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Optimized and Cost Efficient Industrial Heat Pumps based on Low Grade Waste Heat

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MASTER THESIS

for

Student Robert Lund

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Optimized and Cost Efficient Industrial Heat Pumps based on Low Grade Waste Heat

Optimale og kosteffektive industrivarmepumper basert på lavtemperatur overskuddsvarme

Background and objective

The project objectives focus on development of industrial waste heat recovery solutions that aim at:

- recovering large quantities of waste heat, upgrading it to higher temperatures and reusing as thermal energy in internal industrial processes,
- high system durability expressed as at least 20 years life span
- reducing primary energy consumption,
- decarburization and resource efficiency of the energy supply

The first two statements define whether the proposed technological solutions and practices will be widely accepted by the end users or not, and are essential conditions, if we aim at their wide market penetration and large scale impact. The last two statements have a straightforward direct impact on optimum exploitation of the energy resources, as well as on environmental protection.

Recently, the heat pumps technology appears considerable application in household sector. Demonstrating the superior performance of the technology is essential in order to facilitate its large penetration in the industrial sector and demonstrate that can apply in pilot plants at industrial sites by improving this technology in terms of:

- heat supply at temperature range (+90 to +140°C or more)
- energy efficiency, e.g. heat recovery from 60°C waste heat (COP/SCOP 5.0 – 6.1)
- cost effectiveness

The following tasks are to be consider:

1. Literature study of heat pump systems for heat recover in industrial applications
2. Evaluate different working fluids for high temperature applications, and its influence on the component design (compressor size, heat transfer in heat exchangers etc.)
3. Developing of a simulation model, using EES, for system optimization related to pipe and component design
4. Optimize the components regarding the component cost, inclusive total system costs and the energy efficiency of the system
5. Draft a scientific paper of the main results from the project

6. Make suggestion for further work

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Within 14 days of receiving the written text on the master thesis, the candidate shall submit a research plan for his project to the department.

When the thesis is evaluated, emphasis is put on processing of the results, and that they are presented in tabular and/or graphic form in a clear manner, and that they are analyzed carefully.

The thesis should be formulated as a research report with summary both in English and Norwegian, conclusion, literature references, table of contents etc. During the preparation of the text, the candidate should make an effort to produce a well-structured and easily readable report. In order to ease the evaluation of the thesis, it is important that the cross-references are correct. In the making of the report, strong emphasis should be placed on both a thorough discussion of the results and an orderly presentation.

The candidate is requested to initiate and keep close contact with his/her academic supervisor(s) throughout the working period. The candidate must follow the rules and regulations of NTNU as well as passive directions given by the Department of Energy and Process Engineering.

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Pursuant to "Regulations concerning the supplementary provisions to the technology study program/Master of Science" at NTNU §20, the Department reserves the permission to utilize all the results and data for teaching and research purposes as well as in future publications.

The final report is to be submitted digitally in DAIM. An executive summary of the thesis including title, student's name, supervisor's name, year, department name, and NTNU's logo and name, shall be submitted to the department as a separate pdf file. Based on an agreement with the supervisor, the final report and other material and documents may be given to the supervisor in digital format.

☐ Work to be done in lab (Water power lab, Fluids engineering lab, Thermal engineering lab)
☐ Field work

Department of Energy and Process Engineering, September 15th 2015

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Preface

This master thesis was carried out at the Department of Energy and Process Engineering of the Norwegian University of Science and Technology (NTNU). The master thesis constitutes 30 study points / ECTS credits and was performed during the first semester of the academic year 2015-2016.

I would like to give my thanks to my supervisor Trygve Magne Eikevik for his guidance and advice throughout this work. I would further like to thank Ignat Tolstorebrov for giving good feedback and answering my questions.

I will be forever grateful to my family and friends for making my time at NTNU truly memorable!

Nature's first green is gold,
Her hardest hue to hold.
Her early leaf's a flower;
But only so an hour.
Then leaf subsides to leaf.
So dawn goes down to day.
Nothing gold can stay.

Trondheim, Norway
February 2016

Robert Hoel Lund

Abstract

The industrial sector is a large consumer of energy. Globally, this sector accounts for over half of the energy used. As anthropogenic emissions increase, and the consequences of these emissions become more evident, the incentive for energy reduction is clear. Heat pumps are readily available for low temperature applications in the residential sector, but has yet to make a breakthrough in industry. This is in stark contrast to the energy and cost savings such installations can induce.

To make heat pumps more attractive for industrial applications, it is important to demonstrate their potential. This master thesis investigates one such case, namely the implementation of a high temperature heat pump in a distillation plant. The heat pump will fulfill two purposes. First, it will cool down a waste stream to allow operation during warm summer months. Due to environmental restrictions, the availability of cooling water from a nearby river is restricted. Secondly, it will replace bought energy by using the condenser heat to preheat a distillation feed.

A vapour compression cycle was simulated by a model made in EES. Suitable natural working fluids for high temperature applications were investigated with respect to cycle performance and their influence on component design. The case investigated gave the boundary conditions for the model, e.g. temperature levels and heat duty. In addition, an economic model was implemented to optimize the heat pump with respect to the total system costs, including component costs and the energy efficiency of the system.

The working fluid with the best performance was n-butane, with a coefficient of performance close to 6. The requirement for this was a large evaporator which gave a small pinch temperature between the working fluid and water. The added investment costs of a larger evaporator could be justified by the reduced total costs due to lower operational costs.

Sammendrag

Globalt står industrisektoren for over halvparten av klodens energiforbruk. Konsekvensene av menneskeskapte klimautslipp blir tydeligere. Dette gir økt fokus på energibruk. Lavtemperaturs varmepumper er utbred i boligsektoren. Derimot er ikke varmepumper like utbredt i industrien. Dette er i sterk kontrast til det store potensialet industrielle varmempumper har, med tanke på besparelser i form av energi og penger.

For å gjøre industrielle varmepumper mer ettertraktede, må potensialet demonstreres. Denne masteroppgaven undersøker hvordan en høy-temperaturs varmepumpe kan implementeres i en fabrikk. Grunnet begrenset tilgang på kjølevann, må fabrikken senke produksjonen om sommeren. Varmepumpen vil derfor oppfylle to formål. Det første formålet er å kjøle ned en avfallsstrøm før behandling, slik at miljøreguleringer blir overholdt. For det andre vil spillvarmen brukes til å forvarme en produktstrøm. Ved å forvarme med spillvarme vil fabrikken spare både energi og penger.

En dampkompresjonsvarmepumpe ble simulert ved å lage en modell i EES. Egnede naturlige arbeidsmedier for høy-temperaturs varmepumper ble undersøkt for å se hvordan de påvirket komponentdesign og systemytelse. Grensebetingelsene til modellen ble satt ved å bruke de reelle dataene fra fabrikken. I tillegg ble en økonomisk modell implementert for å optimalisere varmepumpen med hensyn til systemets totale kostnader. Herunder både komponentkostnader og driftskostnader.

Arbeidsmediet med best ytelse var n-butan med en effektfaktor på nesten 6. For å få til dette måtte fordamperen være stor, slik at pinchtemperaturen mellom arbeidsmedium og vann i varmeveksleren ble lav. Denne ekstra investeringskostnaden kunne forsvares ved at driftskostnadene ble mindre, og dermed også de årlige totalkostnadene.

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Nomenclature

Symbol	**Units**	**Description**
Bo	-	boling number
C	kJ/K	heat capacity
D	m	diameter
d_h	m	hydraulic diameter
f_D	-	Darcy's friction factor
g	m/s^2	gravitational constant
H	kJ	enthalpy
h	W/m^2K	heat transfer coefficient
i	kJ/kg	specific enthalpy
k	W/mK	thermal conductivity
L	m	length
m	kg	mass
$\dot{m}$	kg/s	mass flow
n	year	the depreciation time
N_{ch}	-	number of channels
p_{el}	NOK/kWh	electricity price
Pr	-	Prandtl number
Q	W	heat duty
q”	W/m^2	heat flux
r	-	interest rate
Re	-	Reynolds number
T	K	temperature
th	m	thickness
U	W/m^2K	overall heat transfer coefficient
u	m/s	velocity
w	m	width
W	W	work
x	-	vapour quality
X_{tt}	-	Lockhart-Martinelli parameter

Subscripts

acc	accelerational
C	cold
comp	compressor
cond	condenser
crit	critical
cross	cross sectional area
el	electric
eq	equivalent
evap	evaporator
fric	frictional
g	vapour
grav	gravitational
H	hot
HP	high pressure
int	intermediate
inv	investment
is	isentropic
l	liquid
LP	low pressure
m	mean
man	manifold
op	operational
plate	plate
ref	refrigerant
tot	total
tp	two-phase
w	water
wall	plate wall

Greek symbols

β	chevron angle
Δi_{lat}	latent heat of evaporation
ϵ	relative roughness
η	efficieny
κ	isentropic expontent
μ	viscosity
ρ	density
σ	surface tension
τ	hours of operation

Abbreviations

GHG	Greenhouse gases
COP	Coefficient of performance
IHP	Industrial heat pump
ODP	Ozone depletion potential
GWP	Global warming potential
CCC	Closed cycle compression
MVR	Mechanical vapour recompression
TVR	Thermal vapour recompression
HCFC	Hydrochlorofluorocarbon
CFC	Chlorofluorocarbon
HFO	Hydrofluoroolefin
HC	Hydrocarbon
EES	Engineering Equation Solver

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Chapter 1

Introduction

1.1 Motivation

According to the Intergovernmental Panel on Climate Change, it is *extremely likely* that more than half of the observed increase in global average surface temperature from 1951 to 2010 was caused by the anthropogenic increase in GHG concentrations and other anthropogenic forcings together. (IPCC, 2014). Due to these changes, it is very likely that heat waves will occur more often and last longer, extreme precipitation events will become more frequent and intense, and global mean sea level will rise. To limit these changes, it is imperative to reduce GHG emissions by reducing the consumption of fossil energy.

Energy use is categorized into four main sectors by U.S Energy Information Administration. Namely the commercial sector, the industrial sector, the residential sector and the transport sector. (EIA, 2015) As a share of the total global energy use, the industrial sector uses 51 % . In Europe, final energy consumption by the industry is 27.5 % (Jakobs et al., 2010). Uncertainty about energy prices and mounting environmental concerns, there is a strong motivation for increased energy efficiency.

Increased energy efficiency is achievable by the implementation of heat pumps. Heat pumps are widely used in the commercial sector for space and water heating purposes with great success. However, this is not yet the case in the industrial sector. Waste heat by industrial processes is often dumped to the environment. With heat pumps, this heat can be upgraded to a higher, more usable temperature level and used for various processes. Primary energy consumption can therefore be greatly reduced. Even with the potential for large savings, heat pumps have yet to be widely implemented in the industrial sector.

1.2 Scope of the master thesis

This thesis is a continuation of previous project work. The aim is to investigate the potential of heat recovery by the use of heat pumps in industrial processes. The potential of heat pumps are based on both technological and economical factors.

The technology must be able to lift the temperature of the waste heat high enough for it to be useful. Working fluids suitable for high temperature heat pump must be found and evaluated. Prices of the components constituting a heat pump are collected from companies. An economic model which take into account both investment and operational costs is made.

A simulation model is developed to investigate n-butane and isobutane's performance at realistic conditions. The simulation model also includes an economic part which finds the cost of the heat pump modeled. The project work is limited to one case, namely the implementation of an IHP at the Perstorp Factory in Arnsberg, Germany. This plant will provide the boundary conditions for the simulation model.

1.3 Structure of the master thesis

Chapter 2 provides an overview of different types of heat pumps and their application in industry.

Chapter 3 establishes which working fluids are applicable for high temperature applications, and investigated how the particular working fluids influences the different components of the heat pump.

Chapter 4 explains the chosen case and how the simulation model works.

Chapter 5 describes the economic model applied.

Chapter 6 sums up the results from the simulation.

Chapter 7 discusses the simulation model and the derived results. The conclusion and proposal for further work follows.

Chapter 2

Technical review

2.1 Heat pumps

A heat pump is a device that is able to lift heat from a low temperature reservoir to a high temperature reservoir by high grade energy input. This is in contrast to the natural flow of heat, which is from high temperature to low temperature. By lifting the temperature, a heat pump makes it possible to recover otherwise unused waste heat. By upgrading the waste heat to a useful temperature, it is possible to replace bought energy, and thus save energy costs. The principle of a heat pump is shown in figure 2.1:

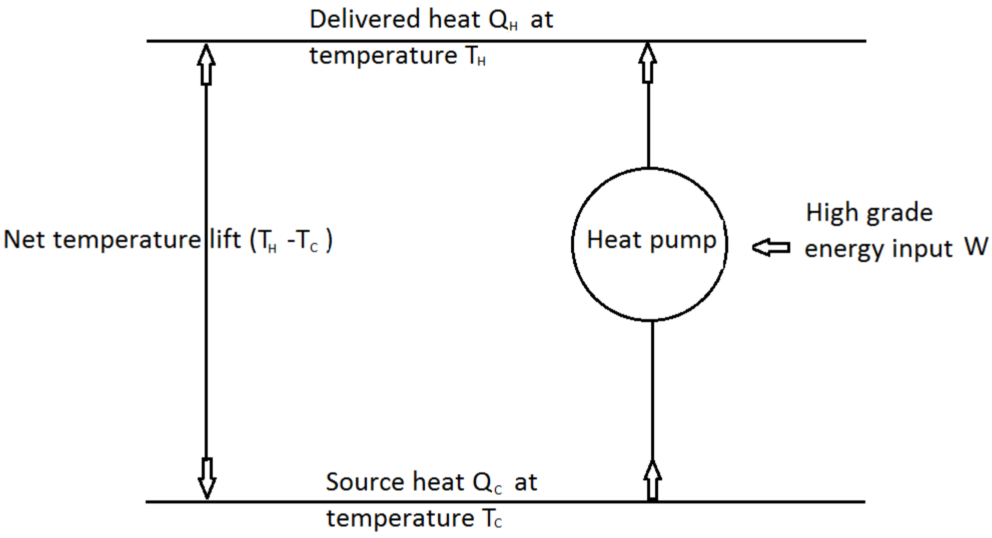


Figure 2.1: Thermodynamic principle of a heat pump.

The coefficient of performance is an important number when comparing the energy efficiency of different systems. It can be defined in various ways.

The most common definition of the COP for a heat pump is the heat output divided by the work input:

$$COP = \frac{Q_{out}}{W_{in}} \tag{2.1}$$

The Carnot efficiency for heat pumps gives the maximum theoretical efficiency for heat pumping between two thermal reservoirs:

$$COP_{Carnot} = \frac{T_H}{T_H - T_C} \tag{2.2}$$

In practice, it is not possible to reach this efficiency due to losses in the process.

The energy saved by the heat pump is given by the relation below:

$$\text{Energy saved} = \left(1 - \frac{1}{COP}\right) * 100\% \tag{2.3}$$

Heat pumps commonly used in industrial applications:

- **Closed Cycle Compression**
- **Open Cycle Mechanical Vapour Recompression**
- **Open Cycle Thermal Vapour Recompression**
- **Closed Cycle Absorption**
- **Closed Cycle Hybrid Compression/Absorption**

2.1.1 Closed Cycle Compression Heat Pump

The closed cycle compression heat pump, CCC, circulates a working fluid between an evaporator and a condenser. The basic cycle consists of two heat exchangers, a throttling valve and a compressor. This arrangement is shown in figure 2.2. The working fluid evaporates in the the evaporator, increasing its enthalpy and vapour fraction. After the evaporator, the gaseous working fluid is compressed to a higher pressure by the compressor, thus increasing its temperature. Afterwards the working fluid condenses in the condenser, releasing its heat. Next, the condensed working fluid is depressurized by the expansion valve. At last, the working fluid once again enters the evaporator, thus closing the loop. More advanced designs may have several throttling valves, compressors and internal heat exchangers. The compressor is usually run either by an electric motor or a combustion engine. A CCC is very flexible, and can provide high temperature lifts.

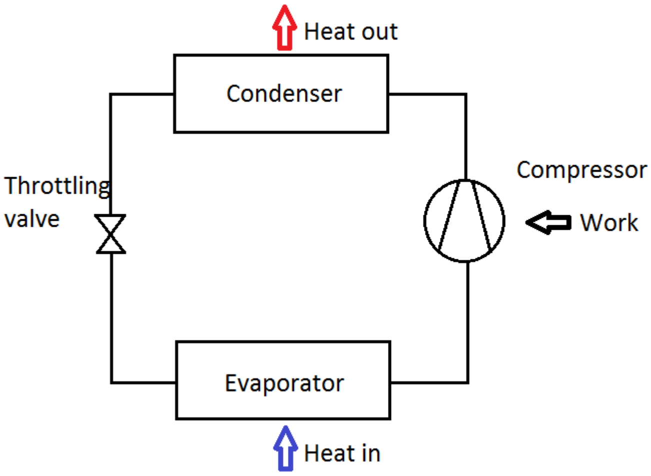


Figure 2.2: Vapour compression heat pump.

2.1.2 Open Cycle Mechanical Vapour Recompression Heat Pumps

Mechanical vapour recompression, MVR, directly compresses a vapour to a higher pressure level, thus increasing its temperature. The working fluid is water vapor, and the heat pump is considered open cycle since the working fluid is the process stream (McMullan, 2003). Compression is done mechanically by a compressor. This implies that electricity or other types of mechanical work is consumed. The high pressure steam can be utilized directly for process purposes, or transfer its heat to a heat sink. Typical values for the pressure increase and temperature lift can be found from industry. Borregaard in Sarpsborg, Norway, concentrates sulphite lye. Their MVR heat pump increases the pressure of the water vapour from 1.2 bar to 1.8 bar, and the temperature from 105°C to 117°C. (Haukås, 2010)

The process shown in figure 2.3 is very similar to the configuration of the TVR heat pump, but the ejector and the high pressure motive steam are replaced by a compressor.

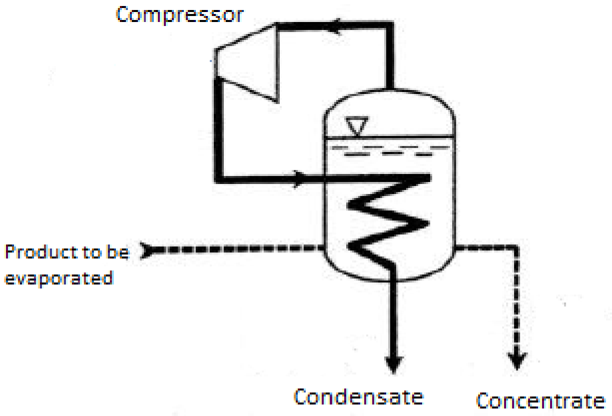


Figure 2.3: Mechanical vapour recompression. From (Haukås, 2010)

This heat pump is very energy efficient, with a COP in the order of 20. (Haukås, 2010)

2.1.3 Open Cycle Thermal Vapour Recompression Heat Pumps

Thermal Vapour Recompression, TVR, is achieved by using high pressure steam and an ejector. The motive steam draws vapour with low pressure into the ejector, where the mixed steam subsequently expands through a diffuser. This expansion converts the kinetic energy of the steam into pressure, thus raising the pressure and temperature of the vapour which have been drawn into the ejector. After the ejector, the vapour condenses in a heat exchanger. The heat of condensation is transferred to the liquid being evaporated, boiling off more vapour. The process is shown in figure 2.4:

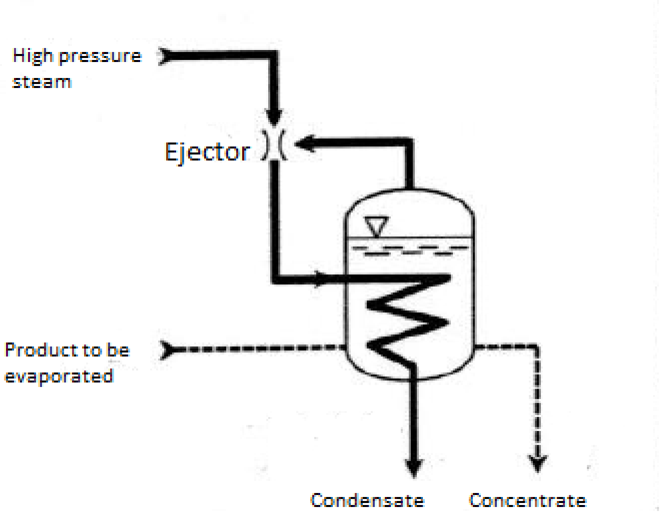


Figure 2.4: Thermal vapour recompression.From (Haukås, 2010)

A major advantage of the TVR heat pump is the fact that there are no moving parts, resulting in low maintenance and high robustness.

2.1.4 Closed Cycle Absorption Heat Pump

The working fluid in an absorption heat pump is a binary mixture of two components. One volatile and one non-volatile. The two most common mixtures are ammonia/water (NH_3/H_2O) and water/lithium bromide(H_2O/LiBr).

There are four heat exchangers in a basic absorption heat pump. Evaporator, absorber, generator/desorber and condenser. More advanced cycles have multiple pressure levels and internal heat exchangers. In the evaporator, the volatile component absorbs heat at a low level. It is then absorbed by the non-volatile component in the absorber. Due to the exothermic reaction of mixing, heat is released at a medium temperature level. The binary mixture is pumped to the generator where the volatile component is being boiled off at high pressure by the high-temperature heat source. This heat is usually provided by waste heat or by burning fossil fuel. The high temperature vapour from the generator condenses in the condenser, releasing heat at a medium temperature level. After condensation, the condensed volatile compound is throttled before once again absorbing heat in the evaporator. The non-volatile component passes directly from the generator back to the absorber through a throttling valve.
The process is shown in figure 2.5:

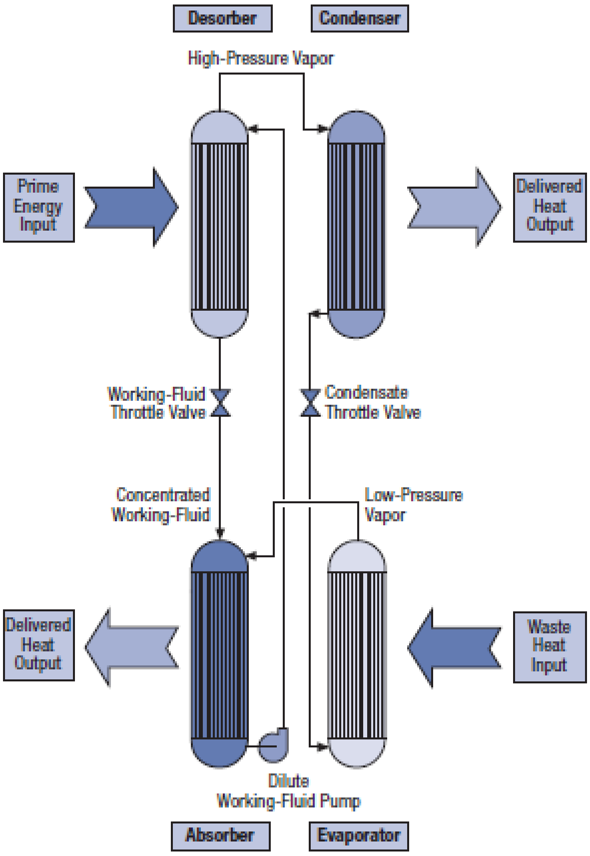


Figure 2.5: Absorption heat pump. From (McMullan, 2003)

Depending on how the pressure levels and fluid concentrations are configured, there are two types of heat pump configurations. Either type 1 (Heat amplifier) or type 2 (Temperature amplifier). Parameters of interest are the COP, maximum temperature lift and maximum temperature. (Soroka, 2010). The COP is defined as the ratio between useful heat output and heat input, while the maximum temperature is the maximum temperature level at which heat can be delivered. This temperature is limited by the risk of crystallization, and is very important for the applicability for high-temperature heat recovery.

The heat amplifier delivers the sum of the high temperature and low temperature heat duties at an intermediate temperature. Thus, a large amount of heat at an intermediate temperature can be generated. This heat pump can simultaneously provide cooling and heating. Cooling is done in the evaporator and heating in the absorber and condenser. Typical COP is 1.6-1.7.

The temperature amplifier allows a medium temperature stream to split into a high temperature stream and a low temperature stream. The low temperature stream is typically rejected to the ambient. (McMullan, 2003) The COP of this heat pump is determined by the ratio of heat delivered at high temperature and heat recovered at medium temperature. Typical COP is between 0.4-0.5. (Soroka, 2010)

A great advantage with the absorption heat pump is the low maintenance costs, due to few moving parts. The main drawback is the high investment cost, which is about twice the cost of a similar sized vapour compression system. (Lewis et al., 2009)

2.1.5 Hybrid Absorption Heat Pump

A compression/absorption hybrid heat pump is based on the combination of a vapor compression- and absorption heat pump. It utilizes a binary mixture of two components, one volatile and the other non-volatile. The most common mixture is NH_3/H_2O.

The main components of a hybrid heat pump is a compressor, a solution pump, a desorber and an absorber. In addition, there are several auxiliary components. Namely an internal heat exchanger, a desuperheater, a rectifier and a liquid/vapour separator.
The cycle works as follows: In the desorber, vapour is boiled off from the strong solution. The vapour from the desorber is compressed by the compressor to a higher pressure. The now weak solution from the desorber is pumped by the solution pump to a higher pressure level. If an internal heat exchanger is present, the weak solution from the desorber exchanges heat with the strong solution from the absorber. High pressure vapour from the compressor and high pressure weak solution from the pump is mixed before entering the absorber. Here the vapour is absorbed into the solution. This exothermic reaction produces the desired heat. From the absorber, the now strong solution is heat exchanged before being depressurized in the throttling valve. The low pressure strong solution enters the desorber again, and the loop is closed. The cycle is shown in figure 2.6:

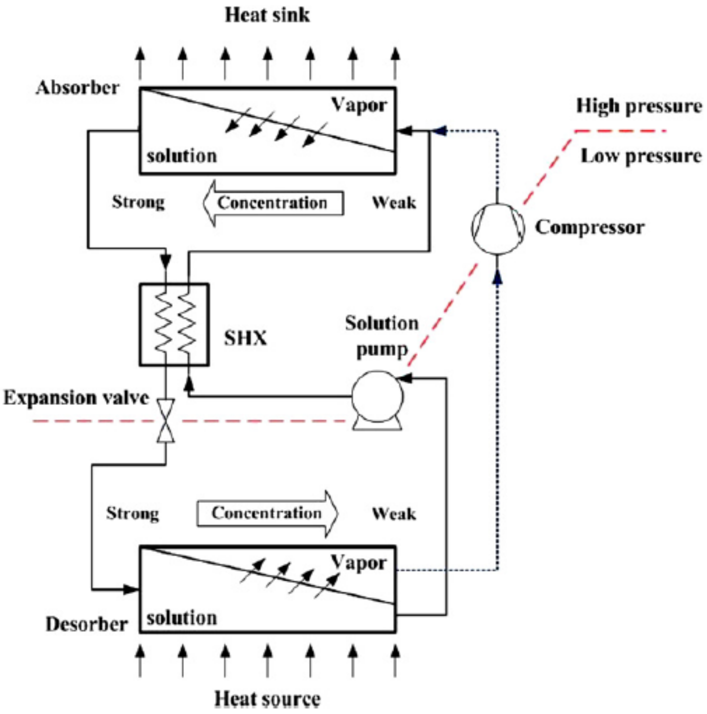


Figure 2.6: Hybrid absorption/compression heat pump. From (Kim et al., 2012)

There are several advantages compared to a standard vapour compression system. The most important advantage is the temperature glide in the absorber and desorber. Other benefits are greater capacity control, higher temperature lift and a more flexible operating range. (Kim et al., 2012)

2.2 Heat pumps in industrial applications

Heat pumps can replace the use of fossil fuels in many industrial applications.

Drying

Drying processes are widely used in industrial processes. Dehumidification and drying processes are usually carried out below 100°C, which makes industrial heat pumps suitable for the task. The process is carried out in a closed system by a vapour compression heat pump. Dry warm air from the condenser absorbs moisture from the drying products in the drying chamber. The absorption of water cools the air. Cool humid air from the drying chamber condenses on the heat pump evaporator, and the condensed water is removed from the system. Subsequently, the cool dry air is once again heated by the heat pump condenser before entering the chamber. In comparison with traditional drying processes, heat pump drying can lead to large energy savings (Wolf et al., 2012). Additional benefits are better product quality and minimization of odour due to the closed system.(Jakobs et al., 2010) Drying processes are prevalent in food, lumber, and paper and pulp industries.

Evaporation/Distillation

Evaporation/distillation processes are mainly used in food and chemical industries. In evaporation, the residue is the main product, while the distillate is the main product in distillation. The heat needed for evaporation is supplied by a MVR heat pump. Vapour is removed from the evaporator and compressed to a slightly higher pressure level, thus increasing the temperature. Afterwards, the vapour is condensed to heat the evaporator. An auxiliary heater is only required to start the evaporation process. The low temperature lift makes this a very efficient system with a high COP. An example of application is the manufacturing of drinking water by desalination.

Process water heating/cooling

Low to medium temperature hot water in the range from 40-90°C is needed for a wide range of industrial applications. Cooling is also required in many industries.The simultaneous requirement of both heating and cooling makes an industrial heat pump ideal. The heat rejected from the refrigeration system can be used as a heat source for the heat pump, leading to highly efficient systems.

Space heating

Industrial heat pumps can utilize waste heat which cannot be used directly for space heating. An example of such waste heat is heat rejected from refrigeration systems, which typically is 30-40°C. Upgrading this heat and replacing heat generated by other means can reduce costs. There are many applications for such heat, among other greenhouses. (Jakobs et al., 2010). Systems utilizing the heat from cooling processes are the most common retrofit heat pump systems in German industry (Wolf et al., 2012). In northern Europe, there are several heat pumps which provide large amounts of heat for district heating. (Jakobs et al., 2010)

Steam production

Industry uses wast amounts of steam for various purposes, including direct use in industrial processes and steam for heat distribution. The application of heat pumps for high temperature steam production can lead lead to large energy savings. High temperature heat pumps can produce steam at temperatures up to 150°C. (Jakobs et al., 2010)

Chapter 3

Evaluation of working fluids

3.1 Working fluids applicable for high temperature heat pumps

The applicability of working fluids for high temperature heat pumps are governed by various parameters. Most important are critical pressure p_{crit}, critical temperature T_{crit}, global warming potential GWP and ozone depletion potential ODP. A high T_{crit} gives the possibility to deliver heat at high temperatures with a sub-critical cycle. P_{crit} should be low, since the pressures needed to deliver heat at high temperatures will be moderate. Zero ODP and low GWP is important from an environmental point of view. In addition, the working fluid should be safe to use. The safety of a working fluid is commonly given by its Ashrae classification. It should ideally belong to safety group A1, meaning it is non toxic and non flammable. (Ashrae, 2010). A working fluid in this category will not need extra precautionary investments.

The Montreal Protocol of 1987 banned and restricted the use of chlorofluorocarbons (CFCs) and hydrochlorofluorocarbons (HCFCs). Since then, there have been a lack of suitable working fluids for high temperature applications (Wolf et al., 2012). In recent years, many new synthetic working fluids have been developed to replace banned CFC. Some of these working fluids display favourable characteristics for high temperature applications. Some of them also have a low GWP, making them very attractive for future high-temperature heat pumps. While new working fluids are being created,there are growing environmental concerns. This has lead to an increased focus on natural working fluids, which are less harmful to the environment than many synthetic working fluids. A natural working fluid can be defined as ”Natural working fluids occur in nature’s biological and chemical cycles without human intervention. These materials include ammonia, carbon dioxide, natural hydrocarbons, water and air."' (Committee, 2011) The EU Regulation No 517/2014 (Mota-Babiloni et al., 2015) dictates a phase out of high GWP-working fluids in the EU the next decades, further promoting the use of natural working fluids.

Working fluids suitable for high temperature heat pumps:

Hydrofluorocarbons, HFCs. Replacement for the gases being phased out by the Montreal Protocol. Though not ozone depleting, they still have large GWP values.

R245fa has a T_{crit} of 154°C and p_{crit} of 36,51 bar. The GWP is 1030. It is classified in safety group B1 due to its toxicity. It is suitable for moderately-high temperature conditions (Pan et al., 2011).

