                          %Diameter factore
F_W=(R_a/R_a0)^0.133;                                       %Wall roughness factor
F_mx=1;                                                     %Quality and flow rate factor, is generally indepedent from this so is put to zero
q_dot=k*(T_film-T_ref_in);                                  %Heat flux
Nucleate_factor=C_F*(q_dot/q0_dot)^n*F_p*F_d*F_W*F_mx;      %Multiplication factor for reference heat transfer coefficient
alpha=Nucleate_factor*alpha_0;                              %Nucleate boiling heat transfer coefficient
end
```

Dry Out Heat Transfer Coefficient

```
%Based on: https://www.sciencedirect.com/science/article/pii/S001793100500027X
%Alternative https://www.sciencedirect.com/science/article/pii/S0017931017343557#b0070
function [h_dryout]=DryoutBoilingHeatTransfer(x_dot_ref,d_hi,rho_ref,G,Pr_V,mu_V,x_crit,h_tp,k_ref,a,b,c,alpha_void)
x=x_dot_ref;

k_v=k_ref(1);

if a>=0.58-0.05 && a<=0.58+0.05 && b<=0.14 && b>=0.1 && c<=0.083 && c>=0.042 %If this is true the pillow plate is of type L
    n6=4.36*c+1.14;                             %Part of power law for fanning factor
    n7=-0.44;                                   %Part of power law for fanning factor
    psi_A=0.94*b+0.4;                           %Dimensionless ratio for area of recirculation zone
    psi_Q=2.16*b+(4.23*c-0.352);                %Dimensionless ratio for heat transfer of recirculation zone
    dh_z1=(-11.22*b+(113*c+1.82))*10^-3;        %Hydraulic diameter of core flow
```

```
    s_star=1;                                   %Correction factor for Reynolds number
elseif a>=1-0.05 && a<=1+0.05 && b<=0.24 && b>=0.17 && c<=0.143 && c>=0.071  %If this is true the pillow plate is of type E
    n6=2.52*c+0.24;                             %Part of power law for fanning factor
    n7=-0.3;                                    %Part of power law for fanning factor
    psi_A=0.75*b+0.46;                          %Dimensionless ratio for area of recirculation zone
    psi_Q=0.75*b+(1.54*c-0.014);                %Dimensionless ratio for heat transfer of recirculation zone
    dh_z1=(-18.31*b+(35.42*c+4.8))*10^-3;       %Hydraulic diameter of core flow
    s_star=1;                                   %Correction factor for Reynolds number
elseif a>=1.71-0.05 && a<=1.71+0.05 && b<=0.24 && b>=0.17 && c<=0.143 && c>=0.071 %If this is true the pillow plate is of type T
    n6=4.62*c+0.6;                              %Part of power law for fanning factor
    n7=-0.34;                                   %Part of power law for fanning factor
    psi_A=0.81*b+0.263;                         %Dimensionless ratio for area of recirculation zone
    psi_Q=0.46*b+(1.17*c-0.042);                %Dimensionless ratio for heat transfer of recirculation zone
    dh_z1=(-8.1*b+(60*c+2.1))*10^-3;            %Hydraulic diameter of core flow
    s_star=1.0761;                              %Correction factor for Reynolds number
else %Check if it is out of range, should be avoided
    n6=4.62*c+0.6;                              %Part of power law for fanning factor
    n7=-0.34;                                   %Part of power law for fanning factor
    psi_A=0.81*b+0.263;                         %Dimensionless ratio for area of recirculation zone
    psi_Q=0.46*b+(1.17*c-0.042);                %Dimensionless ratio for heat transfer of recirculation zone
```

```
    dh_z1=(-8.1*b+(60*c+2.1))*10^-3;        %Hydraulic diameter of core flow
    s_star=1.0761;
end