R365mfc has a T_{crit} of 186.9°C and p_{crit} of 32,6 bar. The GWP is 794. (Fröba et al., 2004)

Hydrofluoroolefins, HFOs. Also known as "fourth generation refrigerants." The working fluids in this group are relatively new, and have very low GWPs. A drawback is their high cost. For instance, R-1234yf are 15 times more expensive than a similar sized container (13,5 kg) of R-134a. Other concerns are toxicity and flammability. (Brown, 2013)

R1234ze(Z) has a T_{crit} of 150.1°C and p_{crit} of 35,33 bar. It is seen as very promising, with a potential capability similar to the banned working fluid CFC-114.(Kondou and Koyama, 2014).

R1234ze(E) has a T_{crit} of 109.4°C and p_{crit} of 36,32 bar. The critical temperature is too low for the highest temperature applications.

R717, ammonia, is a natural working fluid with zero ODP and GWP and excellent physical and thermophysical properties. R717 has a low molar mass of 17 $kg/kmol$, which leads to small pipeline dimensions. It also has a very high specific enthalpy of evaporation, which means a low mass flow and working fluid charge in the system. Many favourable properties in combination with a high critical temperature of 132.2°C, makes R717 a suitable working media for medium to high temperature applications. The main drawback of R717 is toxicity and flammability, which requires some extra safety measures. The Lower Explosion Limit (LEL) and auto ignition temperature of 651°C in 15 % by volume in dry air classifies R717 as only moderately flammable, and no extra explosion-proof measures are required. IDLH concentration (Immediately Dangerous to Life or Health) is 500 ppm, and lowest fatal concentration reported is 5000 ppm. It has an extremely pungent odour, which is noticeable for humans in concentrations as low as 5-50 ppm. This eases detection of leakages. (Stene, 1999).

Another drawback in addition to the safety aspect, is a high critical pressure of 112.8 bar. Current R717 heat pumps reach temperatures of up to 90°C. Due to the high working pressure needed to increase it further, compressor technology must be improved. (Kondou and Koyama, 2014)

R744,CO_2, is a natural working fluid with zero ODP and a GWP of 1. It is also non-toxic and non-flammable, making it a very safe working fluid. Due to the critical temperature of only 31°C, heat pumps using R744 have to use a transcritical cycle to deliver useful heat at high temperatures. The heat is delivered with a large temperature glide in a gas cooler. The close temperature fit of the water and CO_2 implies low exergy loss and a high efficiency. To achieve a high COP, the CO_2 has to be cooled down as much as possible in the gas cooler. A CO_2 heat pump would therefore be most efficient when heating for instance cold tap water to high temperatures.

Hydrocarbons, HCs, are natural occurring substances with low GWPs. For high temperature applications, n-butane(R600), and Iso-butane (R600a) are the most promising. Butane and isobutane have critical temperatures of 152°C and 134.7°C, respectively. The critical pressure is 37,96 bar and 36,4 bar, respectively. The main drawback with HCs are their flammability, which makes safety an issue.

Water, H_2O is a common working fluid with good properties. A high T_{crit} of 374.1°C and heat capacity makes it suitable for high temperature applications. It is primarily used in the MVR and TVR cycles, but also in absorption cycles in conjunction with LiBr or NH_3. Its applicability in closed cycle mechanical compression cycles is limited because of water vapour's low density.

3.2 Physical properties' influence on the component design

Several different physical properties influence the design of the various components in a heat pump.The most important properties are listed below:

- **Density** ρ
- **Viscosity** μ
- **Thermal conductivity** k
- **Surface tension** σ
- **Enthalpy of vaporization** Δi_{lat}

Density: The density of the working fluid greatly affects the component design. The specific volume is the inverse of the density. For a given mass flow, a working fluid with a high density gives a low volume flow. A low volume flow is beneficial, since both compressor size and duty is reduced. This can be seen by equation 3.5.

Viscosity: The viscosity is a measure the ”thickness” of a fluid. More precisely, it is the fluids' resistance to deformation by shear or tensile stress. The viscosity influences the Reynolds number. For a given mass flux, a lower viscosity will give a higher Reynolds number, and thus more turbulent conditions. This will in turn lead to a higher heat transfer rate, and a smaller heat transfer surface.

Thermal conductivity: The thermal conductivity of the working fluid is a measure of the fluids ability to conduct heat. A high thermal conductivity will decrease the thermal boundary layer in the interface between wall and fluid. This will lead to a higher heat transfer rate, and decrease the heat transfer surface needed.

Surface tension: The surface tension governs the heat transfer. A low surface tension reduces the superheat needed for nucleation and bubble growth. A low surface tension will therefore increase the heat transfer rate for a boiling liquid. Hence, a smaller evaporator is needed.

Enthalpy of vaporization: Also known as latent heat of vaporization. It is temperature dependent, and decreases with increasing pressure. For a given heat duty, a high enthalpy of vaporization reduces the mass flow of refrigerant needed. This can be seen in equation 3.1.

3.3 Heat exchangers

The basic heat exchanger design is not influenced by the working fluid passing through. However, the way the plates are hold together may depend on the working fluid and operating conditions. The common plate and frame heat exchanger uses gaskets which distributes the flow and seals the spaces between plates. For high temperature applications, the critical pressure of the working fluid will dictate the pressure level needed to release the heat. For high pressure and high temperature conditions, gaskets are not resilient enough.(Li et al., 2011) One solution to this problem is brazed heat exchangers. Here the plates are brazed together with a filler metal. Copper is the most common. For ammonia applications, nickel is used as the filler metal. This is due to hydrous ammonia corroding copper. Another benefit with a brazed heat exchanger is the reduced risk of leakage.

The plates are corrugated into patterns to increase the turbulence and heat transfer. There are various patterns available. In figure 3.1 a chevron type pattern is shown. β is the angle of corrugation, or chevron angle. When the angle is 0°,the working fluid would passes vertically down the plate. A higher angle will promote higher heat transfer through increased turbulence and longer channels.

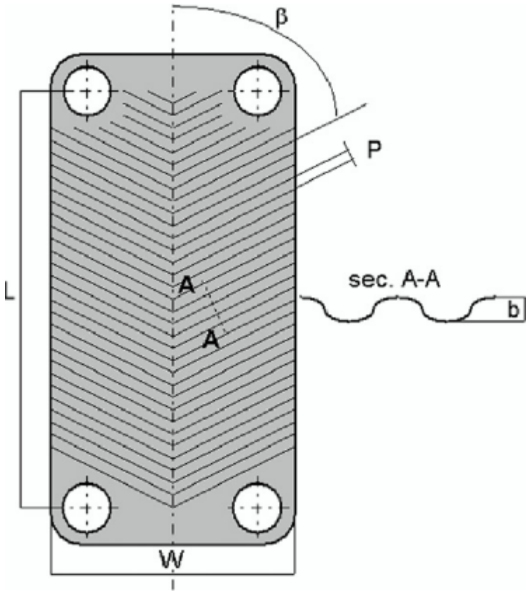


Figure 3.1: Plate with chevron pattern. From (Longo, 2010).

3.4 Compressors

The working fluids' properties influence the compressor size to a large extent. A fluid which requires comparatively higher volume flow to deliver the same amount of heat will result in a larger compressor. Other important parameters are the pressure levels and the pressure ratio. A high maximum pressure puts a lot of stress on the compressor. Another problems are internal leakages, where high pressure gas flows back to the low pressure sections of the compressor.

Governing equations

The mass flow of working fluid is given by:

$$\dot{m} = \frac{Q_{evap}}{\Delta i_{lat}} \tag{3.1}$$

where Q_{evap} is the heat duty and Δi_{lat} is the enthalpy of vaporization.

Further, the volume flow into the compressor is given by:

$$\dot{V} = \frac{\dot{m}}{\rho_g} \tag{3.2}$$

where $\dot{m}$ is the mass flow and ρ_g is the density of the suction gas at the compressor inlet. A certain amount of superheat from the evaporator is needed to make sure there is no liquid entering the compressor. Higher superheat will decrease the density of the gas, and thus increase the volume flow into the compressor.

The volumetric heating capacity is an useful expression when comparing different working fluids. The volumetric heating capacity is given by:

$$VHC = \Delta i_{lat} \rho_g \tag{3.3}$$

The heat duty divided by the the volumetric heating capacity equals the volume flow:

$$\dot{V} = \frac{Q_{evap}}{VHC} \tag{3.4}$$

The theoretical work needed to compress a given volume flow to the desired pressure is given by:

$$\dot{W} = \dot{V} p_1 \frac{\kappa}{\kappa - 1} \left[\left(\frac{p_2}{p_1} \right)^{\frac{\kappa-1}{\kappa}} - 1 \right] \tag{3.5}$$

where κ is the isentropic exponent, and the numbers 1 and 2 refers to the inlet and outlet pressures, respectively. As seen by equation 3.5, an increase in the pressure ratio will lead to increased work required to compress the gas.

Theoretical performance of various working fluids:

The theoretical properties of various working media is summarized in table 3.1

Working fluid	R600a	R600	R245fa	R365mfc	R1234ze(Z)	R717
$\dot{m}$ $[\frac{kg}{s}]$	2,233	1,671	3,183	2,792	2,679	0,4599
$\dot{V}$ $[\frac{m^3}{h}]$	252,6	268,1	296,2	577,8	259,1	55,36
VRC $[\frac{kJ}{m^3}]$	4276	4029	3646	1869	4168	19509
Δi_{lat} $[\frac{kJ}{kg}]$	134,4	179,5	94,25	107,4	112,2	652,4
P_{cond}[bar]	26,01	20,22	17,45	8,302	18,39	83,22
P_{ratio} [bar]	2,151	2,229	2,508	2,711	2,419	2,244

Table 3.1: Compressor results

The values are calculated for an evaporating temperature of 75°C and a condensing temperature 115°. Q_{evap} is 300 kW. The fluid entering the compressor is saturated vapour. The pressure ratio is the condenser pressure divided by the evaporator pressure.

As seen in table 3.1, ammonia scores well at all the properties except P_{cond} . This leads to a high pressure requirement for high temperature duties.

3.5 Pipe losses

The pressure loss in the pipes are given by the Darcy-Weisbach equation:

$$\Delta P = f_D \frac{L}{D} \frac{\rho u^2}{2} \tag{3.6}$$

The pressure drop for the applicable working fluids are summarized in table 3.2

Working fluid	R600a	R600	R245fa	R365mfc	R1234ze(Z)	R717
$\Delta P[Pa]$	7855	11082	6633	14809	6761	8532

Table 3.2: Pressure losses

These numbers are found by using the EES' built in function ”Pipe Flow” which calculates equation 3.6:

$$\text{call PipeFlow}(R\$; T_{ref}; P_{evap}; \dot{m}_{ref} P; D_{pipe}; L_{pipe}; RelRough :; ; \Delta P_{pipe}; ; ;) \tag{3.7}$$

This equation takes the working fluid, temperature, pressure, mass flow, pipe diameter, pipe length and relative roughness as inputs, and gives the pressure drop as output. The input variables for this particular case are: $T_{ref} = 75°$, $\dot{m} = 1{,}5\ kg/s$, $L_{pipe} = 10$ m , $D_{pipe} = 6$ cm and relative roughness $\epsilon = 5,0 * 10^5$.

Note that these results are for a given mass flow. The mass flow in the pipes would be different for the various working fluids given the same heat output.

Chapter 4

EES Model

4.1 Case

The case investigated is the integration of an IHP into a distillation process in a Perstorp factory in Arnsberg, Germany. Two dirty condensate streams generated by the process must be cooled down due to environmental regulations. This is because the amount of river water available for cooling is limited. As a consequence, the factory has to scale down production in the summer. To allow plant operation during the summer, the condensate streams must be cooled down from 90.3°C to 80°C to reduce the heat rejected to the river. This waste heat will be used to preheat a distillation feed from 96°C to 110°C. The IHP would therefore fulfill two purposes; allow plan operation in the summer and reduce energy consumption.

A simplified flow sheet of the Perstorp factory is shown in figure 4.1:

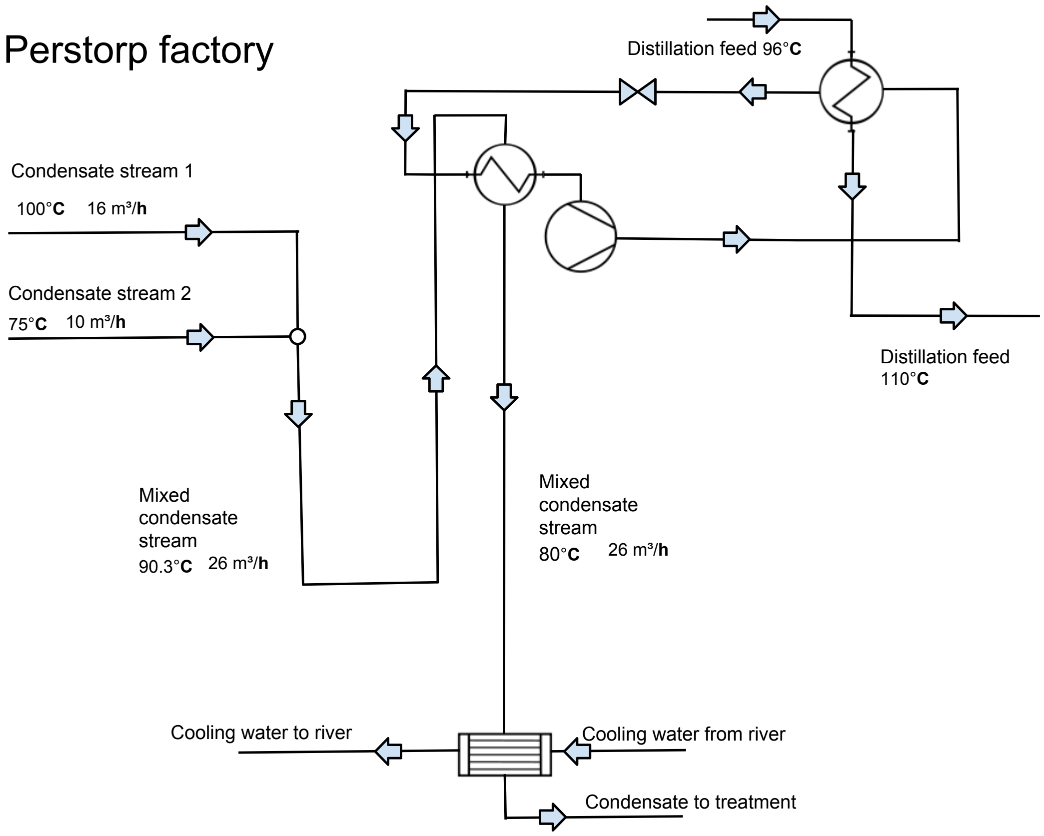


Figure 4.1: Flow sheet of the Perstorp factory.

The boundary conditions for the simulation model is taken from the investigated case. The mixed condensate stream consists of two streams. Stream 1 has a volume flow of 16 m^3/h and temperature of 100°C. Stream 2 has a volume flow of 10 m^3/h and a temperature of 75°C . These streams are mixed before entering the evaporator. The resulting stream has a volume flow of 26 m^3/h and a temperature of 90.3°C. This stream is cooled down to 80°C. Hence, the heat duty delivered to the evaporator is 300 kW. The heat pump will preheat a distillation feed from 96°C to 110°C. The mass flow and composition of this distillation feed is not known.

These boundary conditions are well suited for the simulation of a high temperature heat pump. Only natural working fluids are tested, namely R600 and R600a. R717 is not realistic to due the high pressures needed. R744 is not applicable because the lowest temperature of the working fluid is around 75°C. The enthalpy difference between the gas cooler inlet and outlet will be too small, and thus the system performance will be poor.

4.2 Heat pump model

The heat pump modeled is a two stage system with an intermediate pressure receiver, as shown in figure 4.2. The heat pump is modeled numerically in EES. Several functions were made and combined to find the most cost efficient heat pump.

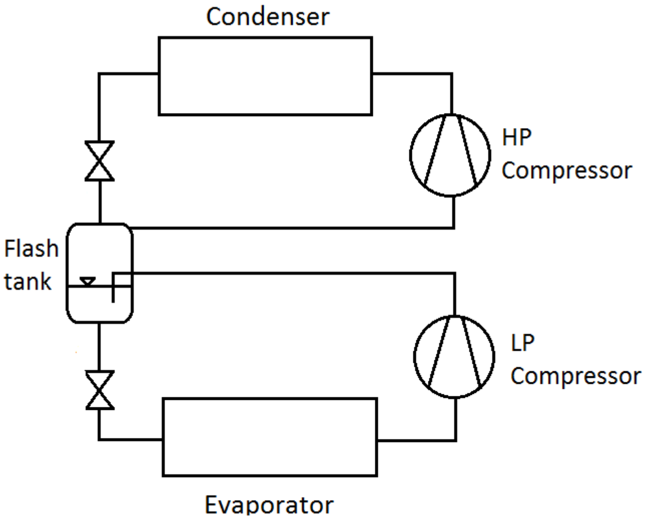


Figure 4.2: Two stage cycle with open flash tank.

The two stage process can be seen in a P-h diagram below:

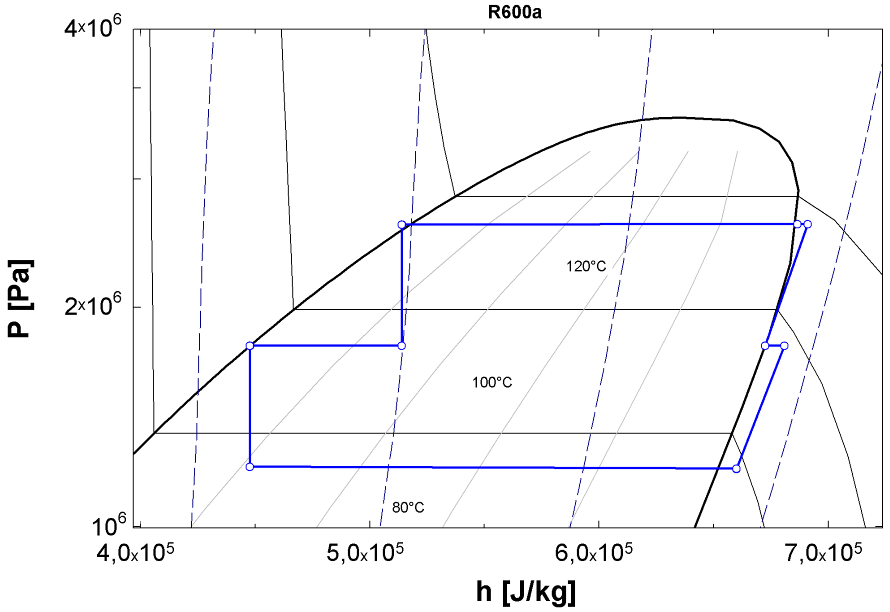


Figure 4.3: Two stage cycle with R600a.

The intermediate pressure is given by:

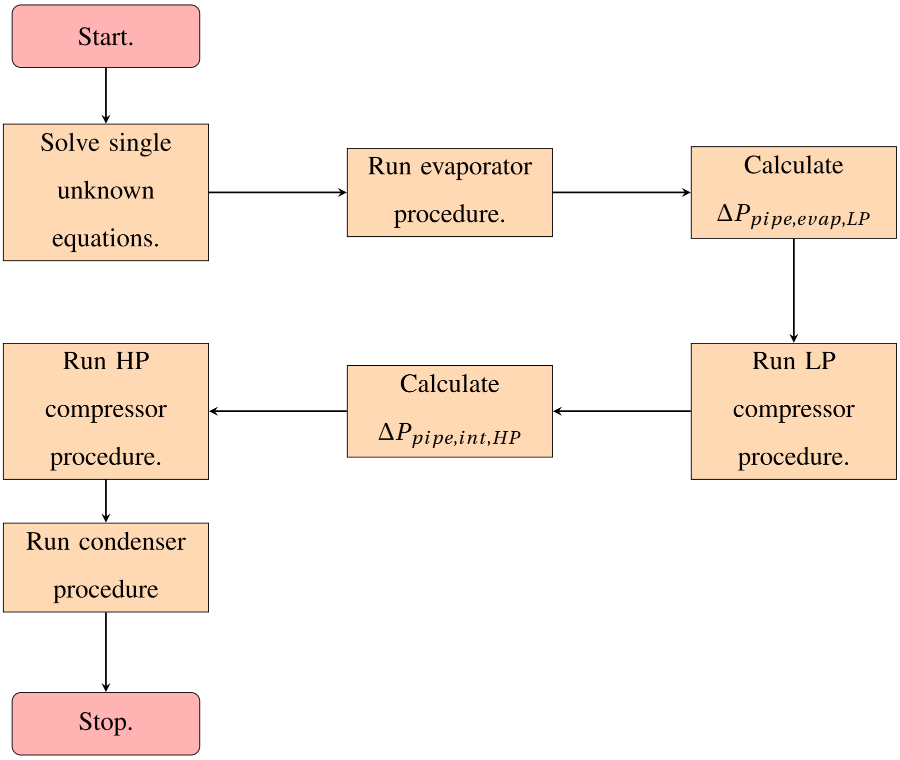
$$P_{int} = \sqrt{P_{evap}P_{cond}} \tag{4.1}$$

The pipe lengths connecting the components are as follows:

- $L_{evap,LP} = 5$ [m]
- $L_{LP,int} = 2$ [m]
- $L_{int,HP} = 2$ [m]
- $L_{HP,cond} = 5$ [m]
- $L_{cond,HT} = 8$ [m]
- $L_{int,LT} = 8$ [m]

The pipe lengths are not based on the real Perstorp factory, but instead a plausible theoretical case. The throttling valves are assumed to be close to the intermediate pressure tank and evaporator. It is assumed that the pressure drop from the throttling valves and to the flash tank and evaporator is negligible. The compressors modeled are reciprocating piston compressors. The evaporator and condenser are plate-plate heat exchangers.

A general flow chart of the EES model is shown on the next page:



When the calculations start, variables are initiated and single unknown equations are calculated. I.e functions converting temperatures from °C to K. Then the external evaporator procedure is executed. This procedure's outputs are used to calculate the pressure loss in the pipe leading to the low pressure compressor. Further, the compressor function is run to find the work needed to compress the gas, in addition to other variables of interest. Subsequently, the values for the pipe loss leading from the intermediate tank to the high pressure compressor are calculated. After that, the high pressure compressor function is run to find the variables of interest. Next, the external condenser procedure is run. The outputs are give back to the main program, and finally the last equations can be calculated. In the end error functions are minimized to find the correct values. The steps are explained further in later sections.

4.3 General modeling of the heat exchangers

Brazed plate and frame heat exchangers are used as a basis of the model. This is due to the high pressures needed. The plates are made from steel, and the chevron angle used in the calculation is 45°. Further, the hydraulic diameter of the channels resulting from the plates being pressed together are 5,6 mm. Plate thickness is 0,5 mm. Gap space is 2,8 mm. Flow channel width 2,2 mm. Plate width is 31,8 cm and plate length is 69 cm.

The heat exchangers are modeled with a numerical procedure to find the heat exchange area and the overall heat transfer coefficient needed to facilitate the necessary heat duties. The heat exchangers are divided into small control volumes. An energy balance is carried out which is described in section 4.4. From the change in water temperature in the control volume, the enthalpy change is found.This enthalpy change is the heat transferred to or from the working fluid. Subsequently all other parameters/variables of interest can be calculated. The next control volume's energy balance and other variables are calculated on the basis of the previous control volume. This process is executed until the desired superheat or sub-cooled temperature is found.

The temperature of the working media undergoing a phase change is given by the saturation pressure. Pressure drop in the heat exchangers leads to a temperature fall. This is further explained in section 4.5.3.

4.4 Deriving the state equations

The temperature change of the water in each control volume is given by the equations derived in this section. An energy balance is applied over a control volume which consists of one half of each channel. A channel is the space between two plates, as seen by figure 4.4.

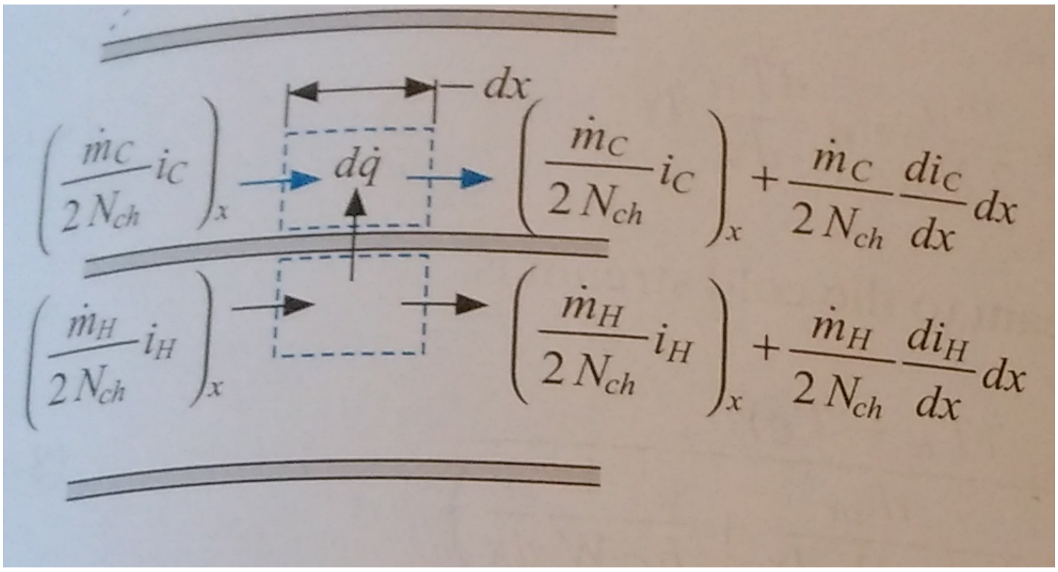


Figure 4.4: The energy balance. From (Nellis and Klein, 2009).

Below is the derivation of the the state equations. This derivation is found in (Nellis and Klein, 2009):

$$\left(\frac{\dot{m}_H}{2N_{ch}} i_H\right)_x = d\dot{q} + \left(\frac{\dot{m}_H}{2N_{ch}} i_H\right)_x + \frac{\dot{m}_H}{2N_{ch}} \frac{di_H}{dx} \tag{4.2}$$

where $\dot{m}_H$ is the hot side mass flow rate, $\dot{q}$ is the heat flow, N_{ch} is the number of channel pairs and i_H is the enthalpy of the hot stream.

$$0 = d\dot{q} + \left(\frac{\dot{m}_H}{2N_{ch}} i_H\right)_x + \frac{\dot{m}_H}{2N_{ch}} \frac{di_H}{dx} \tag{4.3}$$

Enthalpy derivative expanded:

$$0 = d\dot{q} + \frac{\dot{m}_H}{2N_{ch}} \left[\left(\frac{\partial i_H}{\partial T}\right)_p \frac{dT_H}{dx} + \left(\frac{\partial i_H}{\partial p}\right)_T \frac{dp_H}{dx}\right] dx \tag{4.4}$$

where T_H is the hot side temperature.

Pressure derived changes neglected.

$$0 = d\dot{q} + \frac{\dot{m}_H}{2N_{ch}} C_H \frac{dT_H}{dx} dx \tag{4.5}$$

where C_H is the hot fluid's heat capacity

Heat transfer rate from the hot to the cold stream:

$$d\dot{q} = \frac{T_H - T_C}{\frac{1}{h_H w dx} + \frac{th_{wall}}{k_{wall} w dx} + \frac{1}{h_C w dx}} \tag{4.6}$$

where T_C is the cold side temperature, h_H is the heat transfer coefficient of the hot stream, w is the width of the heat exchanger, th_{wall} is the thickness of the plate, k_{wall} is the conductivity of the plate metal, and h_C is the cold stream heat transfer coefficient.

Substituting equation 4.6 into equation 4.5:

$$0 = \frac{T_H - T_C}{\frac{1}{h_H w dx} + \frac{th_{wall}}{k_{wall} w dx} + \frac{1}{h_C w dx}} + \frac{\dot{m}_H}{2N_{ch}} C_h \frac{dT_H}{dx} dx \tag{4.7}$$

gives the temperature change per distance heat exchanger:

$$\frac{dT_H}{dx} = -\frac{2N_{ch}(T_H - T_C)}{\dot{m}_H C_H \left(\frac{1}{h_H w} + \frac{th_{wall}}{k_{wall} w} + \frac{1}{h_C w}\right)} \tag{4.8}$$

An similar derivation of the cold side fluid temperature change per distance yields:

$$\frac{dT_C}{dx} = \frac{2N_{ch}(T_H - T_C)}{\dot{m}_C C_C \left(\frac{1}{h_H w} + \frac{th_{wall}}{k_{wall} w} + \frac{1}{h_C w}\right)} \tag{4.9}$$

4.5 Heat exchangers

4.5.1 Evaporator

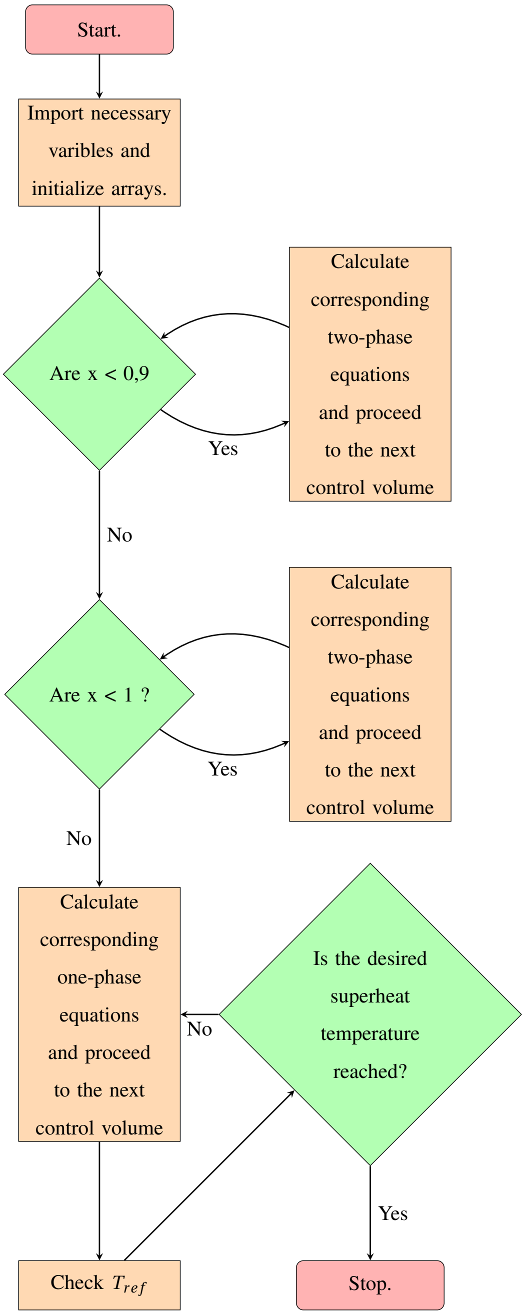
The hot side in the evaporator is the water being cooled. Thus, equation 4.8 is used to find the water's temperature change per length. The necessary input variables are given to the procedure, i.e inlet water temperature and evaporation temperature. These values are used to start the process of calculating the overall heat transfer coefficient U and the area A.

In the evaporator, heat flows from the water to the working fluid, increasing its quality and enthalpy. The working fluid enters the evaporator in a two-phase state after being throttled by the low stage throttling valve. The working fluid is boiled until it reaches saturated conditions. Then the gas is further heated to the desired superheat temperature. A superheat is needed to avoid feeding the low stage compressor with liquid. Separate heat transfer coefficients are used in the boiling phase and in the superheat phase. These heat transfer coefficients are used in the calculation of equation 4.8.

When the vapour quality reaches 0,9 the heat transfer coefficient decreases linearly down to 600 W/m^2K This is due to the vapour film on the plate walls, decreasing the heat transfer. (Tolstorebrov, 2015)

The temperature of the working fluid in the two-phase region is given by the pressure. Therefore, the temperature will drop due to the pressure loss through the heat exchanger. This is taken into consideration in the model. The water side pressure is not taken into account, since the energy consumption of the pump needed to force the water through the heat exchanger is outside the system boundary.