Re_H=G*d_hi/mu_V*(x+rho_ref(2)/rho_ref(1)*(1-x)); %Homogeneous Reynolds number
Y=1-0.1*((rho_ref(2)/rho_ref(1)-1)*(1-x))^0.4;     %Correction factor for two phase flow
Re_z1=Re_H*(s_star/(1-psi_A));               %Reynolds number for core flow
zeta_f=n6*Re_H^n7;                           %Fanning factor power law for core flow
Nu_ref=((zeta_f/8)*Re_z1*Pr_V)/(1.07+12.7*sqrt(zeta_f/8)*(Pr_V^(2/3)-1));   %Nusselt number for core flow
h_z1=Nu_ref*k_v/dh_z1;                       %Convective heat transfer coefficient for core flow
h_mist=h_z1*((1-psi_A)/(1-psi_Q))*Y^(-1.83); %Heat transfer for mist flow

h_dryout=h_mist;    %Define heat transfer for dry out to be the same as mist flow
end
```

Pressure Loss Function

```
function [Delta_P_grav_2ph,Delta_P_acc_2ph,Delta_P_Friedel]=PressureLossPP(Width,Height,a,b,c,Re_tot,u_m,d_hi,T_ref_in,Refrigerant_fluid,m_dot_ref,x_dot_ref,Nr_plates,A_csitot,Nr_cell_vertical)

%Calculates the two phase pressure loss based Friedels model as decribed in
%https://www.osti.gov/servlets/purl/850141 while accounting for the one
%phase pressure loss within the pillow plate from:
%https://www.sciencedirect.com/science/article/pii/S1290072917302363

%Refrigerant gas
```

```
h_ref(1)=py.CoolProp.CoolProp.PropsSI('H','T',T_ref_in,'Q',1,Refrigerant_fluid);            %Enthalpy of refrigerant at inlet, currently unused
Pr_ref(1)=py.CoolProp.CoolProp.PropsSI('PRANDTL','T',T_ref_in,'Q',1,Refrigerant_fluid);      %Prandtl number of refrigerant
rho_ref(1)=py.CoolProp.CoolProp.PropsSI('D','T',T_ref_in,'Q',1,Refrigerant_fluid);           %Density of refrigerant
mu_ref(1)=py.CoolProp.CoolProp.PropsSI('viscosity','T',T_ref_in,'Q',1,Refrigerant_fluid);    %Dynamic viscosity of refrigerant
cp_ref(1)=py.CoolProp.CoolProp.PropsSI('C','T',T_ref_in,'Q',1,Refrigerant_fluid);            %Specific heat capacity of refrigerant
k_ref(1)=py.CoolProp.CoolProp.PropsSI('conductivity','T',T_ref_in,'Q',1,Refrigerant_fluid); %Thermal conductivity of refrigerant
T_ref_ss=py.CoolProp.CoolProp.PropsSI('T','T',T_ref_in,'Q',1,Refrigerant_fluid); %Thermal conductivity of refrigerant

%Refrigerant liquid
h_ref(2)=py.CoolProp.CoolProp.PropsSI('H','T',T_ref_in,'Q',0,Refrigerant_fluid);            %Enthalpy of refrigerant at inlet, currently unused
Pr_ref(2)=py.CoolProp.CoolProp.PropsSI('PRANDTL','T',T_ref_in,'Q',0,Refrigerant_fluid);      %Prandtl number of refrigerant
rho_ref(2)=py.CoolProp.CoolProp.PropsSI('D','T',T_ref_in,'Q',0,Refrigerant_fluid);           %Density of refrigerant
mu_ref(2)=py.CoolProp.CoolProp.PropsSI('viscosity','T',T_ref_in,'Q',0,Refrigerant_fluid);    %Dynamic viscosity of refrigerant
cp_ref(2)=py.CoolProp.CoolProp.PropsSI('C','T',T_ref_in,'Q',0,Refrigerant_fluid);            %Specific heat capacity of refrigerant
k_ref(2)=py.CoolProp.CoolProp.PropsSI('conductivity','T',T_ref_in,'Q',0,Refrigerant_fluid); %Thermal conductivity of refrigerant
sigma=py.CoolProp.CoolProp.PropsSI('surface_tension','T',T_ref_in,'Q',0,Refrigerant_fluid); %Thermal conductivity of refrigerant
```