A flow chart of the evaporator procedure is shown on the next page:



To calculate the state equations described in section 4.4, the heat transfer coefficients for the water and working fluid must be found. There are a wide variety of correlations for heat transfer and pressure drop for plate heat exchangers found in literature. Summaries of these correlations can be found in (Eldeeb et al., 2015) and (Amalfi et al., 2015b). The correlations are found from experimental set ups, and are valid for only certain ranges of i.e mass flux and Reynolds numbers. The correlations used in this model would therefore not give an exact representation of real conditions. However, it is an useful approximation.

Water side heat transfer coefficient:

The water side heat transfer coefficient and friction factor coefficient is given by the Martin correlation (Martin, 1996) for one-phase flow. This correlation was the model which gave the most realistic values, compared to the Chisholm and Wanniarachchi or Kim correlations for one phase flow (García-Cascales et al., 2007).

The Nusselt number is given as:

$$Nu = 0,122Pr^{(1/3)}(\frac{\mu_m}{\mu_{wall}})^{(1/6)}(fRe^2 sin(2\beta))^{0,374} \tag{4.10}$$

The heat transfer coefficient is thus given by:

$$h_w = \frac{Nuk_w}{d_h} \tag{4.11}$$

where the d_h is the hydraulic diameter of the heat exchanger.

The friction factor f is found by:

$$\frac{1}{\sqrt{f}} = \frac{cos\beta}{(0,18tan\beta + 0,36sin\beta + \frac{f_0}{cos\beta})^{1/2}} + \frac{1 - cos\beta}{\sqrt{3,8f_1}} \tag{4.12}$$

where f_0 and f_1 are defined in the range of the Reynolds numbers:

$$Re < 2000 \Rightarrow \begin{cases} f_0 = \frac{64}{Re} \\ f_1 = \frac{597}{Re} + 3,85 \end{cases} \tag{4.13}$$

$$Re \geq 2000 \Rightarrow \begin{cases} f_0 = (1,8log_{10}Re - 1,5)^{-2} \\ f_1 = \frac{39}{Re^{0,289}} \end{cases} \tag{4.14}$$

Refrigerant side heat transfer coefficient and friction factor:

The boiling refrigerant's heat transfer coefficient is given by the correlation found in (Longo et al., 2015a). This correlation is based on a comparison of previous experimental data sets from other authors. The mean absolute percentage deviation between the experimental data and calculated date is around 20 %.

$$h_{tp} = 0,122 \left(\frac{k_l}{d_h} \right) Re_{eq}^{0,8} Pr_l^{1/3} \tag{4.15}$$

where the equivalent Reynolds number is given by:

$$Re_{eq} = \frac{G_{eq} d_h}{\mu_l} \tag{4.16}$$

The equivalent mass flux:

$$G_{eq} = G \left((1-x) + x \left(\frac{\rho_g}{\rho_l} \right) \right) \tag{4.17}$$

The mass flux:

$$G = \frac{\dot{m}}{A_{cross} N_{ch}} \tag{4.18}$$

Equation 4.15 is valid for $BoX_{tt} < 0,15 * 10^{-3}$, which indicates convective boiling. The boiling number Bo defined as:

$$Bo = \frac{q''}{G \Delta i_{lat}} \tag{4.19}$$

where q'' is the heat flux and X_{tt}, the Lockhart Martinelli parameter, is defined as:

$$X_{tt} = \left(\frac{1-x_m}{x_m} \right)^{0,9} \left(\frac{\rho_g}{\rho_l} \right)^{0,5} \left(\frac{\mu_l}{\mu_g} \right)^{0,1} \tag{4.20}$$

The two phase friction factor is given by the correlation proposed in (Amalfi et al., 2015a). This correlation is found by a best fit from a large experimental databank:

$$f_{tp} = C * 15,698 \left(\frac{G^2 D_h}{\rho_m \sigma} \right)^{-0,475} \left(\frac{(\rho_l - \rho_g) g D_h^2}{\sigma} \right)^{0,255} \left(\frac{\rho_l}{\rho_g} \right)^{-0,571} \tag{4.21}$$

where C is given by:

$$C = 2,125\left(\frac{\beta}{\beta_{max}}\right)^{9,993} + 0,955 \tag{4.22}$$

β_{max} is 70° and the mean density is:

$$\rho_m = \frac{1}{\frac{x_m}{\rho_g} + \frac{1-x_m}{\rho_l}} \tag{4.23}$$

The correlation used for the superheated vapour is the same as the correlation for water, namely the Martin correlation.

4.5.2 Condenser

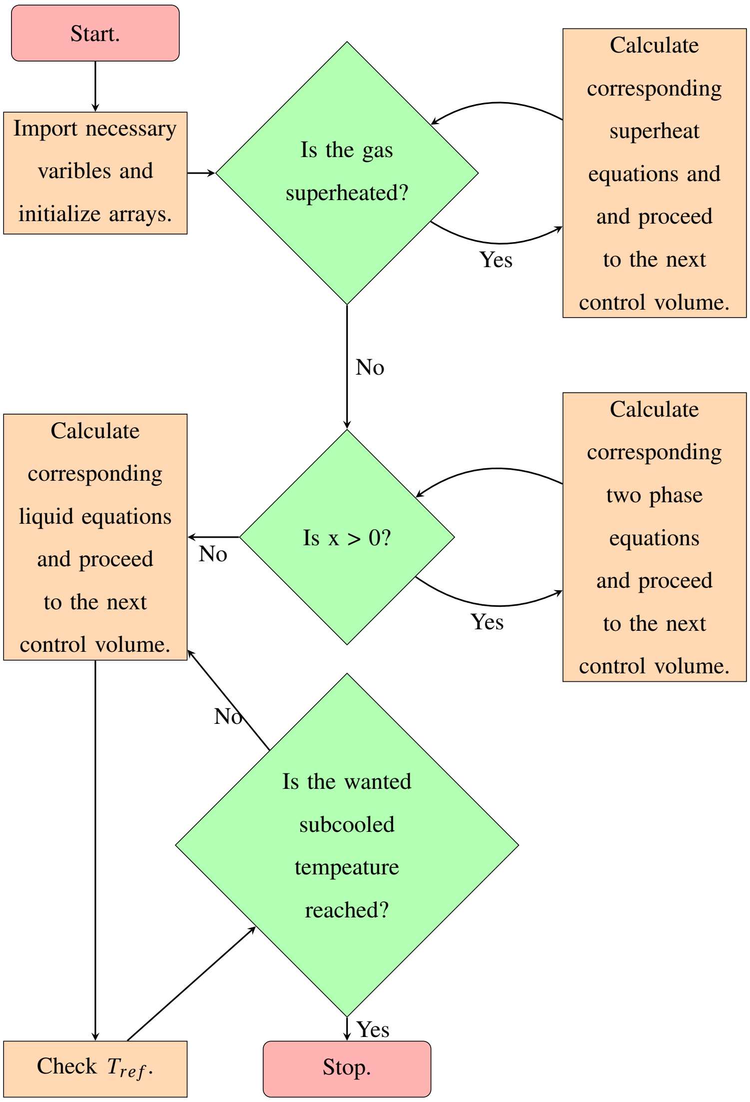
The condenser procedure is similar to the evaporator procedure, but with some differences.The superheated gas from the high pressure compressor enters the condenser inlet, and has to be cooled down to saturation temperature before condensation starts. The condensation continues until the working fluid reaches saturated liquid conditions.To prevent vapour flow into the high stage throttling valve, the working fluid is further sub-cooled to the desired temperature. Separate heat transfer coefficients are used for two-phase condensation and the one-phase cool down of the working fluid.

In the condenser, the heat flows from the working fluid to the water. Thus, the water side is the cold side of the heat exchanger. Equation 4.9 is used to find the water's temperature change per length.

The heat duty of the condenser is given by:

$$Q_{cond} = Q_{evap} + \dot{W}_{comp,LP} + \dot{W}_{comp,HP} \tag{4.24}$$

A flow chart of the procedure is shown below:



Superheat section

The water side heat transfer coefficient correlation is the same as the evaporator procedure, namely the one-phase Martin correlation.

The refrigerant side heat transfer coefficient and friction factor used is also the Martin correlation.

Condensing section

The two phase heat transfer coefficient is given by (Longo et al., 2015b). This correlation is proposed on the basis of several other data sets, and focused on hydrocarbon condensation. The enlargement factor described in the paper is in this model assumed to be one. Note that this correlation is found by experiments with herringbone corrugated plates:

$$h_{tp} = 1,875\left(\frac{k_l}{d_h}\right) Re_{eq}^{0,445} Pr_l^{1/3} \tag{4.25}$$

The frictional pressure drop is found by the kinetic model described in (Longo, 2010):

$$KE/V = \frac{G^2}{2\rho_m} \tag{4.26}$$

and

$$\Delta p_{fric}[kPa] = 1,90 KE/V \tag{4.27}$$

which gives

$$\Delta p_{fric} = 1,90\left(\frac{G^2}{2\rho_m}\right) \tag{4.28}$$

The friction factor shows a linear dependence on the kinetic energy per unit volume. This model cannot calculate the pressure loss per control volume. The temperature of the working fluid will therefore not be exactly correct. However, the pressure drop through the condenser is very small. The temperature drop during the condensation phase is negligible.

Subcooled section: The correlations for heat transfer coefficient and friction factors for the sub-cooled refrigerant is given by the Martin correlation.

4.5.3 Pressure drop

The total pressure drop in the heat exchangers is the sum of several smaller pressure drops (Amalfi et al., 2015b):

- Frictional pressure loss ΔP_{fric}
- Gravitational pressure loss ΔP_{grav}
- Acceleration pressure loss ΔP_{acc}
- Manifold pressure loss ΔP_{man}

The total pressure loss in the evaporator is given by:

$$\Delta P_{tot} = \Delta P_{fric} + \Delta P_{grav} + \Delta P_{acc} + \Delta P_{man} \tag{4.29}$$

The total pressure loss in the condenser is given by:

$$\Delta P_{tot} = \Delta P_{fric} - \Delta P_{grav} - \Delta P_{acc} + \Delta P_{man} \tag{4.30}$$

The two-phase frictional pressure drop is the pressure drop due to the roughness of the channel walls and molecules colliding. It is given by:

$$\Delta P_{fric} = \frac{2 f_{tp} G^2 \Delta y}{d_h \rho_m} \tag{4.31}$$

Gravitational pressure loss ΔP_g is the loss or gain due to the effects of gravity. In the condenser the fluid enters the top of the heat exchanger. Pressure is gained rather than lost. The opposite happens in the evaporator.

$$\Delta P_{grav} = \rho_m g L_{plate} \tag{4.32}$$

ΔP_{acc} is the loss or gain due to either acceleration or deacceleration, respectively. In the condenser the gas in deaccelerated and pressure is gained rather than lost. In the evaporator, there is a loss due to acceleration.

$$\Delta P_{acc} = G^2 \Delta x \left(\frac{1}{\rho_g} - \frac{1}{\rho_l} \right) \tag{4.33}$$

The manifold pressure loss ΔP_{man} is the loss due the fluid entering the inlet and outlet ports and manifolds.

$$\Delta P_p = 1,5 \frac{\rho_m \left(\frac{G}{\rho_m}\right)^2}{2} \tag{4.34}$$

4.6 Pipes

The pipes connect the various components in the system. The pressure loss through the pipes influences the compressor work.

As a general rule, the temperature drop of the working fluid through the pipes should be kept below 2° C. The corresponding temperature drop due to a pressure drop can be found by plotting the saturation pressure versus the temperature. A function describing this plot can then be used to calculate a pressure's corresponding saturation temperature. A polynomial function of degree six is found by EES's CurveFit function. This function does not describe the relationship between the pressure and temperature accurately enough for the whole range from 0° C to T_{crit}. However, for a smaller interval, i.e from 75°C to 115°C, it is sufficiently accurate.

An example of a plot which is described above is seen in figure 4.5 for R600a:

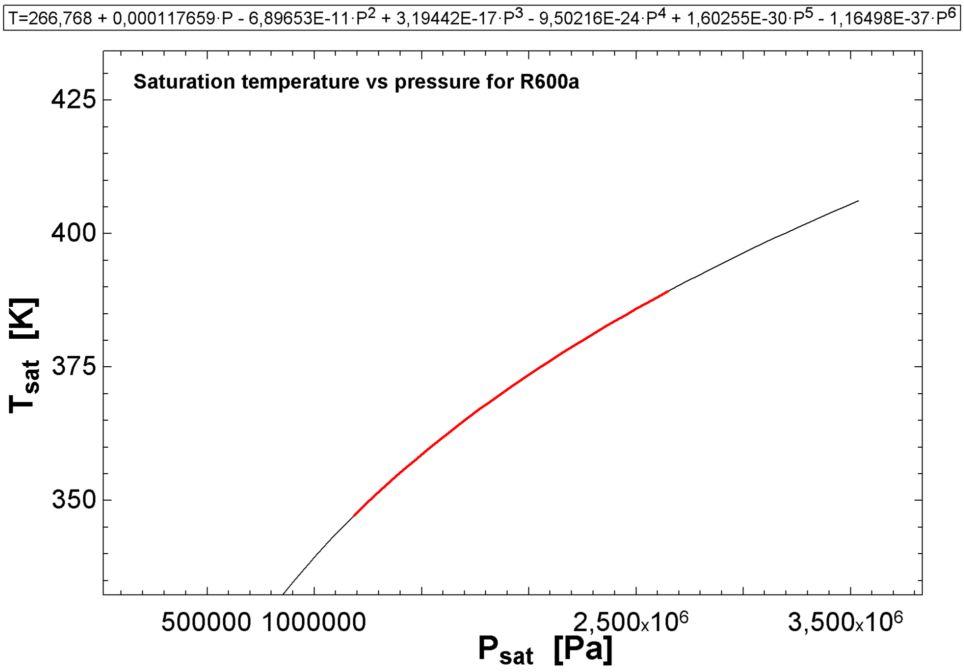


Figure 4.5: Saturation temperature versus pressure for R600a.

Each working fluid has a particular function for the relationship between the saturation pressure and temperature. The pressure loss is found by the built in EES function ”Pipe Flow”. The pressure loss is calculated for each pipe segment, and the corresponding temperature loss is then calculated by applying the method described above.

Another criteria for the pipe dimensions are the required flow velocity. The fluid velocity must be high enough for the lubricant oil to be circulated sufficiently. If the velocity is too low, the working fluid is not able to carry the lubricant oil droplets. It would lead to an oil build up somewhere in the system which can lead to decreased performance, and potentially lead to system failure. Rules of thumb have been developed to prevent such build ups. For vertical pipes, the following should be adhered to:

For gasous flow, equation 4.35 must be satisfied (Bäckström, 1946):

$$\rho_g u_g^2 \geq 126 \tag{4.35}$$

Using equation 4.35 for R600a with input variables $\dot{m}$ = 1,5 kg/s , L_{pipe} = 5 m ,T_{ref} = 75°C and relative roughness $\epsilon = 5{,}0 * 10^5$ results in a maximum inner diameter of 17,4 cm for a gasous flow.

For liquid flow, a minimum velocity of 0,5 m/s is needed.
For a saturated liquid flow from the condenser with input variables T_{ref} = 115°C and $\dot{m}$ = 2,1 kg/s, the minimum diameter needed to have a fluid velocity larger than 0,5 m/s is 25,5 cm.

4.7 Compressors

The model has two compressors, one for the low pressure loop, and one for the high pressure loop. The built in EES function ”Compressor2_CL” from the program’s component library is used to calculated the compressors’ outlet enthalpy and power requirement. The function assumes a constant isentropic efficiency. An isentropic efficiency of 0,7 is used for both compressors.
The function needs several inputs. Namely; the inlet enthalpy, the inlet and outlet pressures, the mass flow, the working fluid being compressed, and the efficiency. The outputs are the outlet

enthalpy and power usage. This is shown below:

Call Compressor2_CL(i_{in}, P_{in}, P_{out}, m_{dot}, F$, η:i_{out}, W_{dot}, η_s)

4.8 Iterative optimization

To initiate the variables in the program, a guess is made. This first estimate will establish the variables, which will be the starting point of optimization. This optimization is done by minimizing the error function which gives the correct values.

Evaporator

To find the mass flow in the lower loop of the heat pump, an error function is minimized to adjust the discrepancy between the wanted heat duty and the actual heat duty caused by the current mass flow chosen. The heat duty of the evaporator is given by:

$$Q_{evap} = \dot{m}_{ref,LP} \Delta i_{lat} \tag{4.36}$$

The error function:

$$err_{heat,evap} = abs(\dot{q}_{neeed,evap} - \dot{q}_{w,evap}) \tag{4.37}$$

where

$$\dot{q}_{needed,evap} = \dot{m}_{w,evap}(i_{w,in,evap} - i_{w,out,needed,evap}) \tag{4.38}$$

and

$$\dot{q}_{w,evap} = \dot{m}_{w,evap}(i_{w,in,evap} - i_{w,out,actual,evap}) \tag{4.39}$$

The mass flow rate and inlet and outlet enthalpy of the water are given from the boundary conditions. The working fluids' mass flow rate influences the outlet enthalpy of the water. Utilizing EES' in-built min/max function, equation 4.37 is minimized by varying $\dot{m}_{ref,LP}$.

Condenser

In the condenser, the working fluid's mass flow rate is given by the mass flow of the lower loop through the relationship below:

$$\dot{m}_{ref,HP} = \dot{m}_{ref,LP} \frac{i_{out,comp,LP} - i_{ref,out,int,liq}}{i_{ref,out,int,vapour} - i_{ref,HT}} \tag{4.40}$$

The error function minimizes the discrepancy between the needed heat duty and the actual heat duty calculated. This is done by varying the mass flow of the water.

$$err_{heat,cond} = abs(\dot{q}_{neeed,cond} - \dot{q}_{w,cond}) \tag{4.41}$$

where

$$\dot{q}_{w,cond} = \dot{m}_{w,cond}(i_{w,out,actual,cond} - i_{w,in,cond}) \tag{4.42}$$

and $\dot{q}_{neeed,cond}$ is given by equation 4.24.

Initially, the number of plates are guessed. As an output from the calculations, the procedures for evaporator and condenser will give the plate length required to facilitate the wanted heat duty. If the plate length calculated by the procedure is too large, more plates and thus more area must be added. If the length is too short, the area is too large, and plates must be removed.The length of the plates are adjusted by varying the number of plates of both evaporator and condenser. The error function is shown below:

$$err_{length} = abs(L_{plate} - L_{plate,out}) \tag{4.43}$$

Compressors

The pressure loss in the pipes must be taken into consideration when using the compressor function described in section 4.7. The function requires the outlet pressure as an input. But this pressure is not known before the pressure loss in the pipes downstream is calculated. To initiate the iterative process, an estimate is given. Then function 4.44 is minimized, and the guess is updated.

$$err_{comp} = abs(\Delta P_{guess,comp} - \Delta P_{pipe}) \tag{4.44}$$

This is done for both the low pressure and high pressure stage compressors.

Chapter 5

Economic model

5.1 General

The total costs of the system are determined by two factors. The investment costs and the operational costs. Higher investment costs leads to lower operational costs, and vice versa. The most cost efficient system is the one with the lowest total yearly costs.
Total yearly costs are given by (Banooni et al., 2014):

$$E_{tot} = aE_{inv} + E_{op} \tag{5.1}$$

where a is the annuity factor. This factor gives the yearly costs of the investment, taking into account the rent and annual payments to repay the investment. It is given by:

$$a = \frac{a}{1-(1+r)^{-n}} \tag{5.2}$$

The investment costs are the costs of heat exchangers, pipes and compressors. The costs of valves and the intermediate pressure vessel are omitted.

$$E_{inv} = E_{inv,comp,LP} + E_{inv,comp,HP} + E_{evap} + E_{cond} + E_{pipes} \tag{5.3}$$

The operational costs are given by the electricity consumption of the compressors. The maintenance costs are omitted.

$$E_{op} = E_{op,comp,LP} + E_{op,comp,HP} \tag{5.4}$$

5.2 Compressors

Investment costs

The investment costs of the compressors are based on the displacement volume needed. A correlation between the price and displacement gives a function for the price of the compressor based on the volume which needs to be displaced. The relation is based on the price of an open reciprocating Bitzer compressor. The prices are given by Schløsser Kulde AS. This relationship is shown in figure 5.1

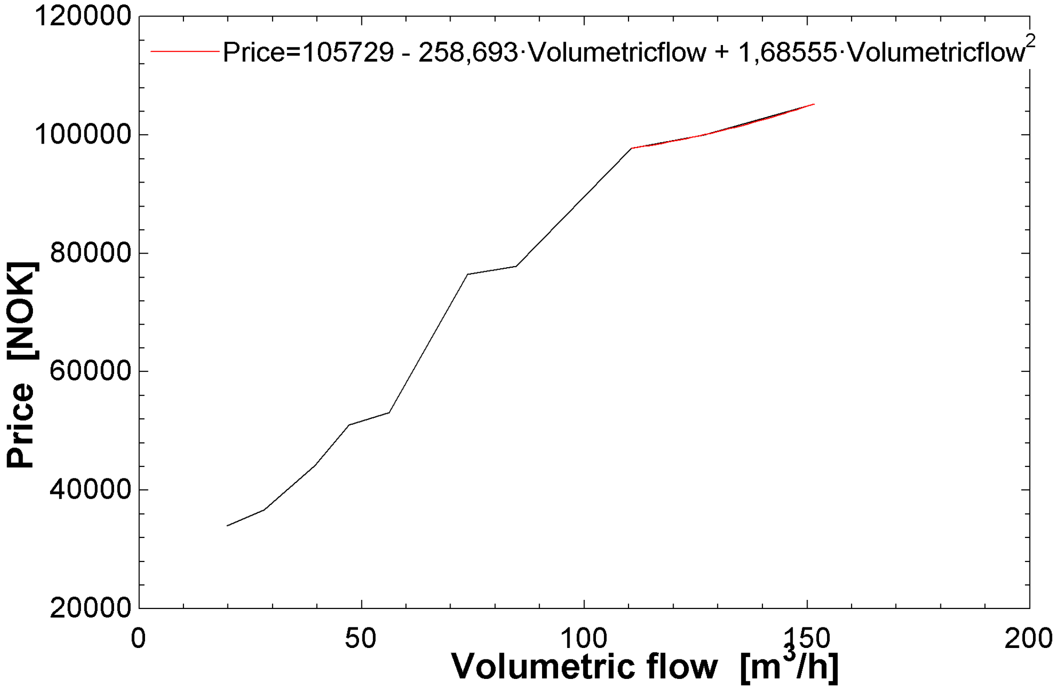


Figure 5.1: Price versus volumetric flow.

Only the upper part of the graph is given as arguments for the equation. The price changes non-linearly due to the increased number of cylinders, ranging from 2 to 6. The price per m^3/h displaced decreases when the size of the compressor increases.

Operational costs

The operational costs are given by the electricity consumption of the compressors.

$$E_{comp} = p_{el}\tau\frac{\dot{W}_{comp}}{\eta_{comp}} \tag{5.5}$$

where η_{comp} is given by:

$$\eta_{comp} = \eta_{is}\eta_{el} \tag{5.6}$$

The total electricity consumption is more than the work needed to compress the working fluid itself. Irreversibilities of the compression process and losses in the electric motor driving the compressor increases the electricity consumption. Assumptions for this model are $\eta_{is} = 0{,}7$ and $\eta_{el} = 0{,}95$.

5.3 Pipes

Investment costs

Copper pipes are used in this model. The price of the pipes is given as a function of the mass of material used. A pipe with a larger diameter will have a higher weight, and thus higher cost. The prices and dimensions used in the following relationships are given by Schløsser Kulde AS.

The weight of the pipe if found by calculating the outside diameter given the inside diameter. The ratio between the outside and inside diameter decreases as the inside diameter increases. The relationship is shown in figure 5.2.

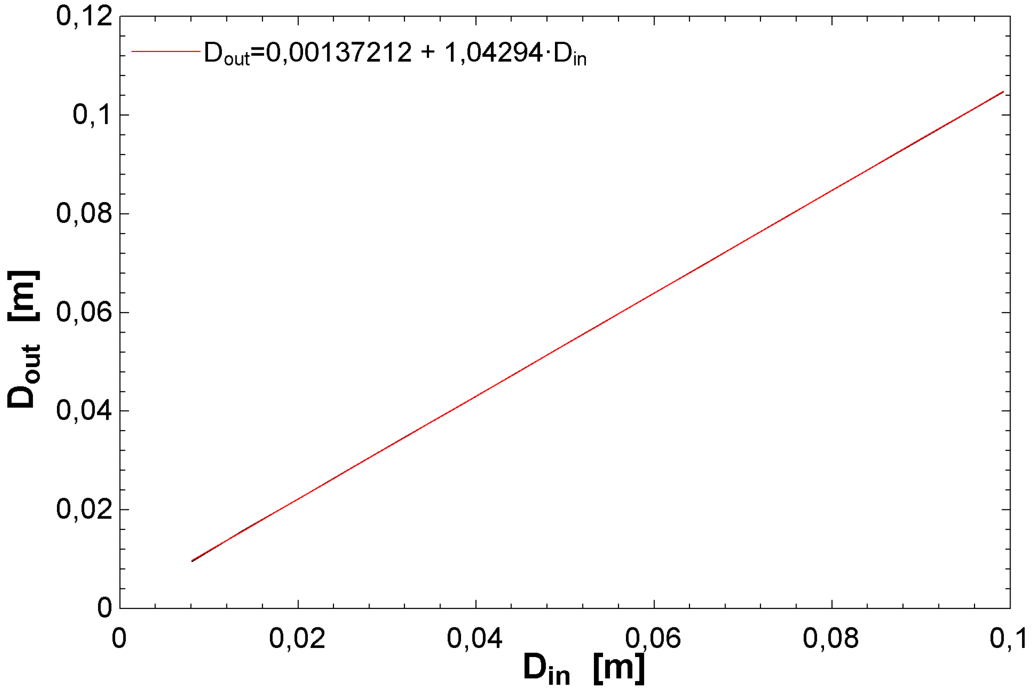


Figure 5.2: D_{out} versus D_{in}

When the outlet diameter is given, the weight per meter tube is found:

$$m_{pipe} = \frac{\rho_{pipe}\pi(D_{out}^2 - D_{in}^2)}{4} \tag{5.7}$$

The price per meter is then found by the relationship between the weight and price shown in figure 5.3.

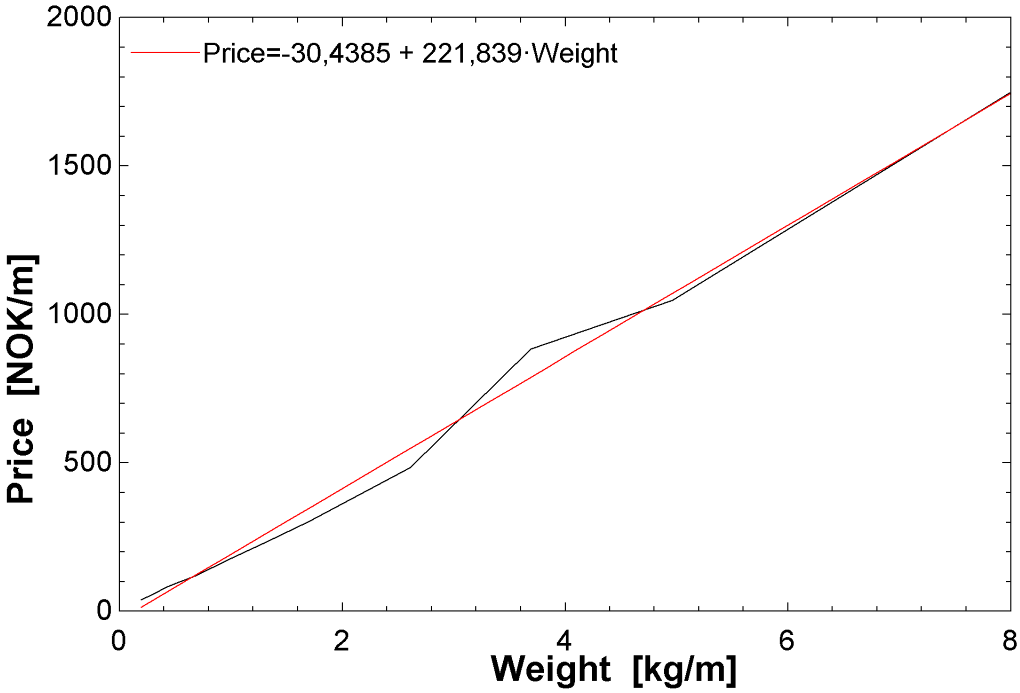


Figure 5.3: Price versus weight for copper pipes.

Operational costs

The operational costs of the compressors are directly linked to the pressure drop through tubes. A higher pipe diameter will decrease the pressure loss through the pipes, and will thus decrease the work done by the compressors.

In figure 5.4, the pressure loss per meter pipe versus the inside diameter of a copper pipe is shown. This particular case is for saturated R600 vapour at 78°C. The mass flow is 1,5 kg/s.

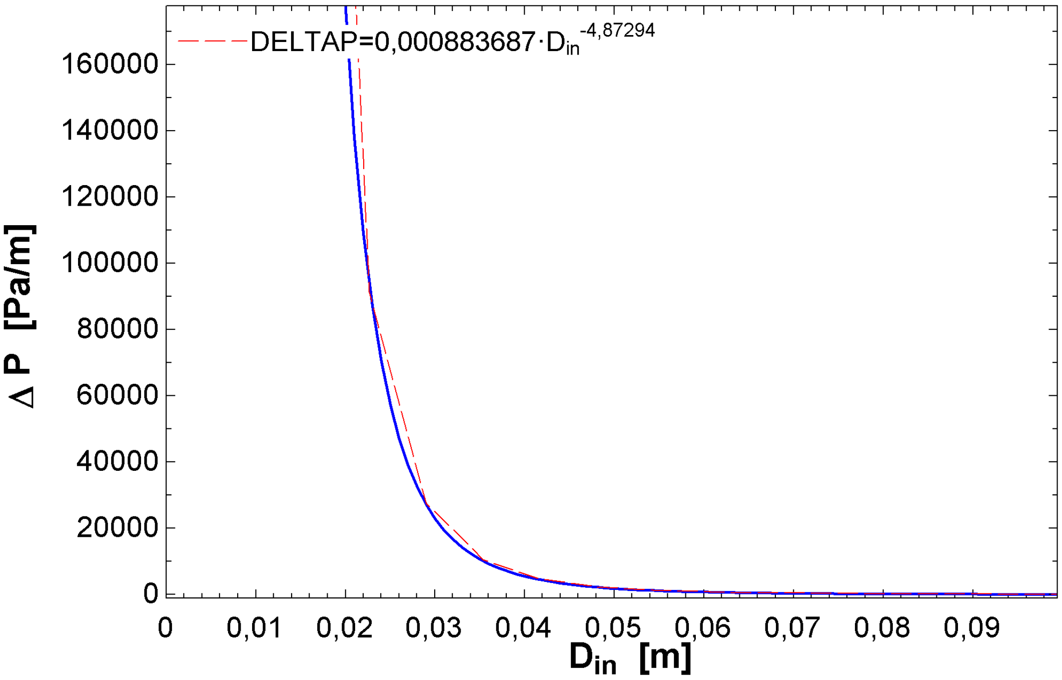


Figure 5.4: Pressure loss per meter versus D_{in}

It is clear that the pressure loss decreases asymptotically towards 0. As a consequence, there is no reason to have a too large pipe diameter. The money saved on decreased work load of the compressor is equalized by an increased investment cost.

5.4 Heat exchangers

Investment costs

The price of the heat exchangers are closely related to the area of the heat exchanger, and thus the number of plates. A correlation between the price and area of the heat exchanger is given in figure 5.5. The prices are given by Schløsser Kulde AS for a GEA BP 110 brazed heat exchanger. The corrugation pattern of the plates is not given.

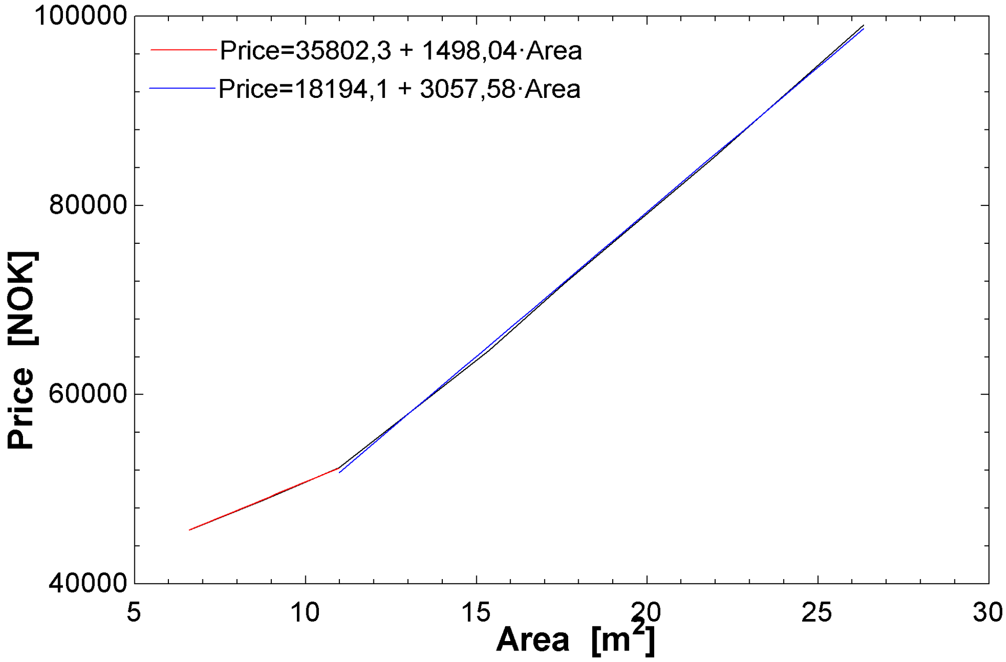


Figure 5.5: Price versus area.

Two different equations are used, depending on the total number of plates. For the lower part of the graph, the cost per m^2 heat exchange surface is higher because the cost of the frame itself constitutes a higher part of the total costs.

Operational costs

The heat exchanger itself does not have any explicit operational costs. However, it influences the compressor work needed. For a given duty, a smaller pinch temperature can be used if the area of the heat exchanger increases. This means that the compression ratios of the compressors get smaller, and thus less work is needed. Further, a higher pressure loss through the heat exchanger gives a higher overall heat transfer coefficient and subsequently smaller area , but increased compressor work. A balance must be found considering these factors to find the optimal values.

5.5 Economic case

Using the economic model in conjunction with the heat pump model, R600 and R600a was tested. The operating time of the heat pump was assumed to be 8000 hours a year, corresponding to an all year round operation with maintenance breaks. The superheat of the working fluid from the evaporator was 3 K. The sub-cool temperature out of the condenser was 1 K The evaporator temperature was varied from 76°C to73°C to see how it influenced the COP and evaporator area. Both investment costs and operational costs were found for the working medias. To find the annual costs of investment, the interest rate was assumed to be 0,1 and the depreciation time n to be 15 years.

To asses how the price of electricity influenced the total system costs, electricity prices ranging from 0,5 NOK/kWh to 0,8 NOK/kWh were tested in the model. The electricity price consists of two parts, the energy price and transmission cost. A price of 0,8 NOK/kWh is conservative estimate. The transmission costs for 2015 were 0,4 NOK/kWh (Hafslund, 2016). The energy price was 0,2 NOK/kWh (Nordpool, 2016). The price changes through the year, but were assumed to be constant in this model. A price of 0,5 NOK/kWh is not realistic, but tested to see if the price of extra heat exchanger surface would exceed the money saved on less compression work.

Chapter 6

Results

The EES model is used to find the COP values for the different pinch temperatures along with the corresponding costs. The results are found below:

The main results using R600:

T_{ref} °C	$A_{evap} [m^2]$	$A_{cond} [m^2]$	COP	$W_{comp,tot}$ [kW]	U_{cond} $[W/m^2K]$	$T_{loss,evap}$ °C
76	20,84	18,65	5,996	60,32	2900	-0,20
75	13,6	18,65	5,789	62,58	3418	-0,33
74	9,435	19,31	5,642	64,96	4111	-0,50
73	6,802	19,09	5,47	67,30	4886	-0,74

Table 6.1: R600.

The main results using R600a:

T_{ref} °C	$A_{evap} [m^2]$	$A_{cond} [m^2]$	COP	$W_{comp,tot}$ [kW]	U_{cond} $[W/m^2K]$	$T_{loss,evap}$ °C
76	21,72	19,75	5,531	66,05	2849	-0,18
75	14,26	20,19	5,39	68,53	3250	-0,28
74	9,874	20,63	5,228	71,12	3871	-0,42
73	7,021	21,06	5,09	73,4	4677	-0,63

Table 6.2: R600a.

The COP versus the evaporator temperature.

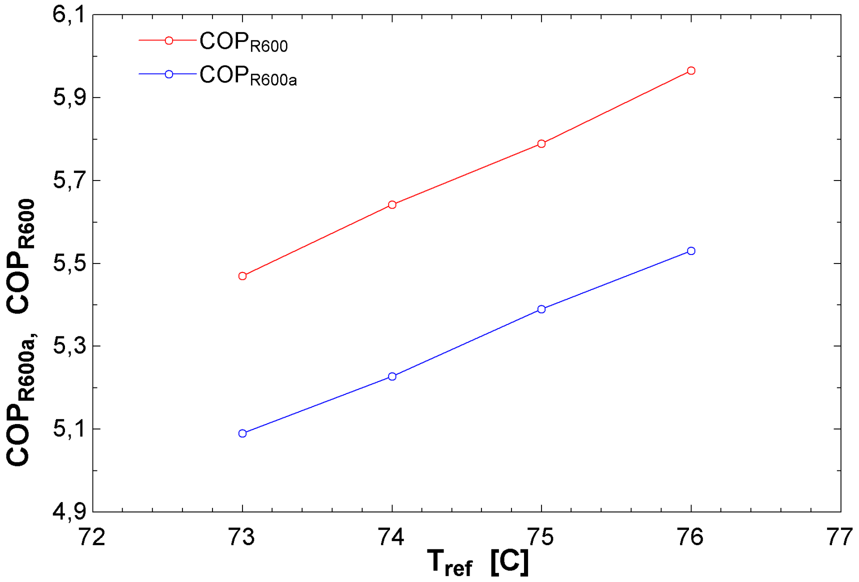


Figure 6.1: The COP vs evaporator temperature.

The total yearly costs versus the evaporator temperature for R600 is shown below:

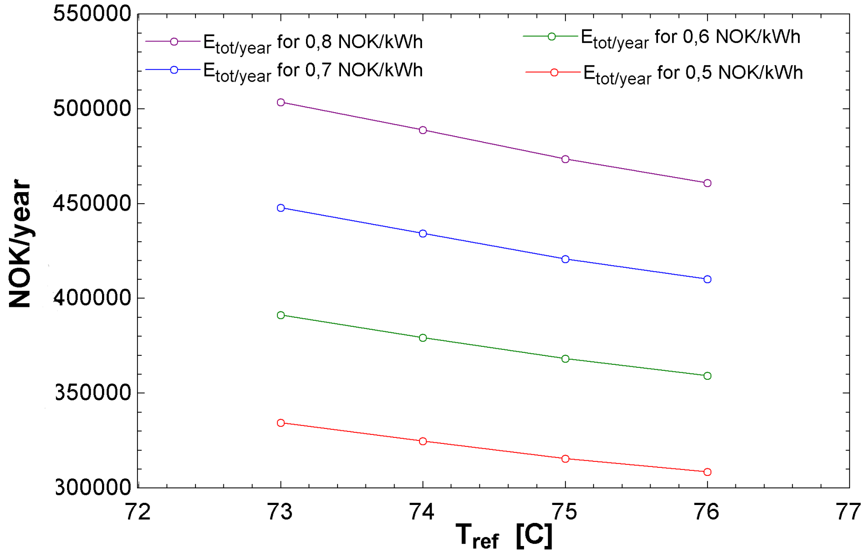


Figure 6.2: Total yearly costs versus the evaporator temperature for R600.

The total yearly costs versus the evaporator temperature for R600 is shown below:

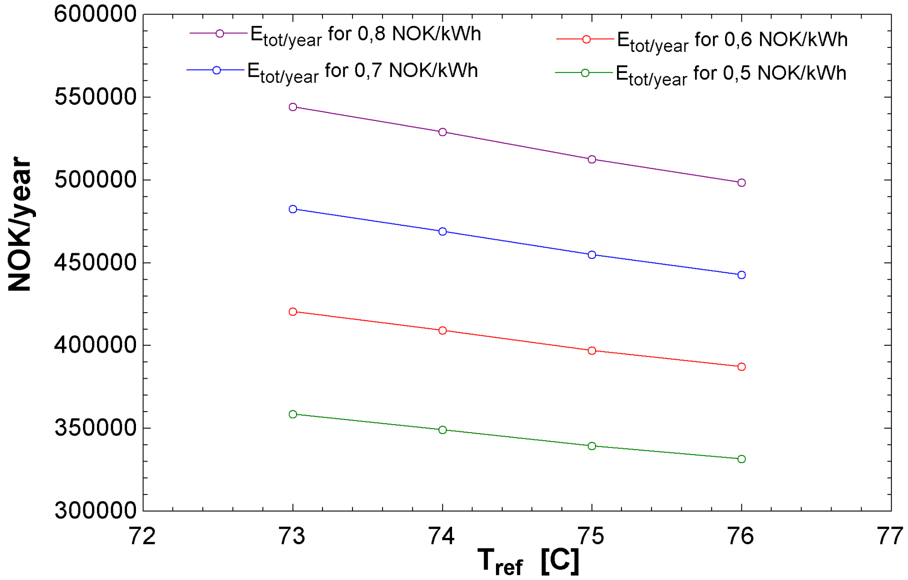


Figure 6.3: Total yearly costs versus the evaporator temperature for R600a.

The sensitivity analysis for the electric price is shown in the following tables:

R600:

T_{ref} °C	E_{inv}	E_{op}	$E_{tot}/year$	$E_{inv}/year$
76	414862	406377	460920	54543,5
75	395787	421560	473596	52035,6
74	390677	437599	488963	51363,8
73	389476	453365	503571	51205,9

Table 6.3: R600 withp_{el} = 0,8 NOK/kWh.

T_{ref} °C	E_{inv}	E_{op}	$E_{tot}/year$	$E_{inv}/year$
76	414862	355580	410123	54543,5
75	395787	368865	420901	52035,6
74	390677	382899	434263	51363,8
73	389476	396695	447900	51205,9

Table 6.4: R600 withp_{el} = 0,7 NOK/kWh.

T_{ref} °C	E_{inv}	E_{op}	$E_{tot}/year$	$E_{inv}/year$
76	414862	304782	359326	54543,6
75	395787	316170	368206	52035,6
74	390677	328199	379463	51363,8
73	389476	340024	391230	51205,9

Table 6.5: R600 withp_{el} = 0,6 NOK/kWh.

T_{ref} °C	E_{inv}	E_{op}	$E_{tot}/year$	$E_{inv}/year$
76	414862	253985	308529	54543,5
75	395787	263475	315511	52035,6
74	390677	273499	324863	51363,8
73	389476	285353	334559	51205,9

Table 6.6: R600 withp_{el} = 0,5 NOK/kWh.

R600a:

T_{ref} °C	E_{inv}	E_{op}	$E_{tot}/year$	$E_{inv}/year$
76	407111	444977	498502	53524,4
75	387681	461660	512630	50969,9
74	379385	479152	529031	49879,2
73	378316	494491	544230	49738,6

Table 6.7: R600a withp_{el} = 0,8 NOK/kWh.

T_{ref} °C	E_{inv}	E_{op}	$E_{tot}/year$	$E_{inv}/year$
76	407111	389355	442880	53524,4
75	387681	403952	454922	50969,9
74	379385	419258	469137	49879,2
73	378316	432680	482680	49738,6

Table 6.8: R600a withp_{el} = 0,7 NOK/kWh.

T_{ref} °C	E_{inv}	E_{op}	$E_{tot}/year$	$E_{inv}/year$
76	407111	333733	387257	53524,4
75	387681	346245	397215	50969,9
74	379385	359364	409243	49879,2
73	378316	370868	420607	49738,6

Table 6.9: R600a withp_{el} = 0,6 NOK/kWh.

T_{ref} °C	E_{inv}	E_{op}	$E_{tot}/year$	$E_{inv}/year$
76	407111	278111	331635	53524,4
75	387681	278111	331635	50969,9
74	379385	278111	331635	49879,2
73	378316	278111	331635	49738,6

Table 6.10: R600a withp_{el} = 0,5 NOK/kWh.

Chapter 7

Discussion

7.1 Discussion

The results shows a good feasibility for the application of a heat pump for the case discussed.A large energy saving in combination with the possibility to adhere to the environmental requirements, shows this is a good investment. Both R600 and R600a has COP values higher than 5. R600 has a COP of almost 6 for the smallest pinch temperature tested.

Both working fluids will give an energy saving above 80 % with a COP of 5 and above, as seen by equation 2.3. It is important to remember that the difference in energy saving between a COP of 5 and 6 is only 3,3 %. For an electricity price of 0,8 NOK/kWh, the heat pump with R600 has a yearly costs of 461000 NOK for the lowest pinch temperature. This is 37500 NOK lower than the heat pump with R600a.

Looking at the total yearly costs, it is clear that the optimal pinch temperature in the evaporator is as small as possible. The added cost of heat exchanger surface never exceeded the savings by less compression work. In addition, increased compression work increases the heat which need to be removed in the condenser, rising the condenser costs. The lowest possible pinch temperature would depend on the needed superheat from the evaporator. The pinch temperature can never be smaller than the needed superheat, since it would mean a temperature cross in the evaporator.

A source of error is the heat transfer coefficients and friction factors used in the calculation.

Most of the correlations found were done by experiments where the temperature of the working fluid was much lower. Typically with saturation temperatures below 40°C. The pressure drop through the heat exchangers are also uncertain. The correlations used were found for a specific plate geometry and flow conditions, which does not fully reflect the operating conditions of this model.

The simulation model itself is not totally perfect, with some small errors. For instance in calculating the pressure drop in the condenser. Another error is the statement which terminates the superheat loop in the evaporator. The pressure used to calculate the enthalpy is the inlet pressure, not the outlet pressure.

The water side of the heat exchangers have not been the primary focus. It is a possibility that the mass fluxes are too large for realistic conditions. This is not investigated further in this thesis.

It is likely that the investment cost of the compressors is too low. The price is an extrapolation of the trend of the the 6-cylinder compressor. More likely, it should be a jump up to a 8-cylinder version needed to handle the high volume flow. The isentropic efficiency of the compressors are assumed to be 0,7, which is conservative. This number could be higher for state of the art compressors, which would reduce the operational costs further.

Instead of using two compressors, it would be possible to use a screw compressor with liquid injection for this system. This may reduce the investment costs, since only one compressor is used instead of two.

The total investment costs of the system would be higher, since the flash tank and expansion valves are omitted. These costs are not significant though, and would be the same for both working fluids. The sensitivity analysis done by varying the electricity price show that the operational costs are much more important than the investment costs. The yearly investment costs are five to six times smaller than the yearly operational costs. A realistic electricity price is 0,8 NOK/kWh based on the Norpool spot price plus transmission costs. In countries with higher electricity prices than Norway, the benefit of having small pinch temperatures is even larger.

The pipe diameter is probably not optimal with respect to the diameter versus pressure drop. However, a diameter of 7 cm should be a decent trade off between pressure drop and price, according to figure 5.4.

The temperature glide of the water is 10°C in the evaporator and 15°C in the condenser. An interesting option which can reduce the exergy loss due to poor temperature fit is the hybrid heat pump. This would lead to an excellent temperature fit in this particular application.

7.2 Conclusion

Industrial heat pumps give large energy and cost savings. This is shown for one case in this thesis. An industrial heat pump is integrated into a distillation process to both cool down a stream to fulfill environmental demands, while at the same time use the waste heat to pre-heat a distillation feed. Both working fluids tested gave good results. The simulation model shows that R600 gives overall a better performance than R600a. For the lowest pinch temperature, R600 has a coefficient of performance close to 6. This would give an energy saving of around 83 %. The heat pump with R600 has a yearly costs of 461000 NOK for the lowest pinch temperature compared to the solution with R600a which costs 498500 NOK per year.

The requirement for this COP is a large evaporator which is able to give a small pinch temperature between the working fluid and the water. However, a high investment cost gives a lower annual operating costs. The total yearly costs including both the investment and operational costs is therefore lower. Even though it is not a perfect simulation model, it gives a good indication of the feasibility of industrial heat pumps in general.

7.3 Further work

The two reciprocating compressors could be replaced with a single screw compressor with liquid injection, to see how the performance and operating costs will change. In addition, other system configurations can be applied.

Pipe insulation costs can be added, and a function for the optimal pipe diameter versus compressor costs due to the pressure drop can be found.

If found, other heat transfer and friction factor correlations can be applied if they are more suitable for high temperature applications.

The simulation model can be elaborated further, finding a better way of calculating the pressure drop in the condenser. Also, some of the error functions could be automated, saving time when testing different heat pump parameters.

The economic model can be elaborated by including costs such as maintenance, valves and other necessary equipment. Prices from several firms can be collected and analyzed.

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FLASHTANKSYSTEM

Equations

`$UnitSystem SI MASS RAD PA K J`

`$Tabstops 0.2 0.3 0.6 3.5 in`

————————PROCEDURES————————

————————-EVAPORATOR PROCEDURE————————-

procedure $EVAP(T_{water;in;evap};\ T_{water;out;evap};\ P_{water;evap};\ P_{ref;evap};\ P_{ref;cond};\ T_{ref;cond};\ T_{ref;evap};\ W\$;\ R\$)$;
$\dot{m}_{water;evap};\ N_{ch;evap};\ th_{water;evap};\ W_{evap};\ \dot{m}_{ref;LP};\ th_{ref;evap};\ th_{m;evap};\ stepsize;\ d_{h;evap};\ A_{cross;evap};$
$\beta_{evap};\ i_{ref;LT};\ P_{LT};\ i_{ref;out;int;liq}$ (1)

$L_{out;evap};\ i_{ref;out;evap};\ i_{ref;evap;1};\ i_{w;out;evap};\ P_{evap;out};\ T_{ref;out;evap};\ \Delta P_{evap;all})$ (2)

`$ARRAYS ON`

$y_{evap;1} = 0$ starting point (3)

Water side

$T_{w;evap;1} = T_{water;in;evap}$ inlet water temperature (4)

$i_{w;evap;1} = \mathrm{h}\left(W\$;\ \mathrm{T} = T_{w;evap;1};\ \mathrm{P} = P_{water;evap}\right)$ inlet water enthalpy (5)

Refrigerant side

$i_{ref;evap;1} = i_{ref;LT}$ Inlet refrigerant enthalpy (6)

$x_{ref;evap;1} = \mathrm{x}\left(R\$;\ \mathrm{T} = T_{ref;evap};\ \mathrm{h} = i_{ref;evap;1}\right)$ inlet refrigerant quality (7)

$T_{ref;evap;1} = T_{ref;evap}$ inlet refrigerant temperature (8)

$P_{ref;evap;local;1} = P_{ref;evap}$ refrigerant pressure (9)

Other

$\dot{q}_{evap;1} = 0$ Starting point for heat transfer (10)

$k_{m;evap;1} = k\left(\text{‘Stainless\_AISI316’}\ ;\ \frac{T_{ref;evap;1} + T_{w;evap;1}}{2}\right)$ metal conductivity at local average temperature at starting point(11

$c_{water;evap;1} = \mathrm{c_p}\left(W\$;\ \mathrm{P} = P_{water;evap};\ \mathrm{x} = 0\right)$ water specific heat capacity at starting point (12)

$T_{superheat} = 3\ [\mathrm{K}]$ superheat temperature (13)

————-HEAT TRANSFER COEFFICIENTS————

Martin correlation for one phase flow

$$G_{w;evap} = \frac{\dot{m}_{water;evap}}{A_{cross;evap} \cdot N_{ch;evap}} \qquad \text{mass flux water} \tag{14}$$

$$\mu_{l;water;1} = \mu\left(W\$;\ \mathrm{T} = T_{w;evap;1};\ \mathrm{x} = 0\right) \qquad \text{viscosity of liquid phase} \tag{15}$$

$$\mu_{wall;water;evap;1} = \mu\left(W\$;\ \mathrm{T} = \left(\frac{T_{ref;evap;1} + T_{w;evap;1}}{2}\right);\ \mathrm{x} = 0\right) \qquad \text{viscosity at wall} \tag{16}$$

$$Re_{w;evap;1} = \frac{G_{w;evap} \cdot d_{h;evap}}{\mu_{l;water;1}} \qquad \text{reynolds number} \tag{17}$$

$$Pr_{w;1} = \mathrm{P_r}\left(W\$;\ \mathrm{T} = T_{w;evap;1};\ \mathrm{x} = 0\right) \qquad \text{prandtl number} \tag{18}$$

$$f0_{martin;1} = \left(1,8 \cdot \log 10(Re_{w;evap;1}) - 1,5\right)^{-2} \qquad \text{parameter for friction factor calculation} \tag{19}$$

$$f1_{martin;1} = \frac{39}{\left(Re_{w;evap;1}^{0,289}\right)} \qquad \text{parameter for friction factor calculation} \tag{20}$$

$$f_{martin;1} = \left(\frac{1}{\left(\frac{\mathrm{Cos}(\beta_{evap})}{\sqrt{0{,}18 \cdot \tan(\beta_{evap}) + 0{,}36 \cdot \sin(\beta_{evap}) + \left(\frac{f0_{martin;1}}{\mathrm{Cos}(\beta_{evap})}\right)}} + \left(\frac{1 - \mathrm{Cos}(\beta_{evap})}{\sqrt{(3{,}8 \cdot f1_{martin;1})}}\right)\right)}\right)^2 \qquad \text{friction factor} \tag{21}$$

$$Nusselt_{w;1} = 0,122 \cdot \left(Pr_{w;1}^{1/3}\right) \cdot \left(\left(\left(\frac{\mu_{l;water;1}}{\mu_{wall;water;evap;1}}\right)^{1/6}\right)\right) \cdot \left(f_{martin;1} \cdot \left(\left(Re_{w;evap;1}^2\right) \cdot \sin(2 \cdot \beta_{evap})\right)\right)^{0,374} \qquad \text{Nusselt number}$$

$$k_{w;1} = \mathrm{k}\left(W\$;\ \mathrm{T} = T_{w;evap;1};\ \mathrm{x} = 0\right) \qquad \text{water conductivity} \tag{23}$$

$$h_{water;evap;1} = \frac{Nusselt_{w;1} \cdot k_{w;1}}{d_{h;evap}} \qquad \text{heat transfer coefficient water} \tag{24}$$

Longo 2015 correlation for convective boiling

$$\rho_{g;1} = \rho\left(R\$;\ \mathrm{T} = T_{ref;evap;1};\ \mathrm{x} = 1\right) \qquad \text{density of vapour phase} \tag{25}$$

$$\rho_{l;1} = \rho\left(R\$;\ \mathrm{T} = T_{ref;evap;1};\ \mathrm{x} = 0\right) \qquad \text{density of liquid phase} \tag{26}$$

$$\mu_{g;1} = \mu\left(R\$;\ \mathrm{T} = T_{ref;evap;1};\ \mathrm{x} = 1\right) \qquad \text{viscosity of vapour phase} \tag{27}$$

$$\mu_{l;1} = \mu\left(R\$;\ \mathrm{T} = T_{ref;evap;1};\ \mathrm{x} = 0\right) \qquad \text{viscosity of liquid phase} \tag{28}$$

$$G_{evap} = \frac{\dot{m}_{ref;LP}}{(A_{cross;evap} \cdot N_{ch;evap})} \qquad \text{mass flux} \tag{29}$$

$$G_{eq;1} = G_{evap} \cdot \left((1 - x_{ref;evap;1}) + x_{ref;evap;1} \cdot \left(\frac{\rho_{l;1}}{\rho_{g;1}}\right)^{0,5}\right) \qquad \text{equivalent mass flux} \tag{30}$$

$$Re_{eq;1} = \frac{G_{eq;1} \cdot d_{h;evap}}{\mu_{l;1}} \qquad \text{equivalent reynolds number} \tag{31}$$

$$Pr_1 = \mathrm{P_r}\left(R\$;\ \mathrm{T} = T_{ref;evap;1};\ \mathrm{x} = 0\right) \quad \text{prandt number liquid phase} \tag{32}$$

$$k_{ref;evap;1} = \mathrm{k}\left(R\$;\ \mathrm{T} = T_{ref;evap;1};\ \mathrm{x} = 0\right) \quad \text{conductivity refrigerant liquid phase} \tag{33}$$

$$h_{tp;1} = 0,122 \cdot \left(\frac{k_{ref;evap;1}}{d_{h;evap}}\right) \cdot Re_{eq;1}^{0,8} \cdot Pr_1^{1/3} \quad \text{two phase heat transfer coefficient} \tag{34}$$

Boiling number and X$_{tt}$

$$x_m = 1 - x_{ref;evap;1} \tag{35}$$

$$\rho_m = \frac{1}{\left(\left(\frac{x_m}{\rho_{g;1}}\right) + \left(\frac{1-x_m}{\rho_{l;1}}\right)\right)} \quad \text{mean density} \tag{36}$$

$$\Delta T_{evap;1} = T_{w;evap;1} - T_{ref;evap;1} \quad \text{temperature difference} \tag{37}$$

$$q_{flux;1} = h_{water;evap;1} \cdot \Delta T_{evap;1} \quad \text{heat flux} \tag{38}$$

$$\gamma_{evap;1} = \left(R\$;\ \mathrm{T} = T_{ref;evap;1}\right) \quad \text{latent heat of evaporation} \tag{39}$$

$$X_{tt;1} = \left(\frac{1 - x_m}{x_m}\right)^{0,9} \cdot \left(\left(\frac{\rho_{g;1}}{\rho_{l;1}}\right)^{0,5}\right) \cdot \left(\frac{\mu_{l;1}}{\mu_{g;1}}\right)^{0,1} \quad \text{lockhart-Martinelli parameter} \tag{40}$$

$$Bo_1 = \frac{q_{flux;1}}{\left(G_{evap} \cdot \gamma_{evap;1}\right)} \quad \text{boiling number} \tag{41}$$

$$BoX_{tt;1} = Bo_1 \cdot X_{tt;1} \quad \text{convective boiling check} \tag{42}$$

State equation for T$_{water}$

$$dtWdy_{evap;1} = -\frac{\left(2 \cdot N_{ch;evap} \cdot \left(T_{w;evap;1} - T_{ref;evap;1}\right)\right)}{\dot{m}_{water;evap} \cdot c_{water;evap;1} \cdot \left(\frac{1}{h_{tp;1} \cdot W_{evap}} + \frac{th_{m;evap}}{k_{m;evap;1} \cdot W_{evap}} + \frac{1}{h_{water;evap;1} \cdot W_{evap}}\right)} \tag{43}$$

$$N_{evap} = 300\ \boxed{\cdot} \quad \text{Number of integration steps, should be minimum} \tag{44}$$

$$\Delta y_{evap} = stepsize/N_{evap} \quad \text{step size} \tag{45}$$

Pressure drop

Amalfi et al 2015 f$_{tp}$

$$\beta_{max} = 70\ [\text{degrees}] \cdot \left|0,017453293\ \frac{\text{rad}}{\text{degrees}}\right| \quad \text{max chevron angle} \tag{46}$$

$$g = 9,81\ \left[\mathrm{m/s^2}\right] \quad \text{grav constant} \tag{47}$$

$$\sigma_{evap;1} = \gamma\left(R\$;\ \mathrm{T} = T_{ref;evap;1}\right) \quad \text{surface tension} \tag{48}$$

$$Bd_{evap;1} = \frac{\left(\rho_{l;1} - \rho_{g;1}\right) \cdot g \cdot \left(d_{h;evap}^2\right)}{\sigma_{evap;1}} \quad \text{Bond number} \tag{49}$$

$$C_{evap} = 2,125 \cdot \left(\left(\beta_{evap}/\beta_{max}\right)^{9,993}\right) + 0,955 \quad \text{Coefficient} \tag{50}$$

$$f_{tp;1} = C_{evap} \cdot 15,698 \cdot \left(\left(\left(\left(\frac{G_{evap}^2 \cdot d_{h;evap}}{\rho_m \cdot \sigma_{evap;1}}\right)^{-0,475}\right) \cdot \left(Bd_{evap;1}^{0,255}\right) \cdot \left(\left(\frac{\rho_{l;1}}{\rho_{g;1}}\right)^{-0,571}\right)\right)\right) \quad \text{two phase friction factor} \tag{51}$$

$$\Delta P_{evap;1} = \frac{2 \cdot f_{tp;1} \cdot \left(G_{evap}^2\right) \cdot \Delta y_{evap}}{d_{h;evap} \cdot \rho_m} \quad \text{pressure drop} \tag{52}$$

————————-WHILE-LOOP————————

$$i = 2 \quad \text{step variable} \tag{53}$$

$$\text{repeat} \tag{54}$$

$$y_{evap;i} = y_{evap;i-1} + \Delta y_{evap} \quad \text{Heat exchanger location} \tag{55}$$

Water side

$$T_{w;evap;i} = T_{w;evap;i-1} + dtWdy_{evap;i-1} \cdot \Delta y_{evap} \quad \text{Water temperature} \tag{56}$$

$$i_{w;evap;i} = \mathrm{h}\left(W\$;\ \mathrm{T} = T_{w;evap;i};\ \mathrm{P} = P_{water;evap}\right) \quad \text{Water specific enthalpy} \tag{57}$$

$$\dot{q}_{evap;i} = \dot{m}_{water;evap} \cdot \frac{i_{w;evap;i-1} - i_{w;evap;i}}{N_{ch;evap}} \quad \text{Heat transfered} \tag{58}$$

Refrigerant side

$$P_{ref;evap;local;i} = P_{ref;evap;local;i-1} - \Delta P_{evap;i-1} \quad \text{Refrigerant local pressuret} \tag{59}$$

$$i_{ref;evap;i} = i_{ref;evap;i-1} + \frac{\dot{q}_{evap;i}}{\left(\dot{m}_{ref;LP}/N_{ch;evap}\right)} \quad \text{Refrigerant specific enthalpy} \tag{60}$$

$$x_{ref;evap;i} = \mathrm{x}\left(R\$;\ \mathrm{h} = i_{ref;evap;i};\ \mathrm{P} = P_{ref;evap;local;i}\right) \quad \text{Refrigerant quality} \tag{61}$$

IF ELSE FOR SUPERHEATED VAPOUR temperature change

$$\text{If } x_{ref;evap;i} = 100 \text{ then} \tag{62}$$

$$T_{ref;evap;i} = \mathrm{T}\left(R\$;\ \mathrm{P} = P_{ref;evap;local;i};\ \mathrm{h} = i_{ref;evap;i}\right) \tag{63}$$

$$\text{else} \tag{64}$$

$$T_{ref;evap;i} = \mathrm{T}\left(R\$;\ \mathrm{P} = P_{ref;evap;local;i};\ \mathrm{x} = x_{ref;evap;i}\right) \quad \text{refrigerant temperature} \tag{65}$$

$$\text{endif} \tag{66}$$

————————————————START OF IF-SENTENCE————————————————

If the refrigerant quality is above 0,9, we have to use another correlation

$$\text{If}(x_{ref;evap;i-1} < 0,9) \text{ then} \quad \text{Two phase region} \tag{67}$$