```
g=9.81;
%% Pressure loss (This not really correctly setup yet)
L_pass=(Height+Width)/Nr_cell_vertical;                    %Lenght the flow has to pass through



if a>=0.58-0.05 && a<=0.58+0.05 && b<=0.14 && b>=0.1 && c<=0.083 && c>=0.042 %If this is true the pillow plate is of type L
    n1=8.76*b+(17*c+0.73);                                  %Coefficient for the pillow plate type, will add the other types later
    n2=-0.38;                                               %Coefficient for the pillow plate type, will add the other types later
elseif a>=1-0.05 && a<=1+0.05 && b<=0.24 && b>=0.17 && c<=0.143 && c>=0.071  %If this is true the pillow plate is of type E
    n1=-15.3*b+(1.4*c+5.4);                                 %Coefficient for the pillow plate type, will add the other types later
    n2=1.725*b+(1.11*c-0-66);                               %Coefficient for the pillow plate type, will add the other types later
elseif a>=1.71-0.05 && a<=1.71+0.05 && b<=0.24 && b>=0.17 && c<=0.143 && c>=0.071 %If this is true the pillow plate is of type T
    n1=1.35*b+(2.8*c+0.92);                                 %Coefficient for the pillow plate type, will add the other types later
    n2=0.3*b+(0.53*c-0.29);                                 %Coefficient for the pillow plate type, will add the other types later
```

```
else %Will be out of range, avoid this
    n1=1.35*b+(2.8*c+0.92);                                                    %Coefficient for the pillow plate type, will add the other types later
    n2=0.3*b+(0.53*c-0.29);                                                    %Coefficient for the pillow plate type, will add the other types later
    disp('Pillow plate parameters are out of range')
end


zeta_p(1)=n1*Re_tot(1)^n2;                                             %Darcy friction factor for only gas flow
Delta_P(1)=zeta_p(1)*L_pass*rho_ref(1)*u_m(1)^2/(2*d_hi);                 %Pressure loss for only gas flow
zeta_p(2)=n1*Re_tot(2)^n2;                                             %Darcy friction factor for only liquid flow
Delta_P(2)=zeta_p(2)*L_pass*rho_ref(2)*u_m(2)^2/(2*d_hi);                 %Pressure loss for only liquid flow

x=x_dot_ref;                                                          %Set the quality to what was obtained from the heat transfer model
G=m_dot_ref/(Nr_plates*A_csitot);                                     %Mass flux within the pillow plate
E=(1-x)^2+x^2*(rho_ref(2)*zeta_p(2))/(rho_ref(1)*zeta_p(1));           %Coefficient for Friedel flow multiplier
F=x^0.78*(1-x)^0.224;                                                  %Coefficient for Friedel flow multiplier
H=(rho_ref(2)/rho_ref(1))^0.91*(mu_ref(1)/mu_ref(2))^0.19*(1-mu_ref(1)/mu_ref(2))^0.7;  %Coefficient for Friedel flow multiplier
rho_h=(x/rho_ref(1)+(1-x)/rho_ref(2))^(-1);                             %Homogeneous density
```

```
Fr_h=G^2/(g*d_hi*rho_h^2);                                    %Coefficient for Friedel flow multiplier
We_l=G^2*d_hi/(sigma*rho_h);                                  %Liquid only Weber number
phi_lo_Friedel=E+(3.24*F*H)/(Fr_h^0.045*We_l^0.035);          %Flow multiplier for pressure loss
Delta_P_Friedel=phi_lo_Friedel*Delta_P(2);                    %Two phase pressure loss


theta=pi/2;                                                   %Angle of the plate, will be either 0 for horizontal or pi/2 for vertical
S=(1-x*(1-rho_ref(2)/rho_ref(1)))^(1/2);                      %Slip factor
alpha=1/(1+(S*(1-x)/x)*rho_ref(1)/rho_ref(2));                %Void fraction
if alpha==1                                                   %To avoid issues with expression at full gas flow
    alpha=0.9999999;
end

Delta_P_grav_2ph=(alpha*rho_ref(1)+(1-alpha)*rho_ref(2))*g*sin(theta)*Height/Nr_cell_vertical;   %Gravitational pressure loss, still needs to be examined
Delta_P_acc_2ph=G^2*((x^2/(alpha*rho_ref(1))+(1-x)^2/((1-alpha)*rho_ref(2)))-(1/(rho_ref(2))));  %Acceleration pressure loss, still needs to be examined

end
```