Martin correlation for one-phase flow water

$$G_{w;evap} = \frac{\dot{m}_{water;evap}}{A_{cross;evap} \cdot N_{ch;evap}} \quad \text{mass flux water} \tag{68}$$

$$\mu_{l;water;i} = \mu\left(W\$;\ \mathrm{T} = T_{w;evap;i};\ \mathrm{x} = 0\right) \quad \text{viscosity of liquid phase} \tag{69}$$

$$\mu_{wall;water;evap;i} = \mu\left(W\$;\ \mathrm{T} = \frac{T_{w;evap;i} + T_{ref;evap;i}}{2};\ \mathrm{x} = 0\right) \tag{70}$$

$$Re_{w;evap;i} = \frac{G_{w;evap} \cdot d_{h;evap}}{\mu_{l;water;i}} \qquad \text{reynolds number} \tag{71}$$

$$Pr_{w;i} = \mathrm{P_r}\left(W\$;\ \mathrm{T} = T_{w;evap;i};\ \mathrm{x} = 0\right) \qquad \text{prandtl number} \tag{72}$$

$$f0_{martin;i} = \left(1,8 \cdot \log 10(Re_{w;evap;i}) - 1,5\right)^{-2} \qquad \text{parameter for friction factor calculation} \tag{73}$$

$$f1_{martin;i} = \frac{39}{\left(Re_{w;evap;i}^{0,289}\right)} \qquad \text{parameter for friction factor calculation} \tag{74}$$

$$f_{martin;i} = \left(\frac{1}{\left(\frac{\mathrm{Cos}(\beta_{evap})}{\sqrt{0,18 \cdot \tan(\beta_{evap}) + 0,36 \cdot \sin(\beta_{evap}) + \left(\frac{f0_{martin;i}}{\mathrm{Cos}(\beta_{evap})}\right)}} + \left(\frac{1 - \mathrm{Cos}(\beta_{evap})}{\sqrt{(3,8 \cdot f1_{martin;i})}}\right)\right)}\right)^2 \qquad \text{friction factor} \tag{75}$$

$$Nusselt_{w;i} = 0,122 \cdot \left(Pr_{w;i}^{1/3}\right) \cdot \left(\left(\frac{\mu_{l;water;i}}{\mu_{wall;water;evap;i}}\right)^{1/6}\right) \cdot \left(f_{martin;i} \cdot \left(\left(Re_{w;evap;i}^2\right) \cdot \sin(2 \cdot \beta_{evap})\right)\right)^{0,374} \qquad \text{Nusselt numb}$$

$$k_{w;i} = \mathrm{k}\left(W\$;\ \mathrm{T} = T_{w;evap;i};\ \mathrm{x} = 0\right) \qquad \text{water conductivity} \tag{77}$$

$$h_{water;evap;i} = \frac{Nusselt_{w;i} \cdot k_{w;i}}{d_{h;evap}} \qquad \text{heat transfer coefficient water} \tag{78}$$

Longo et al 2015 two phase correlation

$$\rho_{g;i} = \rho\left(R\$;\ \mathrm{T} = T_{ref;evap;i};\ \mathrm{x} = 1\right) \qquad \text{density of vapour phase} \tag{79}$$

$$\rho_{l;i} = \rho\left(R\$;\ \mathrm{T} = T_{ref;evap;i};\ \mathrm{x} = 0\right) \qquad \text{density of liquid phase} \tag{80}$$

$$\mu_{g;i} = \mu\left(R\$;\ \mathrm{T} = T_{ref;evap;i};\ \mathrm{x} = 1\right) \qquad \text{viscosity of vapour phase} \tag{81}$$

$$\mu_{l;i} = \mu\left(R\$;\ \mathrm{T} = T_{ref;evap;i};\ \mathrm{x} = 0\right) \qquad \text{viscosity of liquid phase} \tag{82}$$

$$Pr_i = \mathrm{P_r}\left(R\$;\ \mathrm{T} = T_{ref;evap;i};\ \mathrm{x} = 0\right) \qquad \text{prandt number liquid phase} \tag{83}$$

$$k_{ref;evap;i} = \mathrm{k}\left(R\$;\ \mathrm{T} = T_{ref;evap;i};\ \mathrm{x} = 0\right) \qquad \text{conductivity refrigerant liquid phase} \tag{84}$$

$$G_{eq;i} = G_{evap} \cdot \left((1 - x_{ref;evap;i}) + x_{ref;evap;i} \cdot \left(\frac{\rho_{l;i}}{\rho_{g;i}}\right)^{0,5}\right) \qquad \text{equivalent mass flux} \tag{85}$$

$$Re_{eq;i} = \frac{G_{eq;i} \cdot d_{h;evap}}{\mu_{l;i}} \qquad \text{equivalent reynolds number} \tag{86}$$

$$\gamma_{evap;i} = \left(R\$;\ \mathrm{T} = T_{ref;evap;i}\right) \qquad \text{latent heat of evaporation} \tag{87}$$

$$X_{tt;i} = \left(\frac{1 - x_m}{x_m}\right)^{0,9} \cdot \left(\left(\frac{\rho_{g;i}}{\rho_{l;i}}\right)^{0,5}\right) \cdot \left(\frac{\mu_{l;i}}{\mu_{g;i}}\right)^{0,1} \qquad \text{Lockhart-Martinelli parameter} \tag{88}$$

$$h_{tp;i} = 0,122 \cdot \left(\frac{k_{ref;evap;i}}{d_{h;evap}}\right) \cdot Re_{eq;i}^{0,8} \cdot Pr_i^{1/3} \qquad \text{two phase heat transfer coefficient} \tag{89}$$

$h_{dummy} = \max\left(h_{tp;i}\right)$ dummy variable which saves the heat transfer coefficient. To be used later for the linear equation(90)

Boiling number and X$_{tt}$

$\Delta T_{evap;i} = T_{w;evap;i} - T_{ref;evap;i}$ Temperature difference (91)

$q_{flux;i} = h_{water;evap;i} \cdot \Delta T_{evap;i}$ Heat flux (92)

$Bo_i = \dfrac{q_{flux;i}}{\left(G_{evap} \cdot \gamma_{evap;i}\right)}$ Boiling number (93)

$BoX_{tt;i} = Bo_i \cdot X_{tt;1}$ (94)

$k_{m;evap;i} = k\left(\text{‘Stainless\_AISI316’}\ ;\ \dfrac{T_{ref;evap;i} + T_{w;evap;i}}{2}\right)$ metal conductivity at local average temperature(95)

$c_{water;evap;i} = \mathrm{c_p}\left(W\$;\ \mathrm{P} = P_{water;evap};\ \mathrm{x} = 0\right)$ specific heat capacity of water (96)

State equation for T$_{water}$, to calculate temperature change

$$dtWdy_{evap;i} = -\frac{\left(2 \cdot N_{ch;evap} \cdot \left(T_{w;evap;i} - T_{ref;evap;i}\right)\right)}{\left(\dot{m}_{water;evap} \cdot c_{water;evap;i} \cdot \left(\frac{1}{h_{tp;i} \cdot W_{evap}} + \frac{th_{m;evap}}{k_{m;evap;i} \cdot W_{evap}} + \frac{1}{h_{water;evap;i} \cdot W_{evap}}\right)\right)} \tag{97}$$

Pressure drop

$\sigma_{evap;i} = \gamma\left(R\$;\ \mathrm{T} = T_{ref;evap;i}\right)$ surface tension (98)

$Bd_{evap;i} = \dfrac{\left(\rho_{l;i} - \rho_{g;i}\right) \cdot g \cdot \left(d^2_{h;evap}\right)}{\sigma_{evap;i}}$ Bond number (99)

$f_{tp;i} = C_{evap} \cdot 15,698 \cdot \left(\left(\left(\left(\dfrac{G^2_{evap} \cdot d_{h;evap}}{\rho_m \cdot \sigma_{evap;i}}\right)^{-0,475}\right) \cdot \left(Bd^{0,255}_{evap;i}\right) \cdot \left(\left(\dfrac{\rho_{l;i}}{\rho_{g;i}}\right)^{-0,571}\right)\right)\right)$ two phase friction factor(100)

$\Delta P_{evap;i} = \dfrac{2 \cdot f_{tp;i-1} \cdot \left(G^2_{evap}\right) \cdot \Delta y_{evap}}{d_{h;evap} \cdot \rho_m}$ pressure drop (101)

$\Delta P_{evap;tot;1} = 0$ initializing array (102)

$\Delta P_{evap;tot;i} = \Delta P_{evap;tot;i-1} + \Delta P_{evap;i}$ pressure drop (103)

$i = i + 1$ counter (104)

else If quality is higher than 0,9 (105)

———-HEAT TRANSFER COEFFICIENTS———

$y_{evap;i} = y_{evap;i-1} + \Delta y_{evap}$ Heat exchanger location (106)

Water side

$T_{w;evap;i} = T_{w;evap;i-1} + dtWdy_{evap;i-1} \cdot \Delta y_{evap}$ Water temperature (107)

$i_{w;evap;i} = \mathrm{h}\left(W\$;\ \mathrm{T} = T_{w;evap;i};\ \mathrm{P} = P_{water;evap}\right)$ Water specific enthalpy (108)

$\dot{q}_{evap;i} = \dot{m}_{water;evap} \cdot \dfrac{i_{w;evap;i-1} - i_{w;evap;i}}{N_{ch;evap}}$ Heat transfered (109)

Refrigerant side

$$P_{ref;evap;local;i} = P_{ref;evap;local;i-1} - \Delta P_{evap;i-1} \qquad \text{pressure of refrigerant} \tag{110}$$

$$i_{ref;evap;i} = i_{ref;evap;i-1} + \frac{\dot{q}_{evap;i}}{\dot{m}_{ref;LP}/N_{ch;evap}} \qquad \text{Refrigerant specific enthalpy} \tag{111}$$

$$x_{ref;evap;i} = \mathrm{x}\left(R\$;\ \mathrm{h} = i_{ref;evap;i};\ \mathrm{P} = P_{ref;evap;local;i}\right) \qquad \text{Refrigerant quality} \tag{112}$$

$$\text{If}(x_{ref;evap;i} = 100)\ \ \mathsf{then} \qquad \text{if else sentence to give the correct temperature while transitioning} \tag{113}$$

$$T_{ref;evap;i} = T_{ref;evap;i-1} \tag{114}$$

$$\mathsf{else} \tag{115}$$

$$T_{ref;evap;i} = \mathrm{T}\left(R\$;\ \mathrm{P} = P_{ref;evap;local;i};\ \mathrm{x} = x_{ref;evap;i}\right) \qquad \text{refrigerant temperature} \tag{116}$$

$$\mathsf{endif} \tag{117}$$

Martin correlation for one-phase flow water

$$G_{w;evap} = \frac{\dot{m}_{water;evap}}{A_{cross;evap} \cdot N_{ch;evap}} \qquad \text{mass flux water} \tag{118}$$

$$\mu_{l;water;i} = \mu\left(W\$;\ \mathrm{T} = T_{w;evap;i};\ \mathrm{x} = 0\right) \qquad \text{viscosity of liquid phase} \tag{119}$$

$$\mu_{wall;water;evap;i} = \mu\left(W\$;\ \mathrm{T} = \frac{T_{w;evap;i} + T_{ref;evap;i}}{2};\ \mathrm{x} = 0\right) \tag{120}$$

$$Re_{w;evap;i} = \frac{G_{w;evap} \cdot d_{h;evap}}{\mu_{l;water;i}} \qquad \text{reynolds number} \tag{121}$$

$$Pr_{w;i} = \mathrm{P_r}\left(W\$;\ \mathrm{T} = T_{w;evap;i};\ \mathrm{x} = 0\right) \qquad \text{prandtl number} \tag{122}$$

$$f0_{martin;i} = \left(1,8 \cdot \log 10(Re_{w;evap;i}) - 1,5\right)^{-2} \qquad \text{parameter for friction factor calculation} \tag{123}$$

$$f1_{martin;i} = \frac{39}{\left(Re_{w;evap;i}^{0,289}\right)} \qquad \text{parameter for friction factor calculation} \tag{124}$$

$$f_{martin;i} = \left(\frac{1}{\left(\frac{\mathrm{Cos}(\beta_{evap})}{\sqrt{0{,}18 \cdot \tan(\beta_{evap}) + 0{,}36 \cdot \sin(\beta_{evap}) + \left(\frac{f0_{martin;i}}{\mathrm{Cos}(\beta_{evap})}\right)}} + \left(\frac{1 - \mathrm{Cos}(\beta_{evap})}{\sqrt{(3{,}8 \cdot f1_{martin;i})}}\right)\right)}\right)^2 \qquad \text{friction factor} \tag{125}$$

$$Nusselt_{w;i} = 0,122 \cdot \left(Pr_{w;i}^{1/3}\right) \cdot \left(\left(\left(\frac{\mu_{l;water;i}}{\mu_{wall;water;evap;i}}\right)^{1/6}\right)\right) \cdot \left(f_{martin;i} \cdot \left(\left(Re_{w;evap;i}^2\right) \cdot \sin(2 \cdot \beta_{evap})\right)\right)^{0,374} \qquad \text{Nusselt nu}$$

$$k_{w;i} = \mathrm{k}\left(W\$;\ \mathrm{T} = T_{w;evap;i};\ \mathrm{x} = 0\right) \qquad \text{water conductivity} \tag{127}$$

$$h_{water;evap;i} = \frac{Nusselt_{w;i} \cdot k_{w;i}}{d_{h;evap}} \qquad \text{heat transfer coefficient water} \tag{128}$$

Two-phase heat transfer coefficient

Linear correlation for wall dry out

$$y_1 = h_{dummy} \tag{129}$$

$$y_2 = 600 \tag{130}$$

$$x_1 = 0,9 \tag{131}$$

$$x_2 = 1,0 \tag{132}$$

$$a = \frac{y_2 - y_1}{x_2 - x_1} \tag{133}$$

$$b = y_2 - a \cdot x_2 \tag{134}$$

$$\text{If}(x_{ref;evap;i} < 100) \ \textsf{then} \qquad \text{if sentence to correctly transition between correlations} \tag{135}$$

$$\quad h_{tp;i} = a \cdot x_{ref;evap;i} + b \tag{136}$$

$$\textsf{else} \tag{137}$$

$$\quad h_{tp;i} = h_{tp;i-1} \tag{138}$$

$$\textsf{endif} \tag{139}$$

Boiling number check

$$\rho_{g;i} = \rho\left(R\$;\ \mathrm{T} = T_{ref;evap;i};\ \mathrm{x} = 1\right) \qquad \text{density of vapour phase} \tag{140}$$

$$\rho_{l;i} = \rho\left(R\$;\ \mathrm{T} = T_{ref;evap;i};\ \mathrm{x} = 0\right) \qquad \text{density of liquid phase} \tag{141}$$

$$\mu_{g;i} = \mu\left(R\$;\ \mathrm{T} = T_{ref;evap;i};\ \mathrm{x} = 1\right) \qquad \text{viscosity of vapour phase} \tag{142}$$

$$\mu_{l;i} = \mu\left(R\$;\ \mathrm{T} = T_{ref;evap;i};\ \mathrm{x} = 0\right) \qquad \text{viscosity of liquid phase} \tag{143}$$

$$Pr_i = \mathrm{P_r}\left(R\$;\ \mathrm{T} = T_{ref;evap;i};\ \mathrm{x} = 0\right) \qquad \text{prandt number liquid phase} \tag{144}$$

$$k_{ref;evap;i} = \mathrm{k}\left(R\$;\ \mathrm{T} = T_{ref;evap;i};\ \mathrm{x} = 0\right) \qquad \text{conductivity refrigerant liquid phase} \tag{145}$$

$$\text{If}(x_{ref;evap;i} < 100) \ \textsf{then} \qquad \text{if else to prevent error given by x less than zero} \tag{146}$$

$$\quad G_{eq;i} = G_{evap} \cdot \left((1 - x_{ref;evap;i}) + x_{ref;evap;i} \cdot \left(\frac{\rho_{l;i}}{\rho_{g;i}} \right)^{0,5} \right) \qquad \text{equivalent mass flux} \tag{147}$$

$$\textsf{else} \tag{148}$$

$$\quad G_{eq;i} = G_{eq;i-1} \tag{149}$$

$$\textsf{endif} \tag{150}$$

$$Re_{eq;i} = \frac{G_{eq;i} \cdot d_{h;evap}}{\mu_{l;i}} \qquad \text{equivalent reynolds number} \tag{151}$$

$$\gamma_{evap;i} = \left(R\$;\ \mathrm{T} = T_{ref;evap;i}\right) \qquad \text{latent heat of evaporation} \tag{152}$$

$$X_{tt;i} = \left(\frac{1 - x_m}{x_m} \right)^{0,9} \cdot \left(\left(\frac{\rho_{g;i}}{\rho_{l;i}} \right)^{0,5} \right) \cdot \left(\frac{\mu_{l;i}}{\mu_{g;i}} \right)^{0,1} \qquad \text{Lockhart-Martinelli parameter} \tag{153}$$

Boiling number and X$_{tt}$

$$\Delta T_{evap;i} = T_{w;evap;i} - T_{ref;evap;i} \quad \text{Temperature difference} \tag{154}$$

$$q_{flux;i} = h_{water;evap;i} \cdot \Delta T_{evap;i} \quad \text{Heat flux} \tag{155}$$

$$Bo_i = \frac{q_{flux;i}}{(G_{evap} \cdot \gamma_{evap;i})} \quad \text{Boiling number} \tag{156}$$

$$BoX_{tt;i} = Bo_i \cdot X_{tt;1} \quad \text{convective boiling check} \tag{157}$$

Misc

$$k_{m;evap;i} = k \left(\text{‘Stainless\_AISI316’} \, ; \, \frac{T_{ref;evap;i} + T_{w;evap;i}}{2} \right) \quad \text{metal conductivity at local average temperature} \tag{158}$$

$$c_{water;evap;i} = \mathrm{c_p} \left(W\$; \ \mathrm{P} = P_{water;evap}; \ \mathrm{x} = 0 \right) \quad \text{specific heat capacity of water} \tag{159}$$

State equation for T$_{water}$, to calculate temperature change

$$dtWdy_{evap;i} = - \frac{(2 \cdot N_{ch;evap} \cdot (T_{w;evap;i} - T_{ref;evap}))}{\left(\dot{m}_{water;evap} \cdot c_{water;evap;i} \cdot \left(\frac{1}{h_{tp;i} \cdot W_{evap}} + \frac{th_{m;evap}}{k_{m;evap;i} \cdot W_{evap}} + \frac{1}{h_{water;evap;i} \cdot W_{evap}} \right) \right)} \tag{160}$$

Pressure drop

$$\sigma_{evap;i} = \gamma \left(R\$; \ \mathrm{T} = T_{ref;evap;i} \right) \quad \text{surface tension} \tag{161}$$

$$Bd_{evap;i} = \frac{(\rho_{l;i} - \rho_{g;i}) \cdot g \cdot \left(d_{h;evap}^2 \right)}{\sigma_{evap;i}} \quad \text{Bond number} \tag{162}$$

$$f_{tp;i} = C_{evap} \cdot 15,698 \cdot \left(\left(\left(\left(\frac{G_{evap}^2 \cdot d_{h;evap}}{\rho_m \cdot \sigma_{evap;i}} \right)^{-0,475} \right) \cdot \left(Bd_{evap;i}^{0,255} \right) \cdot \left(\left(\frac{\rho_{l;i}}{\rho_{g;i}} \right)^{-0,571} \right) \right) \right) \quad \text{two phase friction factor} \tag{163}$$

$$\Delta P_{evap;i} = \frac{2 \cdot f_{tp;i-1} \cdot \left(G_{evap}^2 \right) \cdot \Delta y_{evap}}{(d_{h;evap} \cdot \rho_m)} \quad \text{pressure drop} \tag{164}$$

$$\Delta P_{evap;tot;i} = \Delta P_{evap;tot;i-1} + \Delta P_{evap;i} \quad \text{cumulative pressure drop} \tag{165}$$

$$i = i + 1 \quad \text{counter} \tag{166}$$

endif (167)

until $(x_{ref;evap;i-1} = 100)$ End condition, saturated vapour (168)

——————————————While loop for superheated part——————————————

repeat (169)

$$y_{evap;i} = y_{evap;i-1} + \Delta y_{evap} \quad \text{Heat exchanger location} \tag{170}$$

$$L_{out;evap} = y_{evap;i} \quad \text{Final length of heat exchanger} \tag{171}$$

Water side

$$T_{w;evap;i} = T_{w;evap;i-1} + dtWdy_{evap;i-1} \cdot \Delta y_{evap} \quad \text{Water temperature} \tag{172}$$

$$i_{w;evap;i} = \mathrm{h}\left(W\$;\ \mathrm{T} = T_{w;evap;i};\ \mathrm{P} = P_{water;evap}\right) \qquad \text{Water specific enthalpy} \tag{173}$$

$$i_{w;out;evap} = i_{w;evap;i-1} \qquad \text{water outlet enthalpy} \tag{174}$$

$$\dot{q}_{evap;i} = \dot{m}_{water;evap} \cdot \frac{i_{w;evap;i-1} - i_{w;evap;i}}{N_{ch;evap}} \qquad \text{Heat transfered} \tag{175}$$

Refrigerant side

$$P_{ref;evap;local;i} = P_{ref;evap;local;i-1} - \Delta P_{evap;i-1} \qquad \text{pressure of refrigerant} \tag{176}$$

$$i_{ref;evap;i} = i_{ref;evap;i-1} + \frac{\dot{q}_{evap;i}}{\dot{m}_{ref;LP}/N_{ch;evap}} \qquad \text{Refrigerant specific enthalpy} \tag{177}$$

$$T_{ref;evap;i} = \mathrm{T}\left(R\$;\ \mathrm{P} = P_{ref;evap;local;i};\ \mathrm{h} = i_{ref;evap;i}\right) \qquad \text{Temperature refrigerant} \tag{178}$$

$$T_{ref;out;evap} = T_{ref;evap;i} \qquad \text{Refrigerant outlet temperature} \tag{179}$$

$$i_{ref;out;evap} = i_{ref;evap;i} \qquad \text{specific enthalpy of refrigerant out of the evaporator} \tag{180}$$

Martin correlation for one-phase flow water

$$G_{w;evap} = \frac{\dot{m}_{water;evap}}{A_{cross;evap} \cdot N_{ch;evap}} \qquad \text{mass flux water} \tag{181}$$

$$\mu_{l;water;i} = \mu\left(W\$;\ \mathrm{T} = T_{w;evap;i};\ \mathrm{x} = 0\right) \qquad \text{viscosity of liquid phase} \tag{182}$$

$$\mu_{wall;water;evap;i} = \mu\left(W\$;\ \mathrm{T} = \frac{T_{w;evap;i} + T_{ref;evap;i}}{2};\ \mathrm{x} = 0\right) \qquad \text{viscosity at wall} \tag{183}$$

$$Re_{w;evap;i} = \frac{G_{w;evap} \cdot d_{h;evap}}{\mu_{l;water;i}} \qquad \text{reynolds number} \tag{184}$$

$$Pr_{w;i} = \mathrm{P_r}\left(W\$;\ \mathrm{T} = T_{w;evap;i};\ \mathrm{x} = 0\right) \qquad \text{prandtl number} \tag{185}$$

$$f0_{martin;i} = \left(1,8 \cdot \log 10(Re_{w;evap;i}) - 1,5\right)^{-2} \qquad \text{parameter for friction factor calculation} \tag{186}$$

$$f1_{martin;i} = \frac{39}{\left(Re_{w;evap;i}^{0,289}\right)} \qquad \text{parameter for friction factor calculation} \tag{187}$$

$$f_{martin;i} = \left(\frac{1}{\left(\frac{\mathrm{Cos}(\beta_{evap})}{\sqrt{0,18 \cdot \tan\left(\beta_{evap}\right) + 0,36 \cdot \sin\left(\beta_{evap}\right) + \left(\frac{f0_{martin;i}}{\mathrm{Cos}(\beta_{evap})}\right)}} + \left(\frac{1 - \mathrm{Cos}(\beta_{evap})}{\sqrt{(3,8 \cdot f1_{martin;i})}}\right)\right)}\right)^2 \qquad \text{friction factor} \tag{188}$$

$$Nusselt_{w;i} = 0,122 \cdot \left(Pr_{w;i}^{1/3}\right) \cdot \left(\left(\left(\frac{\mu_{l;water;i}}{\mu_{wall;water;evap;i}}\right)^{1/6}\right)\right) \cdot \left(f_{martin;i} \cdot \left(\left(Re_{w;evap;i}^2\right) \cdot \sin\left(2 \cdot \beta_{evap}\right)\right)\right)^{0,374} \qquad \text{Nusselt numb}$$

$$k_{w;i} = \mathrm{k}\left(W\$;\ \mathrm{T} = T_{w;evap;i};\ \mathrm{x} = 0\right) \qquad \text{water conductivity} \tag{190}$$

$$h_{water;evap;i} = \frac{Nusselt_{w;i} \cdot k_{w;i}}{d_{h;evap}} \qquad \text{heat transfer coefficient water} \tag{191}$$

Martin correlation for one-phase flow refrigerant

$$\rho_{g;i} = \rho\left(R\$;\ \mathrm{T} = T_{ref;evap;i};\ \mathrm{x} = 1\right) \quad \text{density of vapour phase} \qquad (192)$$

$$\rho_{l;i} = \rho\left(R\$;\ \mathrm{T} = T_{ref;evap;i};\ \mathrm{x} = 0\right) \quad \text{density of liquid phase} \qquad (193)$$

$$\mu_{g;i} = \mu\left(R\$;\ \mathrm{T} = T_{ref;evap;i};\ \mathrm{x} = 1\right) \quad \text{viscosity of vapour phase} \qquad (194)$$

$$\mu_{l;i} = \mu\left(R\$;\ \mathrm{T} = T_{ref;evap;i};\ \mathrm{x} = 0\right) \quad \text{viscosity of liquid phase} \qquad (195)$$

$$Pr_i = \mathrm{P_r}\left(R\$;\ \mathrm{T} = T_{ref;evap;i};\ \mathrm{x} = 0\right) \quad \text{prandt number liquid phase} \qquad (196)$$

$$G_{ref;evap} = \frac{\dot{m}_{ref;LP}}{A_{cross;evap} \cdot N_{ch;evap}} \quad \text{mass flux refrigerant} \qquad (197)$$

$$\mu_{wall;ref;evap;i} = \mu\left(R\$;\ \mathrm{T} = \frac{T_{ref;evap;i} + T_{w;evap;i}}{2};\ \mathrm{x} = 0\right) \qquad (198)$$

$$Re_{ref;evap;i} = \frac{G_{ref;evap} \cdot d_{h;evap}}{\mu_{g;i}} \quad \text{reynolds number} \qquad (199)$$

$$Pr_{ref;evap;i} = \mathrm{P_r}\left(R\$;\ \mathrm{T} = T_{ref;evap;i};\ \mathrm{x} = 1\right) \quad \text{prandtl number} \qquad (200)$$

$$f0_{martin;ref;evap;i} = \left(1,8 \cdot \log 10(Re_{ref;evap;i}) - 1,5\right)^{-2} \quad \text{parameter for friction factor calculation} \qquad (201)$$

$$f1_{martin;ref;evap;i} = \frac{39}{\left(Re_{ref;evap;i}^{0,289}\right)} \quad \text{parameter for friction factor calculation} \qquad (202)$$

$$f_{martin;ref;evap;i} = \left(\frac{1}{\left(\frac{\mathrm{Cos}(\beta_{evap})}{\sqrt{0,18 \cdot \tan(\beta_{evap}) + 0,36 \cdot \sin(\beta_{evap}) + \left(\frac{f0_{martin;ref;evap;i}}{\mathrm{Cos}(\beta_{evap})}\right)}} + \left(\frac{1 - \mathrm{Cos}(\beta_{evap})}{\sqrt{(3,8 \cdot f1_{martin;ref;evap;i})}}\right)\right)}\right)^2 \quad \text{friction fact}$$

$$Nusselt_{ref;evap;i} = 0,122 \cdot \left(Pr_i^{1/3}\right) \cdot \left(\left(\frac{\mu_{g;i}}{\mu_{wall;ref;evap;i}}\right)^{1/6}\right) \cdot \left(f_{martin;ref;evap;i} \cdot \left(\left(Re_{ref;evap;i}^2\right) \cdot \sin(2 \cdot \beta_{evap})\right)\right)^{0,374} \quad \text{Nu}$$

$$k_{ref;evap;i} = \mathrm{k}\left(R\$;\ \mathrm{T} = T_{ref;evap;i};\ \mathrm{x} = 1\right) \quad \text{refrigerant conductivity} \qquad (205)$$

$$h_{ref;i} = \frac{Nusselt_{ref;evap;i} \cdot k_{ref;evap;i}}{d_{h;evap}} \quad \text{heat transfer coefficient} \qquad (206)$$

Misc

$$k_{m;evap;i} = k\left(\text{‘Stainless\_AISI316’};\ \frac{T_{ref;evap;i} + T_{w;evap;i}}{2}\right) \quad \text{metal conductivity at local average temperature}(207)$$

$$c_{water;evap;i} = \mathrm{c_p}\left(W\$;\ \mathrm{P} = P_{water;evap};\ \mathrm{x} = 0\right) \quad \text{specific heat capacity of water} \qquad (208)$$

State equation for T_{water}, to calculate temperature change

$$dtWdy_{evap;i} = -\frac{\left(2 \cdot N_{ch;evap} \cdot \left(T_{w;evap;i} - T_{ref;evap}\right)\right)}{\left(\dot{m}_{water;evap} \cdot c_{water;evap;i} \cdot \left(\frac{1}{h_{ref;i} \cdot W_{evap}} + \frac{th_{m;evap}}{k_{m;evap;i} \cdot W_{evap}} + \frac{1}{h_{water;evap;i} \cdot W_{evap}}\right)\right)} \qquad (209)$$

Misc pressure drops except the frictional pressure drop

$\Delta P_{evap;i} = \dfrac{2 \cdot f_{martin;ref;evap;i} \cdot \left(G^2_{ref;evap}\right) \cdot \Delta y_{evap}}{\left(d_{h;evap} \cdot \rho_{g;i}\right)}$ pressure drop (210)

$\Delta P_{evap;tot;i} = \Delta P_{evap;tot;i-1} + \Delta P_{evap;i}$ cumulative pressure drop (211)

$g = 9,81\ \left[\mathrm{m/s^2}\right]$ gravitational constant (212)

$\Delta Pgravity_{evap} = \rho_m \cdot g \cdot L_{out;evap}$ gravity driven acceleration (213)

$\Delta x_{ref;evap} = abs\left(1 - x_{ref;evap;1}\right)$ quality change (214)

$\Delta Pacc_{evap} = \left(G^2_{evap}\right) \cdot \Delta x_{ref;evap} \cdot \left(\left(\dfrac{1}{\rho_{g;1}}\right) - \left(\dfrac{1}{\rho_{l;1}}\right)\right)$ acceleration pressure (215)

$\Delta Pmanifold_{evap} = 1,5 \cdot \left(\rho_m \cdot \left(\left(G_{evap}/\rho_m\right)^2\right) \cdot 0,5\right)$ inlet/outlet pressure loss (216)

$\Delta P_{evap;all} = \Delta P_{evap;tot;i} + \Delta Pgravity_{evap} + \Delta Pacc_{evap} + \Delta Pmanifold_{evap}$ total pressure loss (217)

$P_{evap;out} = P_{ref;evap} - \Delta P_{evap;all}$ outlet pressure (218)

$i = i + 1$ (219)

$i_{ref;evap;needed} = \mathrm{h}\left(R\$;\ \mathrm{T} = T_{ref;evap} + T_{superheat};\ \mathrm{P} = P_{ref;evap}\right)$ needed rerigerant enthalpy to reach the correct superheat

until $\left(i_{ref;evap;i-1} > i_{ref;evap;needed}\right)$ End condition (221)

end (222)

——————————————————-END OF EVAPORATOR PROCEDURE——————————————————-

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——————————————————-CONDENSER PROCEDURE——————————————————-

procedure $COND(T_{water;in;cond};\ P_{water;cond};\ P_{ref;cond};\ T_{ref;cond};\ T_{water;out;cond};\ W_{cond\$};\ R\$;\ \dot m_{water;cond})$;

$th_{water;cond};\ W_{cond};\ \dot m_{ref;HP};\ th_{ref;cond};\ th_{m;cond};\ stepsize;\ i_{out;comp;HP};\ N_{ch;cond};\ d_{h;cond};\ A_{cross;cond};$

$\beta_{cond};\ p_{co;cond};\ T_{out;comp;HP};\ T_{subcool};\ T_{loss;HP;cond}$ (223)

$i_{ref;out;cond};\ x_{ref;out;cond};\ T_{w;real;out};\ L_{out;cond};\ i_{w;out;cond};\ P_{cond;out};\ T_{ref;cond;out})$ (224)

`$ARRAYS ON`

——————————————————-DESUPERHEATER PART——————————————————-

Initializing variables, starting points

$y_{cond;1} = 0$ Starting point (225)

Water side

$T_{w;cond;1} = T_{water;in;cond}$ Inlet water temperature (226)

$i_{w;cond;1} = \mathrm{h}\left(W_{cond\$};\ \mathrm{T} = T_{w;cond;1};\ \mathrm{P} = P_{water;cond}\right)$ Inlet water enthalpy (227)

$$i_{w;out;wanted;cond} = \mathrm{h}\left(W_{cond\$};\ \mathrm{T} = T_{water;out;cond};\ \mathrm{P} = P_{water;cond}\right) \quad \text{wanted outlet enthalpy} \tag{228}$$

Refrigerant side

$$i_{ref;cond;1} = i_{out;comp;HP} \quad \text{Inlet refrigerant enthalpy} \tag{229}$$

$$T_{ref;cond;1} = T_{out;comp;HP} + T_{loss;HP;cond} \quad \text{Inlet refrr temperature} \tag{230}$$

$$P_{ref;cond;local;1} = P_{ref;cond} \quad \text{local refrigerant pressure} \tag{231}$$

subcooling of refrigerant

$$x_{ref;cond;1} = \mathrm{x}\left(R\$;\ \mathrm{h} = i_{ref;cond;1};\ \mathrm{P} = P_{ref;cond;local;1}\right) \quad \text{refrigerant quality} \tag{232}$$

Other

$$\dot{q}_{cond;1} = 0 \quad \text{Starting point for heat transfer} \tag{233}$$

$$k_{m;cond;1} = k\left(\text{'Stainless\_AISI316'};\ \frac{T_{ref;cond} + T_{w;cond;1}}{2}\right) \quad \text{Metal conductivity at local average temperature at starting point(234}$$

$$c_{water;cond;1} = \mathrm{c_p}\left(W_{cond\$};\ \mathrm{P} = P_{water;cond};\ \mathrm{x} = 0\right) \quad \text{Water specific heat capacity at starting point} \tag{235}$$

$$N_{cond} = 300\ ▯ \quad \text{Number of integration steps} \tag{236}$$

$$\Delta y_{cond} = stepsize/N_{cond} \quad \text{step size} \tag{237}$$

Variables related to pressure drop

$$\rho_{g;cond;1} = \rho\left(R\$;\ \mathrm{T} = T_{ref;cond;1};\ \mathrm{x} = 1\right) \quad \text{density of vapour phase} \tag{238}$$

$$\rho_{l;cond;1} = \rho\left(R\$;\ \mathrm{T} = T_{ref;cond;1};\ \mathrm{x} = 0\right) \quad \text{density of liquid phase} \tag{239}$$

$$\rho_{m;cond} = \frac{\rho_{g;cond;1} + \rho_{l;cond;1}}{2} \quad \text{mean density} \tag{240}$$

———-HEAT TRANSFER COEFFICIENTS———

Martin correlation for one-phase flow

$$G_{w;cond} = \frac{\dot{m}_{water;cond}}{\left(A_{cross;cond} \cdot N_{ch;cond}\right)} \quad \text{mass flux water} \tag{241}$$

$$\mu_{l;water;cond;1} = \mu\left(W_{cond\$};\ \mathrm{T} = T_{w;cond;1};\ \mathrm{x} = 0\right) \quad \text{viscosity of liquid phase} \tag{242}$$

$$\mu_{wall;water;cond;1} = \mu\left(W_{cond\$};\ \mathrm{T} = \frac{T_{w;cond;1} + T_{ref;cond;1}}{2};\ \mathrm{x} = 0\right) \quad \text{viscosity at wall} \tag{243}$$

$$Re_{w;cond;1} = \frac{G_{w;cond} \cdot d_{h;cond}}{\mu_{l;water;cond;1}} \quad \text{reynolds number} \tag{244}$$

$$Pr_{w;cond;1} = \mathrm{P_r}\left(W_{cond\$};\ \mathrm{T} = T_{w;cond;1};\ \mathrm{x} = 0\right) \quad \text{prandtl number} \tag{245}$$

$$f0_{martin;cond;1} = \left(1,8 \cdot \log 10(Re_{w;cond;1}) - 1,5\right)^{-2} \quad \text{parameter for friction factor calculation} \tag{246}$$

$$f1_{martin;cond;1} = \frac{39}{\left(Re_{w;cond;1}^{0,289}\right)} \quad \text{parameter for friction factor calculation} \tag{247}$$

$$f_{martin;cond;1} = \left(\frac{1}{\left(\frac{\text{Cos}(\beta_{cond})}{\sqrt{0{,}18 \cdot \tan(\beta_{cond}) + 0{,}36 \cdot \sin(\beta_{cond}) + \left(\frac{f0_{martin;cond;1}}{\text{Cos}(\beta_{cond})} \right)}} + \left(\frac{1 - \text{Cos}(\beta_{cond})}{\sqrt{(3{,}8 \cdot f1_{martin;cond;1})}} \right) \right)} \right)^2 \quad \text{friction factor(248)}$$

$$Nusselt_{w;cond;1} = 0,122 \cdot \left(Pr_{w;cond;1}^{1/3} \right) \cdot \left(\left(\frac{\mu_{l;water;cond;1}}{\mu_{wall;water;cond;1}} \right)^{1/6} \right) \cdot \left(f_{martin;cond;1} \cdot \left(\left(Re_{w;cond;1}^2 \right) \cdot \sin(2 \cdot \beta_{cond}) \right) \right)^{0,374} \quad \text{Nusse}$$

$$k_{w;cond;1} = \text{k}\,(W_{cond\$};\ \text{T} = T_{w;cond;1};\ \text{x} = 0) \quad \text{water conductivity} \tag{250}$$

$$h_{water;cond;1} = \frac{Nusselt_{w;cond;1} \cdot k_{w;cond;1}}{d_{h;cond}} \quad \text{heat transfer coefficient water} \tag{251}$$

$$\text{If}(x_{ref;cond;1} = 100) \ \text{then} \quad \text{SUPERHEATED VAPOUR} \tag{252}$$

Martin correlation for one-phase flow refrigerant

$$G_{ref;cond} = \frac{\dot{m}_{ref;HP}}{A_{cross;cond} \cdot N_{ch;cond}} \quad \text{mass flux refrigerant} \tag{253}$$

$$\mu_{l;ref;cond;1} = \mu\,(R\$;\ \text{T} = T_{ref;cond;1};\ \text{x} = 0) \quad \text{viscosity of liquid phase} \tag{254}$$

$$\mu_{wall;ref;cond;1} = \mu \left(R\$;\ \text{T} = \frac{T_{ref;cond;1} + T_{w;cond;1}}{2};\ \text{x} = 0 \right) \quad \text{viscosity at wall} \tag{255}$$

$$Re_{ref;cond;1} = \frac{G_{ref;cond} \cdot d_{h;cond}}{\mu_{l;ref;cond;1}} \quad \text{reynolds number} \tag{256}$$

$$Pr_{ref;cond;1} = \text{P}_\text{r}\,(R\$;\ \text{T} = T_{ref;cond;1};\ \text{x} = 0) \quad \text{prandtl number} \tag{257}$$

$$f0_{martin;ref;cond;1} = (1,8 \cdot \log 10(Re_{ref;cond;1}) - 1,5)^{-2} \quad \text{parameter for friction factor calculation} \tag{258}$$

$$f1_{martin;ref;cond;1} = \frac{39}{\left(Re_{ref;cond;1}^{0,289} \right)} \quad \text{parameter for friction factor calculation} \tag{259}$$

$$f_{martin;ref;cond;1} = \left(\frac{1}{\left(\frac{\text{Cos}(\beta_{cond})}{\sqrt{0{,}18 \cdot \tan(\beta_{cond}) + 0{,}36 \cdot \sin(\beta_{cond}) + \left(\frac{f0_{martin;ref;cond;1}}{\text{Cos}(\beta_{cond})} \right)}} + \left(\frac{1 - \text{Cos}(\beta_{cond})}{\sqrt{(3{,}8 \cdot f1_{martin;ref;cond;1})}} \right) \right)} \right)^2 \quad \text{friction fac}$$

$$Nusselt_{ref;cond;1} = 0,122 \cdot \left(Pr_{ref;cond;1}^{1/3} \right) \cdot \left(\left(\frac{\mu_{l;ref;cond;1}}{\mu_{wall;ref;cond;1}} \right)^{1/6} \right) \cdot \left(f_{martin;ref;cond;1} \cdot \left(\left(Re_{w;cond;1}^2 \right) \cdot \sin(2 \cdot \beta_{cond}) \right) \right)^{0,374}$$

$$k_{ref;cond;1} = \text{k}\,(R\$;\ \text{T} = T_{ref;cond;1};\ \text{x} = 0) \quad \text{water conductivity} \tag{262}$$

$$h_{ref;cond;1} = \frac{Nusselt_{ref;cond;1} \cdot k_{ref;cond;1}}{d_{h;cond}} \quad \text{heat transfer coefficient water} \tag{263}$$

total pressure drop

else (264)

Longo 2015 correlation for condensation of saturated gas

$$G_{ref;cond} = \frac{\dot{m}_{ref;HP}}{(A_{cross;cond} \cdot N_{ch;cond})} \quad \text{mass flux} \tag{265}$$

$$\rho_{g;cond;1} = \rho\left(R\$;\ \mathrm{T} = T_{ref;cond;1};\ \mathrm{x} = 1\right) \quad \text{density of vapour phase} \tag{266}$$

$$\rho_{l;cond;1} = \rho\left(R\$;\ \mathrm{T} = T_{ref;cond;1};\ \mathrm{x} = 0\right) \quad \text{density of liquid phase} \tag{267}$$

$$G_{eq;cond;1} = G_{ref;cond} \cdot \left((1 - x_{ref;cond;1}) + x_{ref;cond;1} \cdot \left(\frac{\rho_{l;cond;1}}{\rho_{g;cond;1}} \right)^{0,5} \right) \quad \text{equivalent mass flux} \tag{268}$$

$$\mu_{l;cond;1} = \mu\left(R\$;\ \mathrm{T} = T_{ref;cond;1};\ \mathrm{x} = 0\right) \quad \text{viscosity of liquid phase} \tag{269}$$

$$Re_{eq;cond;1} = \frac{G_{eq;cond;1} \cdot d_{h;cond}}{\mu_{l;cond;1}} \quad \text{equivalent reynolds number} \tag{270}$$

$$Pr_{l;cond;1} = \mathrm{P_r}\left(R\$;\ \mathrm{T} = T_{ref;cond;1};\ \mathrm{x} = 0\right) \quad \text{prandtl number liquid phase} \tag{271}$$

$$k_{ref;cond;1} = \mathrm{k}\left(R\$;\ \mathrm{T} = T_{ref;cond;1};\ \mathrm{x} = 0\right) \quad \text{conductivity refrigerant} \tag{272}$$

$$h_{ref;cond;1} = 1,875 \cdot \left(\frac{k_{ref;cond;1}}{d_{h;cond}} \right) \cdot \left(Re_{eq;cond;1}^{0,445} \right) \cdot \left(Pr_{l;cond;1}^{1/3} \right) \quad \text{two phase heat trans coeff} \tag{273}$$

endif (274)

State equation for T$_{water}$

$$dtWdy_{cond;1} = \frac{(2 \cdot N_{ch;cond} \cdot (T_{ref;cond;1} - T_{w;cond;1}))}{\dot{m}_{water;cond} \cdot c_{water;cond;1} \cdot \left(\frac{1}{h_{ref;cond;1} \cdot W_{cond}} + \frac{th_{m;cond}}{k_{m;cond;1} \cdot W_{cond}} + \frac{1}{h_{water;cond;1} \cdot W_{cond}} \right)} \tag{275}$$

$$i = 2 \quad \text{step counter} \tag{276}$$

repeat (277)

$$y_{cond;i} = y_{cond;i-1} + \Delta y_{cond} \quad \text{Location in heat exchanger} \tag{278}$$

$$L_{out;cond} = y_{cond;i} \quad \text{plate length} \tag{279}$$

Water side

$$T_{w;cond;i} = T_{w;cond;i-1} + dtWdy_{cond;i-1} \cdot \Delta y_{cond} \quad \text{Water temperature} \tag{280}$$

$$i_{w;cond;i} = \mathrm{h}\left(W_{cond\$};\ \mathrm{T} = T_{w;cond;i};\ \mathrm{P} = P_{water;cond}\right) \quad \text{Water specific enthalpy} \tag{281}$$

$$\dot{q}_{cond;i} = \dot{m}_{water;cond} \cdot \frac{i_{w;cond;i-1} - i_{w;cond;i}}{N_{ch;cond}} \quad \text{Heat transfered} \tag{282}$$

$$T_{w;real;out} = T_{w;cond;i-1} \quad \text{Water temperature at outlet} \tag{283}$$

$$i_{w;out;cond} = i_{w;cond;i-1} \quad \text{water enthalpy at outlet} \tag{284}$$

Refrigerant side

$$P_{ref;cond;local;i} = P_{ref;cond} \quad \text{local pressure} \tag{285}$$

$$i_{ref;cond;i} = i_{ref;cond;i-1} + \frac{\dot{q}_{cond;i}}{\dot{m}_{ref;HP}/N_{ch;cond}} \quad \text{Refrigerant specific enthalpy} \tag{286}$$

$$x_{ref;cond;i} = \mathrm{x}\left(R\$;\ \mathrm{h} = i_{ref;cond;i};\ \mathrm{P} = P_{ref;cond;local;i}\right) \quad \text{Refrigerant quality} \tag{287}$$

————————————————————————————START OF IF SENTENCE——————————————————————

$$\mathrm{If}(x_{ref;cond;i} = 100)\ \text{then} \quad \text{If superheated vapour} \tag{288}$$

$$T_{ref;cond;i} = \mathrm{T}\left(R\$;\ \mathrm{P} = P_{ref;cond;local;i};\ \mathrm{h} = i_{ref;cond;i}\right) \quad \text{Refrigerant temperature} \tag{289}$$

$$T_{ref;cond;out} = T_{ref;cond;i-1} \quad \text{outlet temperature of refrigerant} \tag{290}$$

$$i_{ref;out;cond} = i_{ref;cond;i-1} \quad \text{refrigerant enthalpy at condenser outlet} \tag{291}$$

$$x_{ref;out;cond} = x_{ref;cond;i-1} \quad \text{refrigerant quality at condenser outlet} \tag{292}$$

$$c_{water;cond;i} = \mathrm{c_p}\left(W_{cond\$};\ \mathrm{T} = T_{ref;cond;i};\ \mathrm{P} = P_{ref;cond;local;i}\right) \quad \text{Specific heat capacity water} \tag{293}$$

$$c_{water;cond1;i} = \mathrm{c_p}\left(W_{cond\$};\ \mathrm{T} = T_{ref;cond;i};\ \mathrm{x} = 0\right) \quad \text{Specific heat liquid water} \tag{294}$$

$$k_{m;cond;i} = k\left(\text{‘Stainless\_AISI316’};\ \frac{T_{ref;cond} + T_{w;cond;i}}{2}\right) \quad \text{Metal conductivity at local average temperature} \tag{295}$$

———-HEAT TRANSFER COEFFICIENTS———

Martin correlation water side

$$G_{w;cond} = \frac{\dot{m}_{water;cond}}{(A_{cross;cond} \cdot N_{ch;cond})} \quad \text{mass flux water} \tag{296}$$

$$\mu_{l;water;cond;i} = \mu\left(W_{cond\$};\ \mathrm{T} = T_{w;cond;i};\ \mathrm{x} = 0\right) \quad \text{viscosity of liquid phase} \tag{297}$$

$$\mu_{wall;water;cond;i} = \mu\left(W_{cond\$};\ \mathrm{T} = \frac{T_{w;cond;i} + T_{ref;cond;i}}{2};\ \mathrm{x} = 0\right) \quad \text{viscosity at wall} \tag{298}$$

$$Re_{w;cond;i} = \frac{G_{w;cond} \cdot d_{h;cond}}{\mu_{l;water;cond;i}} \quad \text{reynolds number} \tag{299}$$

$$Pr_{w;cond;i} = \mathrm{P_r}\left(W_{cond\$};\ \mathrm{T} = T_{w;cond;i};\ \mathrm{x} = 0\right) \quad \text{prandtl number} \tag{300}$$

$$f0_{martin;cond;i} = \left(1,8 \cdot \log 10(Re_{w;cond;i}) - 1,5\right)^{-2} \quad \text{parameter for friction factor calculation} \tag{301}$$

$$f1_{martin;cond;i} = \frac{39}{\left(Re_{w;cond;i}^{0,289}\right)} \quad \text{parameter for friction factor calculation} \tag{302}$$

$$f_{martin;cond;i} = \left(\frac{1}{\left(\frac{\mathrm{Cos}(\beta_{cond})}{\sqrt{0{,}18 \cdot \tan(\beta_{cond}) + 0{,}36 \cdot \sin(\beta_{cond}) + \left(\frac{f0_{martin;cond;i}}{\mathrm{Cos}(\beta_{cond})}\right)}} + \left(\frac{1 - \mathrm{Cos}(\beta_{cond})}{\sqrt{(3{,}8 \cdot f1_{martin;cond;i})}}\right)\right)}\right)^{2} \quad \text{friction factor(303}$$

$$Nusselt_{w;cond;i} = 0,122 \cdot \left(Pr_{w;cond;i}^{1/3}\right) \cdot \left(\left(\frac{\mu_{l;water;cond;i}}{\mu_{wall;water;cond;i}}\right)^{1/6}\right) \cdot \left(f_{martin;cond;i} \cdot \left(\left(Re_{w;cond;i}^{2}\right) \cdot \sin\left(2 \cdot \beta_{cond}\right)\right)\right)^{0,374}$$

$$k_{w;cond;i} = \mathrm{k}\left(W_{cond\$};\ \mathrm{T} = T_{w;cond;i};\ \mathrm{x} = 0\right) \quad \text{water conductivity} \tag{305}$$

$$h_{water;cond;i} = \frac{Nusselt_{w;cond;i} \cdot k_{w;cond;i}}{d_{h;cond}} \quad \text{heat transfer coefficient water} \tag{306}$$

Martin correlation refrigerant side

$$G_{ref;cond} = \frac{\dot{m}_{ref;HP}}{A_{cross;cond} \cdot N_{ch;cond}} \quad \text{mass flux refrigerant} \tag{307}$$

$$\mu_{l;ref;cond;i} = \mu\left(R\$;\ \mathrm{T} = T_{ref;cond;i};\ \mathrm{x} = 0\right) \quad \text{viscosity of liquid phase} \tag{308}$$

$$\mu_{wall;ref;cond;i} = \mu\left(R\$;\ \mathrm{T} = \left(\frac{T_{ref;cond;i} + T_{w;cond;i}}{2}\right);\ \mathrm{x} = 0\right) \quad \text{viscosity at wall} \tag{309}$$

$$Re_{ref;cond;i} = \frac{G_{ref;cond} \cdot d_{h;cond}}{\mu_{l;ref;cond;i}} \quad \text{reynolds number} \tag{310}$$

$$Pr_{ref;cond;i} = \mathrm{P_r}\left(R\$;\ \mathrm{T} = T_{ref;cond;i};\ \mathrm{x} = 0\right) \quad \text{prandtl number} \tag{311}$$

$$f0_{martin;ref;cond;i} = \left(1,8 \cdot \log 10(Re_{ref;cond;i}) - 1,5\right)^{-2} \quad \text{parameter for friction factor calculation} \tag{312}$$

$$f1_{martin;ref;cond;i} = \frac{39}{\left(Re_{ref;cond;i}^{0,289}\right)} \quad \text{parameter for friction factor calculation} \tag{313}$$

$$f_{martin;ref;cond;i} = \left(\frac{1}{\left(\frac{\mathrm{Cos}(\beta_{cond})}{\sqrt{0,18 \cdot \tan(\beta_{cond}) + 0,36 \cdot \sin(\beta_{cond}) + \left(\frac{f0_{martin;ref;cond;i}}{\mathrm{Cos}(\beta_{cond})}\right)}} + \left(\frac{1 - \mathrm{Cos}(\beta_{cond})}{\sqrt{(3,8 \cdot f1_{martin;ref;cond;i})}}\right)\right)}\right)^{2} \quad \text{friction}$$

$$Nusselt_{ref;cond;i} = 0,122 \cdot \left(Pr_{ref;cond;i}^{1/3}\right) \cdot \left(\left(\frac{\mu_{l;ref;cond;i}}{\mu_{wall;ref;cond;i}}\right)^{1/6}\right) \cdot \left(f_{martin;ref;cond;i} \cdot \left(\left(Re_{w;cond;i}^{2}\right) \cdot \sin(2 \cdot \beta_{cond})\right)\right)^{0,37}$$

$$k_{ref;cond;i} = \mathrm{k}\left(R\$;\ \mathrm{T} = T_{ref;cond;i};\ \mathrm{x} = 0\right) \quad \text{water conductivity} \tag{316}$$

$$h_{ref;cond;i} = \frac{Nusselt_{ref;cond;i} \cdot k_{ref;cond;i}}{d_{h;cond}} \quad \text{heat transfer coefficient water} \tag{317}$$

Pressure drop

$$dtWdy_{cond;i} = \frac{\left(2 \cdot N_{ch;cond} \cdot (T_{ref;cond} - T_{w;cond;i})\right)}{\dot{m}_{water;cond} \cdot c_{water;cond;i} \cdot \left(\frac{1}{h_{ref;cond;i} \cdot W_{cond}} + \frac{th_{m;cond}}{k_{m;cond;i} \cdot W_{cond}} + \frac{1}{h_{water;cond;i} \cdot W_{cond}}\right)} \tag{318}$$

State equation for T_{water}

$$i = i + 1 \tag{319}$$

else CONDENSING SECTION (320)

$$T_{ref;cond;i} = \mathrm{T}\left(R\$;\ \mathrm{P} = P_{ref;cond;local;i};\ \mathrm{h} = i_{ref;cond;i}\right) \quad \text{refrigerant temperature} \tag{321}$$

$$T_{ref;cond;out} = T_{ref;cond;i-1} \qquad \text{outlet temperature of rerigerant} \tag{322}$$

$$i_{ref;out;cond} = i_{ref;cond;i-1} \qquad \text{refrigerant enthalpy at condenser outlet} \tag{323}$$

$$x_{ref;out;cond} = x_{ref;cond;i-1} \qquad \text{refrigerant quality at condenser outlet} \tag{324}$$

$$c_{water;cond;i} = \mathrm{c_p}\left(W_{cond\$};\ \mathrm{P} = P_{water;cond};\ \mathrm{T} = T_{w;cond;i}\right) \qquad \text{Specific heat capacity water} \tag{325}$$

$$k_{m;cond;i} = k\left(\text{‘Stainless\_AISI316’ };\ \frac{T_{ref;cond;i} + T_{w;cond;i}}{2}\right) \qquad \text{Metal conductivity at local average temperature} \tag{326}$$

———-HEAT TRANSFER COEFFICIENTS———

Martin correlation one phase flow

$$G_{w;cond} = \frac{\dot{m}_{water;cond}}{\left(A_{cross;cond} \cdot N_{ch;cond}\right)} \qquad \text{mass flux water} \tag{327}$$

$$\mu_{l;water;cond;i} = \mu\left(W_{cond\$};\ \mathrm{T} = T_{w;cond;i};\ \mathrm{x} = 0\right) \qquad \text{viscosity of liquid phase} \tag{328}$$

$$\mu_{wall;water;cond;i} = \mu\left(W_{cond\$};\ \mathrm{T} = \left(\frac{T_{w;cond;i} + T_{ref;cond;i}}{2}\right);\ \mathrm{x} = 0\right) \qquad \text{viscosity gas phase} \tag{329}$$

$$Re_{w;cond;i} = \frac{G_{w;cond} \cdot d_{h;cond}}{\mu_{l;water;cond;i}} \qquad \text{reynolds number} \tag{330}$$

$$Pr_{w;cond;i} = \mathrm{P_r}\left(W_{cond\$};\ \mathrm{T} = T_{w;cond;i};\ \mathrm{x} = 0\right) \qquad \text{prandtl number} \tag{331}$$

$$f0_{martin;cond;i} = \left(1,8 \cdot \log 10(Re_{w;cond;i}) - 1,5\right)^{-2} \qquad \text{parameter for friction factor calculation} \tag{332}$$

$$f1_{martin;cond;i} = \frac{39}{\left(Re_{w;cond;i}^{0,289}\right)} \qquad \text{parameter for friction factor calculation} \tag{333}$$

$$f_{martin;cond;i} = \left(\frac{1}{\left(\frac{\mathrm{Cos}(\beta_{cond})}{\sqrt{0,18 \cdot \tan(\beta_{cond}) + 0,36 \cdot \sin(\beta_{cond}) + \left(\frac{f0_{martin;cond;i}}{\mathrm{Cos}(\beta_{cond})}\right)}} + \left(\frac{1 - \mathrm{Cos}(\beta_{cond})}{\sqrt{(3,8 \cdot f1_{martin;cond;i})}}\right)\right)}\right)^2 \qquad \text{friction factor(334}$$

$$Nusselt_{w;cond;i} = 0,122 \cdot \left(Pr_{w;cond;i}^{1/3}\right) \cdot \left(\left(\frac{\mu_{l;water;cond;i}}{\mu_{wall;water;cond;i}}\right)^{1/6}\right) \cdot \left(f_{martin;cond;i} \cdot \left(\left(Re_{w;cond;i}^{2}\right) \cdot \sin(2 \cdot \beta_{cond})\right)\right)^{0,374}$$

$$k_{w;cond;i} = \mathrm{k}\left(W_{cond\$};\ \mathrm{T} = T_{w;cond;i};\ \mathrm{x} = 0\right) \qquad \text{water conductivity} \tag{336}$$

$$h_{water;cond;i} = \frac{Nusselt_{w;cond;i} \cdot k_{w;cond;i}}{d_{h;cond}} \qquad \text{heat transfer coefficient water} \tag{337}$$

Longo 2015 correlation for condensation of saturated gas

$$G_{ref;cond} = \frac{\dot{m}_{ref;HP}}{\left(A_{cross;cond} \cdot N_{ch;cond}\right)} \qquad \text{mass flux} \tag{338}$$

$$\rho_{g;cond;i} = \rho\left(R\$;\ \mathrm{T} = T_{ref;cond;i};\ \mathrm{x} = 1\right) \qquad \text{density of vapour phase} \tag{339}$$

$$\rho_{l;cond;i} = \rho\left(R\$;\ \mathrm{T} = T_{ref;cond;i};\ \mathrm{x} = 0\right) \qquad \text{density of liquid phase} \tag{340}$$

$$\mathrm{If}(x_{ref;cond;i} > -100)\ \text{then} \qquad \text{IF sentence to avoid problems with quality below zero} \tag{341}$$

$$G_{eq;cond;i} = G_{ref;cond} \cdot \left((1 - x_{ref;cond;i}) + x_{ref;cond;i} \cdot \left(\frac{\rho_{l;cond;i}}{\rho_{g;cond;i}} \right)^{0,5} \right) \qquad \text{equivalent mass flux} \tag{342}$$

$$\text{else} \tag{343}$$

$$G_{eq;cond;i} = G_{eq;cond;i-1} \tag{344}$$

$$\text{endif} \tag{345}$$

$$\mu_{l;cond;i} = \mu\left(R\$;\ \mathrm{T} = T_{ref;cond;i};\ \mathrm{x} = 0\right) \qquad \text{viscosity of liquid phase} \tag{346}$$

$$Re_{eq;cond;i} = \frac{G_{eq;cond;i} \cdot d_{h;cond}}{\mu_{l;cond;i}} \qquad \text{equivalent reynolds number} \tag{347}$$

$$Pr_{l;cond;i} = \mathrm{P_r}\left(R\$;\ \mathrm{T} = T_{ref;cond;i};\ \mathrm{x} = 0\right) \qquad \text{prandtl number liquid phase} \tag{348}$$

$$k_{ref;cond;i} = \mathrm{k}\left(R\$;\ \mathrm{T} = T_{ref;cond;i};\ \mathrm{x} = 0\right) \qquad \text{conductivity refrigerant} \tag{349}$$

$$\mathrm{If}(x_{ref;cond;i} = -100)\ \text{then} \tag{350}$$

$$h_{ref;cond;i} = h_{ref;cond;i-1} \tag{351}$$

$$\text{else} \tag{352}$$

$$h_{ref;cond;i} = 1,875 \cdot \left(\frac{k_{ref;cond;i}}{d_{h;cond}} \right) \cdot \left(Re_{eq;cond;i}^{0,445} \right) \cdot \left(Pr_{l;cond;i}^{1/3} \right) \qquad \text{two phase heat trans coeff} \tag{353}$$

$$\text{endif} \tag{354}$$

State equation for T_{water}

$$dtWdy_{cond;i} = \frac{(2 \cdot N_{ch;cond} \cdot (T_{ref;cond;i} - T_{w;cond;i}))}{\dot{m}_{water;cond} \cdot c_{water;cond;i} \cdot \left(\frac{1}{h_{ref;cond;i} \cdot W_{cond}} + \frac{th_{m;cond}}{k_{m;cond;i} \cdot W_{cond}} + \frac{1}{h_{water;cond;i} \cdot W_{cond}} \right)} \tag{355}$$

Pressure drop

Misc pressure drops except the frictional pressure drop

$$g = 9,81\ \left[\mathrm{m/s^2}\right] \qquad \text{gravitational constant} \tag{356}$$

$$\Delta Pgravity_{cond} = \rho_{m;cond} \cdot g \cdot L_{out;cond} \qquad \text{gravity driven acceleration} \tag{357}$$

$$\Delta Pacc_{cond} = \left(G_{ref;cond}^2\right) \cdot 1 \cdot \left(\left(\frac{1}{\rho_{g;cond;i}} \right) - \left(\frac{1}{\rho_{l;cond;i}} \right) \right) \qquad \text{acceleration pressure, quality change is 1} \tag{358}$$

$$\Delta Pmanifold_{cond} = 1,5 \cdot \left(\rho_{m;cond} \cdot \left((G_{ref;cond}/\rho_{m;cond})^2 \right) \cdot 0,5 \right) \qquad \text{inlet/outlet pressure loss} \tag{359}$$

$$\Delta P_{cond;fric} = 1000 \cdot \frac{1,9 \cdot G_{ref;cond}^2}{2 \cdot \rho_{m;cond}} \qquad \text{kinetic model from Longo hydrocarbon paper} \tag{360}$$

$$\Delta P_{cond;all} = \Delta P_{cond;fric} + \Delta Pmanifold_{cond} - \Delta Pacc_{cond} + -\Delta Pgravity_{cond} \tag{361}$$

$$P_{cond;out} = P_{ref;cond} - \Delta P_{cond;all} \qquad \text{Pressure at condenser outlet} \tag{362}$$

$$i = i + 1 \tag{363}$$

endif (364)

until $(x_{ref;cond;i-1} = -100)$ End condition (365)

———————————————WHILE LOOP FOR SUBCOOLING OF RERIGERANT——————————————

repeat (366)

$$y_{cond;i} = y_{cond;i-1} + \Delta y_{cond} \quad \text{Location in heat exchanger} \tag{367}$$

$$L_{out;cond} = y_{cond;i} \quad \text{plate length} \tag{368}$$

Water side

$$T_{w;cond;i} = T_{w;cond;i-1} + dtWdy_{cond;i-1} \cdot \Delta y_{cond} \quad \text{Water temperature} \tag{369}$$

$$i_{w;cond;i} = \mathrm{h}\left(W_{cond\$};\ \mathrm{T} = T_{w;cond;i};\ \mathrm{P} = P_{water;cond}\right) \quad \text{Water specific enthalpy} \tag{370}$$

$$\dot{q}_{cond;i} = \dot{m}_{water;cond} \cdot \frac{i_{w;cond;i-1} - i_{w;cond;i}}{N_{ch;cond}} \quad \text{Heat transfered} \tag{371}$$

$$T_{w;real;out} = T_{w;cond;i-1} \quad \text{Water temperature at outlet} \tag{372}$$

$$i_{w;out;cond} = i_{w;cond;i-1} \quad \text{water enthalpy at outlet} \tag{373}$$

Refrigerant side

$$P_{ref;cond;local;i} = P_{ref;cond} \quad \text{local pressure} \tag{374}$$

$$i_{ref;cond;i} = i_{ref;cond;i-1} + \frac{\dot{q}_{cond;i}}{\dot{m}_{ref;HP}/N_{ch;cond}} \quad \text{Refrigerant specific enthalpy} \tag{375}$$

$$T_{ref;cond;i} = \mathrm{T}\left(R\$;\ \mathrm{P} = P_{ref;cond;local;i};\ \mathrm{h} = i_{ref;cond;i}\right) \tag{376}$$

$$T_{ref;cond;out} = T_{ref;cond;i-1} \quad \text{Temperature of refrigerant} \tag{377}$$

$$i_{ref;out;cond} = i_{ref;cond;i-1} \quad \text{refrigerant enthalpy at condenser outlet} \tag{378}$$

$$x_{ref;out;cond} = -100 \quad \text{refrigerant quality at condenser outlet} \tag{379}$$

$$c_{water;cond;i} = \mathrm{c_p}\left(W_{cond\$};\ \mathrm{T} = T_{ref;cond;i};\ \mathrm{P} = P_{ref;cond;local;i}\right) \quad \text{Specific heat capacity water} \tag{380}$$

$$c_{water;cond1;i} = \mathrm{c_p}\left(W_{cond\$};\ \mathrm{T} = T_{ref;cond;i};\ \mathrm{x} = 0\right) \quad \text{Specific heat liquid water} \tag{381}$$

$$k_{m;cond;i} = k\left(\text{'Stainless\_AISI316'}\ ;\ \frac{T_{ref;cond} + T_{w;cond;i}}{2}\right) \quad \text{Metal conductivity at local average temperature} \tag{382}$$

———-HEAT TRANSFER COEFFICIENTS———

Martin correlation water side

$$G_{w;cond} = \frac{\dot{m}_{water;cond}}{\left(A_{cross;cond} \cdot N_{ch;cond}\right)} \quad \text{mass flux water} \tag{383}$$

$$\mu_{l;water;cond;i} = \mu\left(W_{cond\$};\ \mathrm{T} = T_{w;cond;i};\ \mathrm{x} = 0\right) \quad \text{viscosity of liquid phase} \tag{384}$$

$$\mu_{wall;water;cond;i} = \mu\left(W_{cond\$};\ \mathrm{T} = \frac{T_{w;cond;i} + T_{ref;cond;i}}{2};\ \mathrm{x} = 0\right) \quad \text{viscosity at wall} \tag{385}$$

$$Re_{w;cond;i} = \frac{G_{w;cond} \cdot d_{h;cond}}{\mu_{l;water;cond;i}} \quad \text{reynolds number} \tag{386}$$

$$Pr_{w;cond;i} = \mathrm{P_r}\left(W_{cond\$};\ \mathrm{T} = T_{w;cond;i};\ \mathrm{x} = 0\right) \quad \text{prandtl number} \tag{387}$$

$$f0_{martin;cond;i} = \left(1,8 \cdot \log 10(Re_{w;cond;i}) - 1,5\right)^{-2} \quad \text{parameter for friction factor calculation} \tag{388}$$

$$f1_{martin;cond;i} = \frac{39}{\left(Re_{w;cond;i}^{0,289}\right)} \quad \text{parameter for friction factor calculation} \tag{389}$$

$$f_{martin;cond;i} = \left(\frac{1}{\left(\frac{\mathrm{Cos}(\beta_{cond})}{\sqrt{0,18\cdot\tan(\beta_{cond})+0,36\cdot\sin(\beta_{cond})+\left(\frac{f0_{martin;cond;i}}{\mathrm{Cos}(\beta_{cond})}\right)}} + \left(\frac{1-\mathrm{Cos}(\beta_{cond})}{\sqrt{(3,8\cdot f1_{martin;cond;i})}}\right)\right)}\right)^2 \quad \text{friction factor} \tag{390}$$

$$Nusselt_{w;cond;i} = 0,122\cdot\left(Pr_{w;cond;i}^{1/3}\right)\cdot\left(\left(\frac{\mu_{l;water;cond;i}}{\mu_{wall;water;cond;i}}\right)^{1/6}\right)\cdot\left(f_{martin;cond;i}\cdot\left(\left(Re_{w;cond;i}^{2}\right)\cdot\sin(2\cdot\beta_{cond})\right)\right)^{0,374} \quad \text{Nu}$$

$$k_{w;cond;i} = \mathrm{k}\left(W_{cond\$};\ \mathrm{T} = T_{w;cond;i};\ \mathrm{x} = 0\right) \quad \text{water conductivity} \tag{392}$$

$$h_{water;cond;i} = \frac{Nusselt_{w;cond;i} \cdot k_{w;cond;i}}{d_{h;cond}} \quad \text{heat transfer coefficient water} \tag{393}$$

Martin correlation refrigerant side

$$G_{ref;cond} = \frac{\dot{m}_{ref;HP}}{A_{cross;cond} \cdot N_{ch;cond}} \quad \text{mass flux refrigerant} \tag{394}$$

$$\mu_{l;ref;cond;i} = \mu\left(R\$;\ \mathrm{T} = T_{ref;cond;i};\ \mathrm{x} = 0\right) \quad \text{viscosity of liquid phase} \tag{395}$$

$$\mu_{wall;ref;cond;i} = \mu\left(R\$;\ \mathrm{T} = \left(\frac{T_{ref;cond;i} + T_{w;cond;i}}{2}\right);\ \mathrm{x} = 0\right) \quad \text{viscosity at wall} \tag{396}$$

$$Re_{ref;cond;i} = \frac{G_{ref;cond} \cdot d_{h;cond}}{\mu_{l;ref;cond;i}} \quad \text{reynolds number} \tag{397}$$

$$Pr_{ref;cond;i} = \mathrm{P_r}\left(R\$;\ \mathrm{T} = T_{ref;cond;i};\ \mathrm{x} = 0\right) \quad \text{prandtl number} \tag{398}$$

$$f0_{martin;ref;cond;i} = \left(1,8 \cdot \log 10(Re_{ref;cond;i}) - 1,5\right)^{-2} \quad \text{parameter for friction factor calculation} \tag{399}$$

$$f1_{martin;ref;cond;i} = \frac{39}{\left(Re_{ref;cond;i}^{0,289}\right)} \quad \text{parameter for friction factor calculation} \tag{400}$$

$$f_{martin;ref;cond;i} = \left(\frac{1}{\left(\frac{\mathrm{Cos}(\beta_{cond})}{\sqrt{0,18\cdot\tan(\beta_{cond})+0,36\cdot\sin(\beta_{cond})+\left(\frac{f0_{martin;ref;cond;i}}{\mathrm{Cos}(\beta_{cond})}\right)}} + \left(\frac{1-\mathrm{Cos}(\beta_{cond})}{\sqrt{(3,8\cdot f1_{martin;ref;cond;i})}}\right)\right)}\right)^2 \quad \text{friction fact}$$

$$Nusselt_{ref;cond;i} = 0,122 \cdot \left(Pr_{ref;cond;i}^{1/3} \right) \cdot \left(\left(\frac{\mu_{l;ref;cond;i}}{\mu_{wall;ref;cond;i}} \right)^{1/6} \right) \cdot \left(f_{martin;ref;cond;i} \cdot \left(\left(Re_{w;cond;i}^{2} \right) \cdot \sin\left(2 \cdot \beta_{cond}\right) \right) \right)^{0,374}$$

$$k_{ref;cond;i} = \mathrm{k}\left(R\$;\ \mathrm{T} = T_{ref;cond;i};\ \mathrm{x} = 0 \right) \quad \text{water conductivity} \tag{403}$$

$$h_{ref;cond;i} = \frac{Nusselt_{ref;cond;i} \cdot k_{ref;cond;i}}{d_{h;cond}} \quad \text{heat transfer coefficient water} \tag{404}$$

State equation for T$_{water}$

$$dtWdy_{cond;i} = \frac{\left(2 \cdot N_{ch;cond} \cdot \left(T_{ref;cond} - T_{w;cond;i}\right)\right)}{\dot{m}_{water;cond} \cdot c_{water;cond;i} \cdot \left(\frac{1}{h_{ref;cond;i} \cdot W_{cond}} + \frac{th_{m;cond}}{k_{m;cond;i} \cdot W_{cond}} + \frac{1}{h_{water;cond;i} \cdot W_{cond}} \right)} \tag{405}$$

$$i = i + 1 \tag{406}$$

until $\left(T_{ref;cond;i-1} < \left(T_{ref;cond} - T_{subcool}\right)\right)$ end condition (407)

end (408)

————————————————END OF CONDENSER PROCEDURE————————————————

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————————————————END OF PROCEDURES————————————————

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————————————————-EVAPORATOR STAGE————————————————-

Inputs

$stepsize = 3\ [\mathrm{m}]$ Step size for control volume length (409)

mass flow rate refrigerant

$\dot{m}_{water;evap} = 6,74\ [\mathrm{kg/s}]$ cold-side mass flow rate, from boundary conditions (410)

GEOMETRY OF HEAT EXCHANGER

number of channel/plate pairs

$$\beta_{evap} = 45\ [\mathrm{degrees}] \cdot \left| 0,017453293\ \frac{\mathrm{rad}}{\mathrm{degrees}} \right| \quad \text{chevron angle} \tag{411}$$

$$d_{h;evap} = 5,6\ [\mathrm{mm}] \cdot \left| 0,001000000\ \frac{\mathrm{m}}{\mathrm{mm}} \right| \quad \text{hydraulic diameter} \tag{412}$$

$$b_{evap} = 2,8\ [\mathrm{mm}] \cdot \left| 0,001000000\ \frac{\mathrm{m}}{\mathrm{mm}} \right| \quad \text{gap space} \tag{413}$$

$$A_{cross;evap} = W_{evap} \cdot b_{evap} \quad \text{cross sectional area plate} \tag{414}$$

$$th_{ref;evap} = 2,2\ [\mathrm{mm}] \cdot \left| 0,001000000\ \frac{\mathrm{m}}{\mathrm{mm}} \right| \quad \text{channel width on hot side} \tag{415}$$

$$th_{water;evap} = 2,2\ [\mathrm{mm}] \cdot \left| 0,001000000\ \frac{\mathrm{m}}{\mathrm{mm}} \right| \quad \text{channel width on cold side} \tag{416}$$

$$th_{m;evap} = 0,5\ [\mathrm{mm}] \cdot \left| 0,001000000\ \frac{\mathrm{m}}{\mathrm{mm}} \right| \quad \text{thickness of plate} \tag{417}$$

$W_{evap} = 31,8\ [\mathrm{cm}] \cdot \left| 0,010000000\ \frac{\mathrm{m}}{\mathrm{cm}} \right|$ widt of heat exhanger (418)

$area_{evap} = L_{out;evap} \cdot W_{evap} \cdot N_{ch;evap}$ area of heat exchanger, not sure if correct (419)

PRESSURES

$P_{ref;evap} = \mathrm{P}\left(R\$;\ \mathrm{T} = T_{ref;evap};\ \mathrm{x} = 1\right)$ hot-side pressure (420)

$P_{water;evap} = 2\ [\mathrm{bar}] \cdot \left| 100000,000\ \frac{\mathrm{Pa}}{\mathrm{bar}} \right|$ cold-side pressure (421)

TEMPERATURES

$T_{water;in;evap} = \mathrm{ConvertTemp}(C;\ K;\ 90,6\ [\mathrm{C}])$ feed inlet temperature (422)

$T_{water;out;evap} = \mathrm{ConvertTemp}(C;\ K;\ 80\ [\mathrm{C}])$ feed outlet temperature (423)

OTHER

$W\$ = \text{'Water'}$ process fluid (424)

Calling the evaporator procedure

call $EVAP(T_{water;in;evap};\ T_{water;out;evap};\ P_{water;evap};\ P_{ref;evap};\ P_{ref;cond};\ T_{ref;cond};\ T_{ref;evap};\ W\$;\ R\$;\ \dot{m}_{water;evap})\ ;$

$N_{ch;evap};\ th_{water;evap};\ W_{evap};\ \dot{m}_{ref;LP};\ th_{ref;evap};\ th_{m;evap};\ stepsize;\ d_{h;evap};\ A_{cross;evap};\ \beta_{evap};\ i_{ref;LT};\ P_{LT};$

$i_{ref;out;int;liq}$ (425)

$L_{out;evap};\ i_{ref;out;evap};\ i_{ref;evap;1};\ i_{w;out;evap};\ P_{evap;out};\ T_{ref;out;evap};\ \Delta P_{evap;all})$ (426)

$i_{w;in;evap} = \mathrm{h}\left(W\$;\ \mathrm{T} = T_{water;in;evap};\ \mathrm{P} = P_{water;evap}\right)$ intlet enthalpy of water at evaporator (427)

$i_{w;out;wanted;evap} = \mathrm{h}\left(W\$;\ \mathrm{T} = T_{water;out;evap};\ \mathrm{P} = P_{water;evap}\right)$ enthalpy needed at evaporator outlet (428)

$\dot{q}_{ref;evap} = \dot{m}_{ref;LP} \cdot \left(i_{ref;out;evap} - i_{ref;evap;1}\right)$ heat transfered to refrigerant (429)

$\dot{q}_{w;evap} = \dot{m}_{water;evap} \cdot \left(i_{w;in;evap} - i_{w;out;evap}\right)$ heat transfered to water (430)

$\dot{q}_{needed;evap} = \dot{m}_{water;evap} \cdot \left(i_{w;in;evap} - i_{w;out;wanted;evap}\right)$ heat needed to water stream (431)

Unbalance

$UB_{evap} = \frac{abs\left(\dot{q}_{w;evap} + \dot{q}_{ref;evap}\right)}{\left(\dot{q}_{w;evap} + 0,0000000001\right)}$ discrepancy between heat streams. This number should be as close to 2 as possible(432)

$err_{heat;evap} = abs\left(\dot{q}_{needed;evap} - \dot{q}_{w;evap}\right)$ error function to find correct refrigerant mass flow (433)

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————————————————————-COMPRESSORS PART————————————————————-

Initializing variables

$\eta_{is} = 0,7$ isentropic efficiency of compressors (434)

Low stage compressor

$$\text{call } Compressor2_{CL}(i_{ref;out;evap};\ P_{ref;inlet;comp;LP};\ P_{out;comp;LP};\ \dot{m}_{ref;LP};\ R\$;\ \eta_i)\ s \tag{435}$$

$$i_{out;comp;LP};\ \dot{W}_{comp;LP};\ \Big) \quad \text{eta}_s \tag{436}$$

$$T_{out;comp;LP} = \mathrm{T}\left(R\$;\ \mathrm{h} = i_{out;comp;LP};\ \mathrm{P} = P_{out;comp;LP}\right) \quad \text{temperature out of LP compressor} \tag{437}$$

$$\rho_{suction;LP} = \rho\left(R\$;\ \mathrm{P} = P_{ref;inlet;comp;LP};\ \mathrm{h} = i_{ref;out;evap}\right) \quad \text{density compressor inlet} \tag{438}$$

$$\dot{V}_{comp;LP} = 3600 \cdot \dot{m}_{ref;LP} \cdot \left(1/\rho_{suction;LP}\right) \quad \text{volume flow compressor inlet} \tag{439}$$

High stage compressor

$$\text{call } Compressor2_{CL}(i_{ref;out;int;v};\ P_{ref;inlet;comp;HP};\ P_{out;comp;HP};\ \dot{m}_{ref;HP};\ R\$;\ \eta_i)\ s \tag{440}$$

$$i_{out;comp;HP};\ \dot{W}_{comp;HP};\ \Big) \quad \text{eta}_s \tag{441}$$

$$T_{out;comp;HP} = \mathrm{T}\left(R\$;\ \mathrm{h} = i_{out;comp;HP};\ \mathrm{P} = P_{out;comp;HP}\right) \quad \text{temperature out of HP compressor} \tag{442}$$

$$\rho_{suction;HP} = \rho\left(R\$;\ \mathrm{P} = P_{ref;inlet;comp;HP};\ \mathrm{h} = i_{ref;out;int;v}\right) \quad \text{density compressor inlet} \tag{443}$$

$$\dot{V}_{comp;HP} = 3600 \cdot \dot{m}_{ref;HP} \cdot \left(1/\rho_{suction;HP}\right) \quad \text{volume flow compressor inlet} \tag{444}$$

Error functions for the compressors

$$\Delta P_{guess;comp;HP} = 206 - 12,6\ \ [\mathrm{Pa}] \quad \text{pressure loss guess for HP comp} \tag{445}$$

$$\Delta P_{guess;comp;LP} = 531,5 - 277,3\ \ [\mathrm{Pa}] \tag{446}$$

$$P_{out;comp;LP} = P_{int} + \Delta P_{guess;comp;LP} \quad \text{outlet pressure from LP compressor} \tag{447}$$

$$P_{out;comp;HP} = P_{ref;cond} + \Delta P_{guess;comp;HP} \quad \text{outlet pressure from HP compressor} \tag{448}$$

$$err_{comp;LP} = abs\left(\Delta P_{guess;comp;LP} - \Delta P_{pipe;LP;int}\right) \quad \text{Adjust guess to match the pipe loss to the flash tank} \tag{449}$$

$$err_{comp;HP} = abs\left(\Delta P_{guess;comp;HP} - \Delta P_{pipe;HP;cond}\right) \quad \text{Adjust guess to match the pipe loss to the condenser inlet} \tag{450}$$

Should also adjust to match the pressure loss in the condenser?

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————————————————INTERMEDIATE FLASH TANK————————————————

$$P_{int} = \sqrt{P_{ref;evap} \cdot P_{ref;cond}} \quad \text{intermediate pressure} \tag{451}$$

$$T_{int} = \mathrm{T}\left(R\$;\ \mathrm{P} = P_{int};\ \mathrm{x} = 0\right) \quad \text{intermediate temperature} \tag{452}$$

$$i_{ref;out;int;liq} = \mathrm{h}\left(R\$;\ \mathrm{P} = P_{int};\ \mathrm{x} = 0\right) \quad \text{enthalpy of saturated liquid} \tag{453}$$

$$i_{ref;out;int;v} = \mathrm{h}\left(R\$;\ \mathrm{P} = P_{int};\ \mathrm{x} = 1\right) \quad \text{enthalpy of saturated vapour} \tag{454}$$

$$P_{HT;guess} = P_{ref;cond} - \Delta P_{guess;comp;HP} \tag{455}$$

$$i_{ref;HT} = \mathrm{h}\left(R\$;\ \mathrm{T} = T_{ref;cond} - T_{subcool};\ \mathrm{P} = P_{HT;guess}\right) \quad \text{inlet enthalpy of HT throttling valve} \tag{456}$$

$$\dot{m}_{ref;HP} = \dot{m}_{ref;LP} \cdot \frac{i_{out;comp;LP} - i_{ref;out;int;liq}}{\left(i_{ref;out;int;v} - i_{ref;HT}\right)} \quad \text{high stage mass flow} \tag{457}$$

$P_{LT} = P_{int} - \Delta P_{pipe;int;LT}$ pressure before low stage throttling valve (458)

$i_{ref;LT} = \text{h}\left(R\$;\ \text{P} = P_{LT};\ \text{x} = 0\right)$ inlet enthalpy of LT throttling valve (459)

Since the outlet pressure of the condenser is a function of the HP mass flow, and the mass flow is a function of the P_{LT} point. We have to guess or neglect the pressure loss in the condenser to start with, and then later optimize to find the correct $\dot{m}_{ref;HP}$

————————————————————-CONDENSER PART——————————————————-

GEOMETRY OF CONDENSER

Try to make it more of a number of plates thing instead of the evap, since we have condensation very much influenced by the size of the plates

number plate pairs

$\beta_{cond} = 45\ [\text{degrees}] \cdot \left|0,017453293\ \frac{\text{rad}}{\text{degrees}}\right|$ chevron angel (460)

$p_{co;cond} = 4\ [\text{mm}] \cdot \left|0,001000000\ \frac{\text{m}}{\text{mm}}\right|$ fin pitch (461)

$d_{h;cond} = 5,6\ [\text{mm}] \cdot \left|0,001000000\ \frac{\text{m}}{\text{mm}}\right|$ hydraulic diameter (462)

$b_{cond} = 2,8\ [\text{mm}] \cdot \left|0,001000000\ \frac{\text{m}}{\text{mm}}\right|$ gap space (463)

$A_{cross;cond} = W_{cond} \cdot b_{cond}$ cross sectional area plate (464)

$W_{cond} = 31,8\ [\text{cm}] \cdot \left|0,010000000\ \frac{\text{m}}{\text{cm}}\right|$ widt of heat exhanger (465)

$th_{ref;cond} = 2,2\ [\text{mm}] \cdot \left|0,001000000\ \frac{\text{m}}{\text{mm}}\right|$ channel width on hot side (466)

$th_{water;cond} = 2,2\ [\text{mm}] \cdot \left|0,001000000\ \frac{\text{m}}{\text{mm}}\right|$ channel width on cold side (467)

$th_{m;cond} = 0,5\ [\text{mm}] \cdot \left|0,001000000\ \frac{\text{m}}{\text{mm}}\right|$ thickness of plate (468)

$area_{cond} = L_{out;cond} \cdot W_{cond} \cdot N_{ch;cond}$ area of heat exchanger, not sure if correct (469)

PRESSURES

$P_{ref;cond} = \text{P}\left(R\$;\ \text{T} = T_{ref;cond};\ \text{x} = 1\right)$ hot-side pressure (470)

$P_{water;cond} = 2\ [\text{bar}] \cdot \left|100000,000\ \frac{\text{Pa}}{\text{bar}}\right|$ cold-side pressure (471)

$\dot{q}_{needed;cond} = \left(\dot{W}_{comp;LP} + \dot{W}_{comp;HP}\right) + \dot{q}_{w;evap}$ Heat needed to be dumped in the condenser (472)

$i_{water;in;cond} = \text{h}\left(W_{cond\$};\ \text{T} = T_{water;in;cond};\ \text{P} = P_{water;cond}\right)$ inlet enthalpy of water into condenser (473)

$\dot{q}_{w;real;cond} = \dot{m}_{water;cond} \cdot \left(i_{w;out;cond} - i_{water;in;cond}\right)$ actual heat transfered to water in the condenser (474)

$\dot{q}_{ref;cond} = \dot{m}_{ref;HP} \cdot \left(i_{out;comp;HP} - i_{ref;out;cond}\right)$ heat transfered to refrigerant (475) water mass flow rate, should be adjusted

$W_{cond\$}$ = 'Water' Water or heated fluid (476)

TEMPERATURES

$T_{ref;cond}$ = ConvertTemp($C;\ K;\ 115\ [\mathrm{C}]$) refrigerant condensing temperature (477)

$T_{water;in;cond}$ = ConvertTemp($C;\ K;\ 96\ [\mathrm{C}]$) feed inlet temperature (478)

$T_{water;out;cond}$ = ConvertTemp($C;\ K;\ 110\ [\mathrm{C}]$) feed outlet temperature (479)

Calling the condenser procedure

call $COND(T_{water;in;cond};\ P_{water;cond};\ P_{ref;cond};\ T_{ref;cond};\ T_{water;out;cond};\ W_{cond\$};\ R\$;\ \dot{m}_{water;cond};\ th_{water;cond};\ W_{cond};\ )$

$\dot{m}_{ref;HP};\ th_{ref;cond};\ th_{m;cond};\ stepsize;\ i_{out;comp;HP};\ N_{ch;cond};\ d_{h;cond};\ A_{cross;cond};\ \beta_{cond};\ p_{co;cond};\ T_{out;comp;HP};$

$T_{subcool};\ T_{loss;HP;cond}$ (480)

$i_{ref;out;cond};\ x_{ref;out;cond};\ T_{w;real;out};\ L_{out;cond};\ i_{w;out;cond};\ P_{cond;out};\ T_{ref;cond;out})$ (481)

$err_{temp;cond} = abs\left(T_{w;real;out} - T_{water;out;cond}\right)$ (482)

$err_{heat;cond} = abs\left(\dot{q}_{needed;cond} - \dot{q}_{w;real;cond}\right)$ discrepancy between heat needed vs water and actual heat dumped to water(483)

$UB_{cond} = \left(\frac{abs\left(\dot{q}_{w;real;cond} + \dot{q}_{ref;cond}\right)}{\dot{q}_{w;real;cond} + 0{,}0000000001}\right)$ discrepancy between heat streams. This number should be as close to 2 as possible(484)

————————————————-PIPE LOSSES————————————————-

$D_{tube} = 7\ [\mathrm{cm}] \cdot \left|0{,}010000000\ \frac{\mathrm{m}}{\mathrm{cm}}\right|$ Diameter of tube (485)

$k_{tube} = 0{,}001 \cdot 10^{-3}\ [\mathrm{m}]$ Absolute roughness of tube, copper (486)

$RelRough = k_{tube}/D_{tube}$ Relative roughness of tube (487)

——————————————From evaporator outlet to LP compressor inlet. EVAP-LP——————————————————————

$L_{evap;LP} = 5\ [\mathrm{m}]$ (488)

$D_{tube;evap;LP} = 7\ [\mathrm{cm}] \cdot \left|0{,}010000000\ \frac{\mathrm{m}}{\mathrm{cm}}\right|$ (489)

call $PipeFlow(R\$;\ T_{ref;out;evap} + 0{,}001;\ P_{evap;out};\ \dot{m}_{ref;LP};\ D_{tube;evap;LP};\ L_{evap;LP};\ RelRoug)\,h$ (490)

$;\ ;\ \Delta P_{pipe;evap;LP};\ ;\ ;\ )$ (491)

$P_{ref;inlet;comp;LP} = P_{evap;out} - \Delta P_{pipe;evap;LP}$ (492)

——————————————From LP compressor outlet to flash tank inlet LP-INT———————————————————————-

$L_{LP;int} = 2\ [\mathrm{m}]$ (493)

$D_{tube;LP;int} = 7\ [\mathrm{cm}] \cdot \left|0{,}010000000\ \frac{\mathrm{m}}{\mathrm{cm}}\right|$ (494)

call $PipeFlow(R\$;\ T_{out;comp;LP} + 0{,}001;\ P_{out;comp;LP};\ \dot{m}_{ref;LP};\ D_{tube;LP;int};\ L_{LP;int};\ RelRoug)\ h$ (495)

$;\ ;\ \Delta P_{pipe;LP;int};\ ;\ ;\ )$ (496)

——————————————————From flash tank outlet for HP compressor inlet INT-HP——————————————————————————————————-

$$L_{int;HP} = 2\ [\text{m}] \tag{497}$$

$$D_{tube;int;HP} = 7\ [\text{cm}] \cdot \left| 0{,}010000000\ \frac{\text{m}}{\text{cm}} \right| \tag{498}$$

call $PipeFlow(R\$;\ T_{int} + 0{,}001;\ P_{int};\ \dot{m}_{ref;HP};\ D_{tube;int;HP};\ L_{int;HP};\ RelRoug)\ h$ (499)

$;\ ;\ \Delta P_{pipe;int;HP};\ ;\ ;\ )$ (500)

$P_{ref;inlet;comp;HP} = P_{int} - \Delta P_{pipe;int;HP}$ inlet pressure high stage compressor (501)

——————————————————From HP compressor outlet to condenser HP-COND——————————————————————————

$$L_{HP;cond} = 5\ [\text{m}] \tag{502}$$

$$D_{tube;HP;cond} = 7\ [\text{cm}] \cdot \left| 0{,}010000000\ \frac{\text{m}}{\text{cm}} \right| \tag{503}$$

call $PipeFlow(R\$;\ T_{int} + 0{,}001;\ P_{out;comp;HP};\ \dot{m}_{ref;HP};\ D_{tube;HP;cond};\ L_{HP;cond};\ RelRoug)\ h$ (504)

$;\ ;\ \Delta P_{pipe;HP;cond};\ ;\ ;\ )$ (505)

——————————————————From condenser outlet to high stage throttling valve COND-HT————————————————————————————————

$$L_{cond;HT} = 8\ [\text{m}] \tag{506}$$

$$D_{tube;cond;HT} = 7\ [\text{cm}] \cdot \left| 0{,}010000000\ \frac{\text{m}}{\text{cm}} \right| \tag{507}$$

call $PipeFlow(R\$;\ T_{ref;cond;out} + 0{,}001;\ P_{cond;out};\ \dot{m}_{ref;HP};\ D_{tube;cond;HT};\ L_{cond;HT};\ RelRoug)\ h$ (508)

$;\ ;\ \Delta P_{pipe;cond;HT};\ ;\ ;\ )$ We have fluid here (509)

——————————————————From flash tank to low stage throttling valve INT-LT——————————————————————————————————————-

$$L_{int;LT} = 8\ [\text{m}] \tag{510}$$

$$D_{tube;int;LT} = 7\ [\text{cm}] \cdot \left| 0{,}010000000\ \frac{\text{m}}{\text{cm}} \right| \tag{511}$$

call $PipeFlow(R\$;\ T_{int} + 0{,}001;\ P_{int};\ \dot{m}_{ref;LP};\ D_{tube;int;LT};\ L_{int;LT};\ RelRoug)\ h$ (512)

$;\ ;\ \Delta P_{pipe;int;LT};\ ;\ ;\ )$ (513)

Assuming negligible pressure drop from throttling valve to evaporator and flash tank. Valve positioned close to inlets

————————————————————————-MISC PART————————————————————-

$COP = \dfrac{\dot{q}_{w;real;cond}}{\dot{W}_{comp;LP} + \dot{W}_{comp;HP}}$ coefficient of performance (514)

$$W_{comp;tot} = \dot{W}_{comp;LP} + \dot{W}_{comp;HP} \quad \text{compressor work} \tag{515}$$

————————————————————-VARIABLES USED FOR OPTIMIZATION———————————————————-

This section contains the variables most frequently changed for calculating the wanted variables like COP and prices

$$R\$ = \text{'R600a'} \quad \text{working fluid} \tag{516}$$

FOR 76 degree T$_{ref}$

FOR 75 degree T$_{ref}$

$$\dot{m}_{ref;LP} = 1,42 \ [\text{kg/s}] \quad \text{mass flow rate refrigerant in evaporator} \tag{517}$$

$$\dot{m}_{water;cond} = 6,25 \ [\text{kg/s}] \quad \text{water mass flow rate in condenser} \tag{518}$$

$$N_{ch;evap} = 65 \tag{519}$$

$$N_{ch;cond} = 92 \tag{520}$$

$$T_{ref;evap} = \text{ConvertTemp}(C;\ K;\ 75\ [\text{C}]) \quad \text{refrigerant inlet temperature} \tag{521}$$

FOR 74 degree T$_{ref}$

FOR 73 degree T$_{ref}$

$$k_{el} = 0,8 \ [\text{NOK/kW} \cdot \text{h}] \quad \text{elprice per kWh. Transmission and power cost combined} \tag{522}$$

$$T_{subcool} = 1 \ [\text{K}] \quad \text{subcooling temperature} \tag{523}$$

$$L_{plate} = 0,69 \ [\text{m}] \tag{524}$$

$$err_{length;evap} = abs\left(L_{plate} - L_{out;evap}\right) \quad \text{Err function for plate length} \tag{525}$$

$$err_{length;cond} = abs\left(L_{plate} - L_{out;cond}\right) \quad \text{Err function for plate length} \tag{526}$$

————————————FINDING THE U-VALUE FOR THE HEAT EXCHANGERS——————————————

Evaporator

$$\Delta_{a;evap} = T_{water;in;evap} - T_{ref;evap} \tag{527}$$

$$\Delta_{b;evap} = T_{water;out;evap} - T_{ref;out;evap} \tag{528}$$

$$\Delta_{LMTD;evap} = \frac{\left(\Delta_{a;evap} - \Delta_{b;evap}\right)}{\ln\left(\frac{\Delta_{a;evap}}{\Delta_{b;evap}}\right)} \tag{529}$$

$$U_{evap} = \frac{\dot{q}_{ref;evap}}{area_{evap} \cdot \Delta_{LMTD;evap}} \tag{530}$$

Condenser

$$\Delta_{a;cond} = T_{ref;cond} - T_{water;out;cond} \tag{531}$$

$$\Delta_{b;cond} = T_{ref;cond;out} - T_{water;in;cond} \tag{532}$$

$$\Delta_{LMTD;cond} = \frac{(\Delta_{a;cond} - \Delta_{b;cond})}{\ln\left(\frac{\Delta_{a;cond}}{\Delta_{b;cond}}\right)} \quad (533)$$

$$U_{cond} = \frac{\dot{q}_{w;real;cond}}{area_{cond} \cdot \Delta_{LMTD;cond}} \quad (534)$$

————————————————————————————————POINTS FOR DRAWING DIAGRAMS——————————————————————————

Volume flows

Volume flow outlet at evaporator outlet

$$V_{evap;outlet} = \mathrm{v}\left(R\$;\ \mathrm{T} = T_{ref;out;evap};\ \mathrm{P} = P_{evap;out}\right) \quad (535)$$

$$\dot{V}_{evap;outlet} = V_{evap;outlet} \cdot \dot{m}_{ref;LP} \quad (536)$$

$$\dot{V}_{evap;outlet;liter} = \dot{V}_{evap;outlet} \cdot \left|1000,00000\ \frac{\mathrm{l}}{\mathrm{m}^3}\right| \quad (537)$$

Volume flow at condenser inlet

$$V_{cond;inlet} = \mathrm{v}\left(R\$;\ \mathrm{T} = T_{out;comp;HP} + T_{loss;HP;cond} + 0,01;\ \mathrm{P} = P_{ref;cond}\right) \quad (538)$$

$$\dot{V}_{cond;inlet} = V_{cond;inlet} \cdot \dot{m}_{ref;HP} \quad (539)$$

$$\dot{V}_{cond;inlet;liter} = \dot{V}_{cond;inlet} \cdot \left|1000,00000\ \frac{\mathrm{l}}{\mathrm{m}^3}\right| \quad (540)$$

———————————————————————————————————PRESSURE/TEMP LOSS LOSS TUBES——————————————————————————-

Temperature loss due to pressure drops in pipes and heat exchangers

Maximum of 2 degree temp loss is accepted

TempLossFunctions

Temperature loss due to pressure loss from evaporator to LP compressor

$$T_{loss;evap} = TempLossR600a(P_{ref;evap};\ P_{evap;out}) \quad (541)$$

$$T_{loss;evap;LP} = TempLossR600a(P_{evap;out};\ P_{ref;inlet;comp;LP}) \quad (542)$$

$$T_{loss;lower} = T_{loss;evap} + T_{loss;evap;LP} \quad (543)$$

Temperature loss due to pressure loss from HP compressor to condenser outlet

$$T_{loss;HP;cond} = TempLossR600a(P_{out;comp;HP};\ P_{ref;cond}) \quad (544)$$

$$T_{loss;cond} = TempLossR600a(P_{ref;cond};\ P_{cond;out}) \quad (545)$$

$$T_{loss;higher} = T_{loss;HP;cond} + T_{loss;cond} \quad (546)$$

Other pipe sections

$$T_{loss;LP;int} = TempLossR600a(P_{out;comp;LP};\ P_{int}) \quad (547)$$

$T_{loss;int;HP} = TempLossR600a(P_{int};\ P_{ref;inlet;comp;HP})$ (548)

$T_{loss;cond;HT} = TempLossR600a(P_{cond;out};\ (P_{cond;out} - \Delta P_{pipe;cond;HT}))$ (549)

$T_{loss;int;LT} = TempLossR600a(P_{int};\ (P_{int} - \Delta P_{pipe;int;LT}))$ (550)

——————————————————————-ECONOMIC CONSIDERATIONS AND OPTMIZATION—————————————————————-

$\tau = 8000\ [\mathrm{h}]$ operating hours of the heat pump (551)

$\eta_{el} = 0,95$ (552)

Heat exchanger costs

$E_{evap} = HexPriceArea(N_{ch;evap})$ Evaporator cost (553)

$E_{cond} = HexPriceArea(N_{ch;cond})$ Condenser cost (554)

Compressor costs

$E_{comp;LP} = CompPrice\left(\dot{V}_{comp;LP}\right)$ LP comp cost (555)

$E_{comp;HP} = CompPrice\left(\dot{V}_{comp;HP}\right)$ HP comp cost (556)

pipe prices

$price_{evap;LP} = Pipeprice(D_{tube;evap;LP})$ (557)

$price_{LP;int} = Pipeprice(D_{tube;LP;int})$ (558)

$price_{int;HP} = Pipeprice(D_{tube;int;HP})$ (559)

$price_{HP;cond} = Pipeprice(D_{tube;HP;cond})$ (560)

$price_{cond;HT} = Pipeprice(D_{tube;cond;HT})$ (561)

$price_{int;LT} = Pipeprice(D_{tube;int;LT})$ (562)

$$E_{pipes} = L_{evap;LP} \cdot price_{evap;LP} + L_{LP;int} \cdot price_{LP;int} + L_{int;HP} \cdot price_{int;HP} + L_{HP;cond} \cdot price_{HP;cond} + L_{cond;HT} \cdot price_{cond;HT} + L_{int;LT} \cdot price_{int;LT}$$ total price for all pipes (563)

Total investment costs

$E_{inv} = E_{evap} + E_{cond} + E_{comp;LP} + E_{comp;HP} + E_{pipes}$ Investment costs. Sum of heat exchangers, compressors and piping costs(564)

Operational costs

$$E_{op} = k_{el} \cdot \tau \cdot \left(\frac{\dot{W}_{comp;LP}}{1000 \cdot \eta_{el}}\right) + k_{el} \cdot \tau \cdot \left(\frac{\dot{W}_{comp;HP}}{1000 \cdot \eta_{el}}\right)$$ Operational costs. The costs needed to run the compressor(565)

$r = 0,1$ interest rate (566)

$y_{invest} = 15$ depreciation rate (567)

$$a_{invest} = \frac{r}{\left(1 - (1+r)^{-y_{invest}}\right)}$$ annular coefficient (568)

$E_{tot} = a_{invest} \cdot E_{inv} + E_{op}$ yearly total costs (569)

Solution

Variables in Main program

$area_{cond} = 20,19\ [\mathrm{m}^2]$
$area_{evap} = 14,26\ [\mathrm{m}^2]$
$A_{cross;cond} = 0,0008904\ [\mathrm{m}^2]$
$A_{cross;evap} = 0,0008904\ [\mathrm{m}^2]$
$a_{invest} = 0,1315$
$\beta_{cond} = 0,7854\ [\mathrm{rad}]$
$\beta_{evap} = 0,7854\ [\mathrm{rad}]$
$b_{cond} = 0,0028\ [\mathrm{m}]$
$b_{evap} = 0,0028\ [\mathrm{m}]$
$COP = 5,39\ [\text{-}]$
$\Delta P_{evap;all} = 7071\ [\mathrm{Pa}]$
$\Delta P_{guess;comp;HP} = 193,4\ [\mathrm{Pa}]$
$\Delta P_{guess;comp;LP} = 254,2\ [\mathrm{Pa}]$
$\Delta P_{pipe;cond;HT} = 561,2\ [\mathrm{Pa}]$
$\Delta P_{pipe;evap;LP} = 1743\ [\mathrm{Pa}]$
$\Delta P_{pipe;HP;cond} = 328,9\ [\mathrm{Pa}]$
$\Delta P_{pipe;int;HP} = 952,6\ [\mathrm{Pa}]$
$\Delta P_{pipe;int;LT} = 1745\ [\mathrm{Pa}]$
$\Delta P_{pipe;LP;int} = 475,1\ [\mathrm{Pa}]$
$\Delta_{a;cond} = 5\ [\mathrm{K}]$
$\Delta_{a;evap} = 15,6\ [\mathrm{K}]$
$\Delta_{b;cond} = 18,03\ [\mathrm{K}]$
$\Delta_{b;evap} = 1,893\ [\mathrm{K}]$
$\Delta_{LMTD;cond} = 10,16\ [\mathrm{K}]$
$\Delta_{LMTD;evap} = 6,499\ [\mathrm{K}]$
$d_{h;cond} = 0,0056\ [\mathrm{m}]$
$d_{h;evap} = 0,0056\ [\mathrm{m}]$
$D_{tube} = 0,07\ [\mathrm{m}]$
$D_{tube;cond;HT} = 0,07\ [\mathrm{m}]$
$D_{tube;evap;LP} = 0,07\ [\mathrm{m}]$
$D_{tube;HP;cond} = 0,07\ [\mathrm{m}]$
$D_{tube;int;HP} = 0,07\ [\mathrm{m}]$
$D_{tube;int;LT} = 0,07\ [\mathrm{m}]$
$D_{tube;LP;int} = 0,07\ [\mathrm{m}]$
$err_{comp;HP} = 135,5\ [\mathrm{Pa}]$
$err_{comp;LP} = 220,9\ [\mathrm{Pa}]$
$err_{heat;cond} = 677,4\ [\mathrm{W}]$
$err_{heat;evap} = 173,4\ [\mathrm{W}]$
$err_{length;cond} = 3,253 \times 10^{-19}\ [\mathrm{m}]$
$err_{length;evap} = 3,253 \times 10^{-19}\ [\mathrm{m}]$
$err_{temp;cond} = 0,0006512\ [\mathrm{K}]$
$\eta_{el} = 0,95\ [\text{-}]$
$\eta_{is} = 0,7\ [\text{-}]$
$E_{comp;HP} = 105446$
$E_{comp;LP} = 109153$
$E_{cond} = 81258\ [\mathrm{NOK}]$
$E_{evap} = 63144\ [\mathrm{NOK}]$
$E_{inv} = 387681\ [\mathrm{NOK}]$
$E_{op} = 461660\ [\mathrm{NOK}]$
$E_{pipes} = 28681\ [\mathrm{NOK}]$
$E_{tot} = 512630\ [\mathrm{NOK}]$
$i_{out;comp;HP} = 691034\ [\mathrm{J/kg}]$
$i_{out;comp;LP} = 680952\ [\mathrm{J/kg}]$
$i_{ref;HT} = 513972\ [\mathrm{J/kg}]$
$i_{ref;LT} = 447739\ [\mathrm{J/kg}]$
$i_{ref;out;cond} = 514077\ [\mathrm{J/kg}]$
$i_{ref;out;evap} = 659920\ [\mathrm{J/kg}]$
$i_{ref;out;int;liq} = 447898\ [\mathrm{J/kg}]$
$i_{ref;out;int;v} = 672513\ [\mathrm{J/kg}]$
$i_{water;in;cond} = 402281\ [\mathrm{J/kg}]$
$i_{w;in;evap} = 379558\ [\mathrm{J/kg}]$
$i_{w;out;cond} = 461382\ [\mathrm{J/kg}]$
$i_{w;out;evap} = 335021\ [\mathrm{J/kg}]$
$i_{w;out;wanted;evap} = 335047\ [\mathrm{J/kg}]$
$k_{el} = 0,8\ [\mathrm{NOK/kW \cdot h}]$
$k_{tube} = 1,000 \times 10^{-6}\ [\mathrm{m}]$
$L_{cond;HT} = 8\ [\mathrm{m}]$
$L_{evap;LP} = 5\ [\mathrm{m}]$
$L_{HP;cond} = 5\ [\mathrm{m}]$
$L_{int;HP} = 2\ [\mathrm{m}]$
$L_{int;LT} = 8\ [\mathrm{m}]$
$L_{LP;int} = 2\ [\mathrm{m}]$
$L_{out;cond} = 0,69\ [\mathrm{m}]$
$L_{out;evap} = 0,69\ [\mathrm{m}]$
$L_{plate} = 0,69\ [\mathrm{m}]$
$\dot{m}_{ref;HP} = 2,087\ [\mathrm{kg/s}]$
$\dot{m}_{ref;LP} = 1,42\ [\mathrm{kg/s}]$
$\dot{m}_{water;cond} = 6,25\ [\mathrm{kg/s}]$
$\dot{m}_{water;evap} = 6,74\ [\mathrm{kg/s}]$
$N_{ch;cond} = 92\ [\text{-}]$
$N_{ch;evap} = 65\ [\text{-}]$
$price_{cond;HT} = 956\ [\mathrm{NOK/m}]$
$price_{evap;LP} = 956\ [\mathrm{NOK/m}]$
$price_{HP;cond} = 956\ [\mathrm{NOK/m}]$
$price_{int;HP} = 956\ [\mathrm{NOK/m}]$
$price_{int;LT} = 956\ [\mathrm{NOK/m}]$
$price_{LP;int} = 956\ [\mathrm{NOK/m}]$
$P_{cond;out} = 2,600 \times 10^{6}\ [\mathrm{Pa}]$
$p_{co;cond} = 0,004\ [\mathrm{m}]$
$P_{evap;out} = 1,202 \times 10^{6}\ [\mathrm{Pa}]$
$P_{HT;guess} = 2,601 \times 10^{6}\ [\mathrm{Pa}]$
$P_{int} = 1,774 \times 10^{6}\ [\mathrm{Pa}]$
$P_{LT} = 1,772 \times 10^{6}\ [\mathrm{Pa}]$
$P_{out;comp;HP} = 2,602 \times 10^{6}\ [\mathrm{Pa}]$
$P_{out;comp;LP} = 1,774 \times 10^{6}\ [\mathrm{Pa}]$
$P_{ref;cond} = 2,601 \times 10^{6}\ [\mathrm{Pa}]$
$P_{ref;evap} = 1,209 \times 10^{6}\ [\mathrm{Pa}]$
$P_{ref;inlet;comp;HP} = 1,773 \times 10^{6}\ [\mathrm{Pa}]$
$P_{ref;inlet;comp;LP} = 1,201 \times 10^{6}\ [\mathrm{Pa}]$
$P_{water;cond} = 200000\ [\mathrm{Pa}]$
$P_{water;evap} = 200000\ [\mathrm{Pa}]$
$\dot{q}_{needed;cond} = 368701\ [\mathrm{W}]$
$\dot{q}_{needed;evap} = 300000\ [\mathrm{W}]$
$\dot{q}_{ref;cond} = 369378\ [\mathrm{W}]$
$\dot{q}_{ref;evap} = 301297\ [\mathrm{W}]$
$\dot{q}_{w;evap} = 300173\ [\mathrm{W}]$
$\dot{q}_{w;real;cond} = 369378\ [\mathrm{W}]$
$r = 0,1$

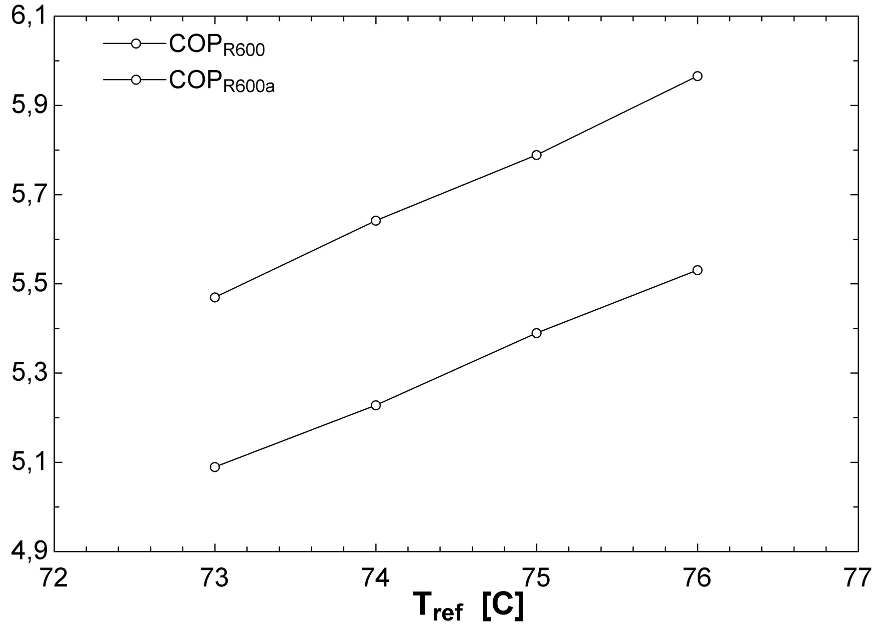
$R\$$ = ‘R600a’
$\rho_{suction;HP} = 49,32 \left[\mathrm{kg/m^3}\right]$
$stepsize = 3\ [\mathrm{m}]$
$th_{m;cond} = 0,0005\ [\mathrm{m}]$
$th_{ref;cond} = 0,0022\ [\mathrm{m}]$
$th_{water;cond} = 0,0022\ [\mathrm{m}]$
$T_{int} = 367,2\ [\mathrm{K}]$
$T_{loss;cond;HT} = -0,01231\ [\mathrm{K}]$
$T_{loss;evap;LP} = -0,0685\ [\mathrm{K}]$
$T_{loss;HP;cond} = -0,004242\ [\mathrm{K}]$
$T_{loss;int;LT} = -0,05141\ [\mathrm{K}]$
$T_{loss;LP;int} = -0,007485\ [\mathrm{K}]$
$T_{out;comp;LP} = 370,3\ [\mathrm{K}]$
$T_{ref;cond;out} = 387,2\ [\mathrm{K}]$
$T_{ref;out;evap} = 351,3\ [\mathrm{K}]$
$T_{water;in;cond} = 369,2\ [\mathrm{K}]$
$T_{water;out;cond} = 383,2\ [\mathrm{K}]$
$T_{w;real;out} = 383,2\ [\mathrm{K}]$
$UB_{evap} = 2,004\ [\text{-}]$
$U_{evap} = 3250 \left[\mathrm{W/m^2\text{-}K}\right]$
$\dot{V}_{comp;HP} = 152,4 \left[\mathrm{m^3/h}\right]$
$\dot{V}_{cond;inlet} = 0,02581 \left[\mathrm{m^3/s}\right]$
$\dot{V}_{evap;outlet} = 0,04596 \left[\mathrm{m^3/s}\right]$
$V_{evap;outlet} = 0,03236 \left[\mathrm{m^3}\right]$
$W_{comp;tot} = 68528\ [\mathrm{W}]$
$W_{cond}\$$ = ‘water’
$\dot{W}_{comp;LP} = 29866\ [\mathrm{W}]$
$x_{ref;out;cond} = -100\ [\text{-}]$

$RelRough = 0,00001429\ [\text{-}]$
$\rho_{suction;LP} = 30,84 \left[\mathrm{kg/m^3}\right]$
$\tau = 8000\ [\mathrm{H}]$
$th_{m;evap} = 0,0005\ [\mathrm{m}]$
$th_{ref;evap} = 0,0022\ [\mathrm{m}]$
$th_{water;evap} = 0,0022\ [\mathrm{m}]$
$T_{loss;cond} = -0,02479\ [\mathrm{K}]$
$T_{loss;evap} = -0,2772\ [\mathrm{K}]$
$T_{loss;higher} = -0,02903\ [\mathrm{K}]$
$T_{loss;int;HP} = -0,02806\ [\mathrm{K}]$
$T_{loss;lower} = -0,3457\ [\mathrm{K}]$
$T_{out;comp;HP} = 389,3\ [\mathrm{K}]$
$T_{ref;cond} = 388,2\ [\mathrm{K}]$
$T_{ref;evap} = 348,2\ [\mathrm{K}]$
$T_{subcool} = 1\ [\mathrm{K}]$
$T_{water;in;evap} = 363,8\ [\mathrm{K}]$
$T_{water;out;evap} = 353,2\ [\mathrm{K}]$
$UB_{cond} = 2\ [\text{-}]$
$U_{cond} = 1801 \left[\mathrm{W/m^2\text{-}K}\right]$
$V_{cond;inlet} = 0,01236 \left[\mathrm{m^3/kg}\right]$
$\dot{V}_{comp;LP} = 165,7 \left[\mathrm{m^3/h}\right]$
$\dot{V}_{cond;inlet;liter} = 25,81 \left[\mathrm{m^3/s}\right]$
$\dot{V}_{evap;outlet;liter} = 45,96\ [\mathrm{l/s}]$
$W\$$ = ‘water’
$W_{cond} = 0,318\ [\mathrm{m}]$
$\dot{W}_{comp;HP} = 38662\ [\mathrm{W}]$
$W_{evap} = 0,318\ [\mathrm{m}]$
$y_{invest} = 15$

Key Variables

$T_{loss;evap} = -0,2772\ [\mathrm{K}]$
$\dot{W}_{comp;HP} = 38662\ [\mathrm{W}]$
$E_{inv} = 387681\ [\mathrm{NOK}]$
$\dot{W}_{comp;LP} = 29866\ [\mathrm{W}]$
$err_{length;cond} = 3,253 \times 10^{-19}\ [\mathrm{m}]$
$U_{evap} = 3250 \left[\mathrm{W/m^2\text{-}K}\right]$
$L_{out;cond} = 0,69\ [\mathrm{m}]$
$err_{length;evap} = 3,253 \times 10^{-19}\ [\mathrm{m}]$
$E_{op} = 461660\ [\mathrm{NOK}]$
$L_{out;evap} = 0,69\ [\mathrm{m}]$
$E_{tot} = 512630\ [\mathrm{NOK}]$
$\dot{m}_{ref;LP} = 1,42\ [\mathrm{kg/s}]$
$COP = 5,39\ [\text{-}]$
$err_{comp;HP} = 135,5\ [\mathrm{Pa}]$
$area_{cond} = 20,19 \left[\mathrm{m^2}\right]$
$area_{evap} = 14,26 \left[\mathrm{m^2}\right]$
$err_{comp;LP} = 220,9\ [\mathrm{Pa}]$
$W_{comp;tot} = 68528\ [\mathrm{W}]$
$err_{heat;cond} = 677,4\ [\mathrm{W}]$
$err_{heat;evap} = 173,4\ [\mathrm{W}]$
$\dot{m}_{ref;HP} = 2,087\ [\mathrm{kg/s}]$
$err_{temp;cond} = 0,0006512\ [\mathrm{K}]$
$\dot{m}_{water;cond} = 6,25\ [\mathrm{kg/s}]$

Plot Window 1: $COP\ vs\ T_{ref}$



Plot Window

Plot Window 3: $E_{tot\ vs\ T_ref\ R600a}$

COMPPRICE

Equations

`$UnitSystem SI MASS RAD PA K J`

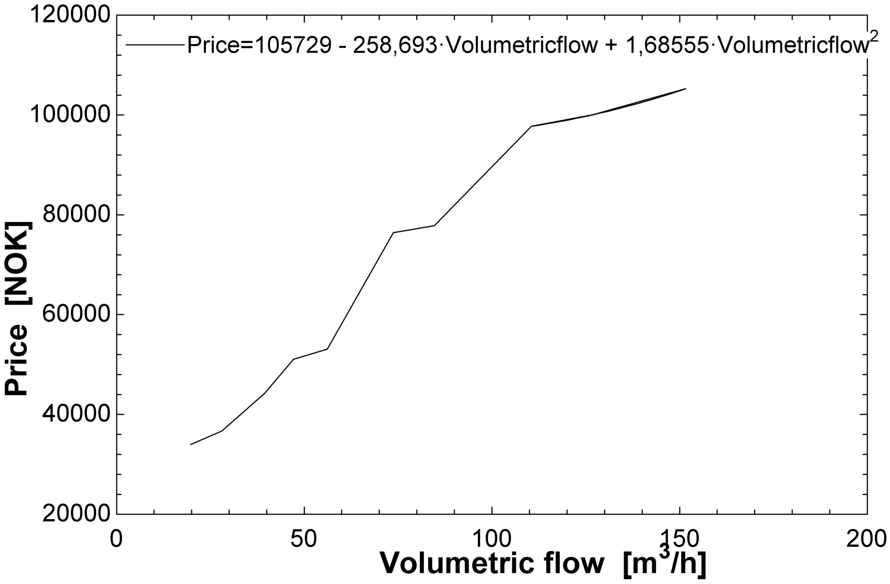
This function returns the price of a compressor as a function of the volume flow

function $CompPrice\left(\dot{V}\right)$ (1)

$$CompPrice = 105728,576 - 258,693062 \cdot \dot{V} + 1,68555164 \cdot \dot{V}^2 \quad (2)$$

end (3)

Plot Window 1: $Price\ vs\ volumeflow$



D_TUBE_MAX

Equations

Function to find the maximum allowed inside diameter of a tube

```
function $D_{tube;max}(\dot{m}_{ref};\ T_{ref};\ P_{ref};\ \rho_g)$
$Arrays On
    $D_{tube;1} = 1\ [\mathrm{cm}] \cdot \left| 0,010000000\ \frac{\mathrm{m}}{\mathrm{cm}} \right|$
    $A_{cross;1} = 0,25 \cdot \pi \cdot D_{tube;1}^2$
    $v_{g;1} = \frac{\dot{m}_{ref}}{\rho_g \cdot A_{cross;1}}$
    $\xi_1 = \rho_g \cdot \left( v_{g;1}^2 \right)$
    $i = 2$
    repeat
        $D_{tube;i} = D_{tube;i-1} + 0,001$
        $A_{cross;i} = 0,25 \cdot \pi \cdot D_{tube;i}^2$
        $v_{g;i} = \frac{\dot{m}_{ref}}{\rho_g \cdot A_{cross;i}}$
        $\xi_i = \rho_g \cdot \left( v_{g;i}^2 \right)$
        $i = i + 1$
    until $(\xi_{i-1} =< 126)$
    $D_{tube;max} = D_{tube;i-1}$
end
```

D_TUBE_MAX_LIQ

Equations

Function to find the maximum allowed pipe diameter for liquid flow

```
function $D_{tube;max;liq}(\dot{m}_{ref};\ T_{ref};\ \rho_l)$    (1)
$Arrays On
    $D_{tube;1} = 1\ [\mathrm{cm}] \cdot \left| 0{,}010000000\ \frac{\mathrm{m}}{\mathrm{cm}} \right|$    (2)
    $A_{cross;1} = 0{,}25 \cdot \pi \cdot D_{tube;1}^2$    (3)
    $v_{l;1} = \frac{\dot{m}_{ref}}{\rho_l \cdot A_{cross;1}}$    (4)
    $i = 2$    (5)
    repeat    (6)
        $D_{tube;i} = D_{tube;i-1} + 0{,}001$    (7)
        $A_{cross;i} = 0{,}25 \cdot \pi \cdot D_{tube;i}^2$    (8)
        $v_{l;i} = \frac{\dot{m}_{ref}}{\rho_l \cdot A_{cross;i}}$    (9)
        $i = i + 1$    (10)
    until $(v_{l;i-1} =< 0{,}5)$    (11)
    $D_{tube;max;liq} = D_{tube;i-1}$    (12)
end    (13)
```

PIPEPRICE

Equations

```
$UnitSystem SI MASS RAD PA K J
```

```
$Tabstops 0.2 0.3 0.6 3.5 in
```

This function calulates the price per meter pipe for a given temperature and inside diameter of the pipe

function $Pipeprice(D_{in})$ (1)

$\rho = 8957$ Density(Copper$_{RRR100}$; T=300) (2)

$D_{out} = 0,00137211672 + 1,04294329 \cdot D_{in}$ From plot (3)

$L = 1$ (4)

$V = 0,25 \cdot \pi \cdot L \cdot \left(D_{out}^2 - D_{in}^2\right)$ volume of copper part of the tube (5)

$m = \rho \cdot V$ mass of tube per meter (6)

$Pipeprice = -30,4384776 + 221,838524 \cdot m$ From plot (7)

end (8)

FROM DI TO DO

Equations

```
$UnitSystem SI MASS RAD PA K J
```

```
$Tabstops 0.2 0.3 0.6 3.5 in
```

This function calulates the price per meter pipe for a given temperature and inside diameter of the pipe

function $PipePrice(D_{in};\ T)$ (1)

$\rho = \rho\left(\text{‘copper’}\ ;\ \mathrm{T} = T\right)$ (2)

$D_{out} = 0,00137211672 + 1,04294329 \cdot D_{in}$ From plot (3)

$V = 0,25 \cdot \pi \cdot \left(D_{out}^2 - D_{in}^2\right)$ volume of copper part of the tube (4)

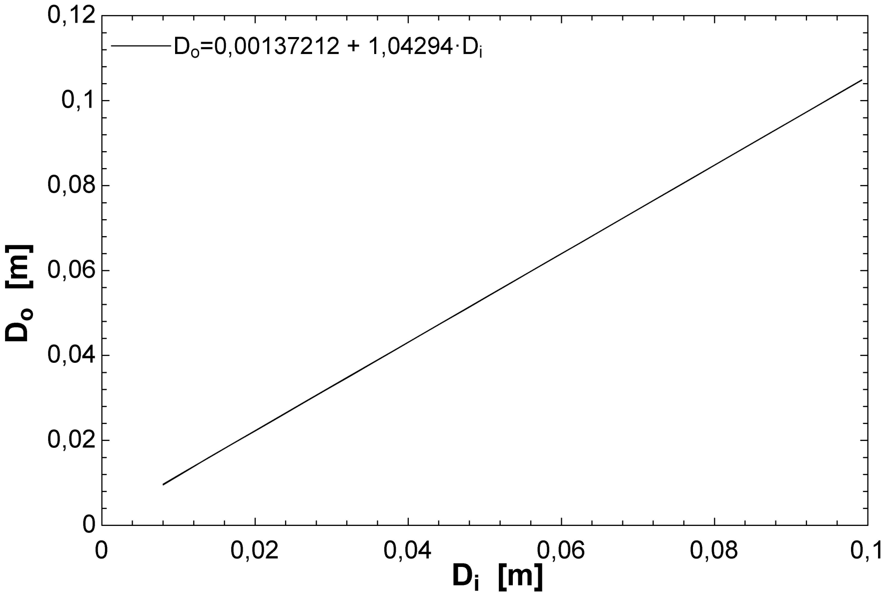
$m = \rho \cdot V$ mass of tube per meter (5)

$PipePrice = -30,4384776 + 221,838524 \cdot m$ From plot (6)

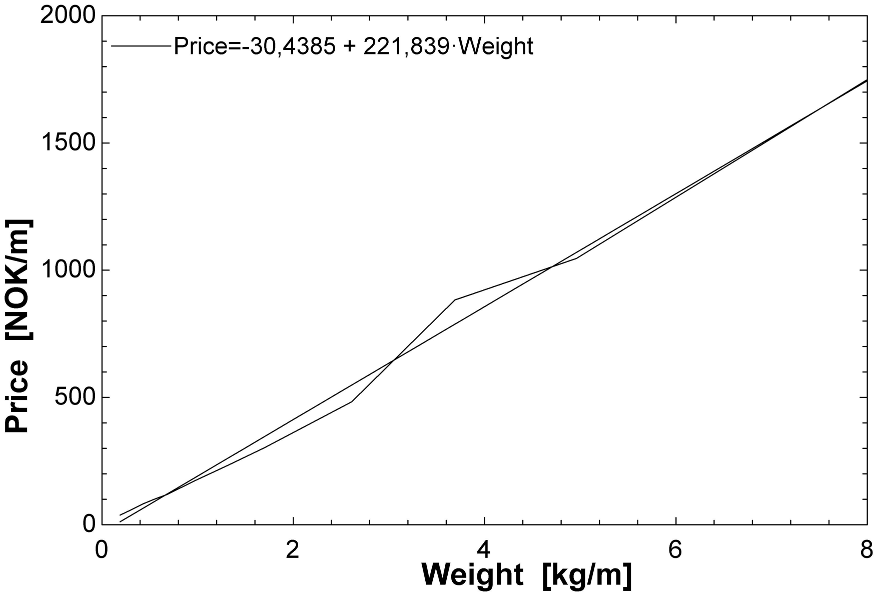
end (7)

Testing the function

Plot Window 1: *Ratio*



Plot Window 2: *Price per kg tube*



HEXPRICEAREA

Equations

```
$UnitSystem SI MASS RAD PA K J

$Tabstops 0.2 0.3 0.6 3.5 in
```

This function calulates the price per meter pipe for a given temperature and inside diameter of the pipe

```
function $HexPriceArea(N_{ch})$
    $N_{plates} = N_{ch} + 2$    number of plates,+2 for end plates
    $W = 0,318\ [\mathrm{m}]$    width of plate
    $L = 0,69\ [\mathrm{m}]$    length of plate
    $area_{plate} = W \cdot L$    area plate
    $area_{tot} = area_{plate} \cdot N_{plates}$
```

Two distinct cost functions

For area less than 11 m^2

```
    If$(area_{tot} < 11)$ then
        $HexPriceArea = 35802,3 + 1498,04 \cdot area_{tot}$    From plot
    else
        $HexPriceArea = 18194,1373 + 3057,57961 \cdot area_{tot}$    From plot
    endif
end
```

TEMPLOSSR600

Equations

`$UnitSystem SI MASS RAD PA K J`

`$Tabstops 0.2 0.3 0.6 3.5 in`

Function which calculates the temperature loss due to pressure loss for R600

————————————CURVE FIT FRA 74 GRADER CELSIUS TIL 116 GRADER CELSIUS——————————————

$$\text{function } TempLossR600(P_{in};\ P_{out}) \tag{1}$$

T=271,184983 + 0,000147704853*P - 1,13999686E-10*P^2 + 6,91983866E-17*P^3&
- 2,68613690E-23*P^4 + 5,90088078E-30*P^5 - 5,57661611E-37*P^6

$$T_{in} = 271,184983 + 0,000147704853 \cdot P_{in} - 1,13999686 \times 10^{-10} \cdot P_{in}^2 + 6,91983866 \times 10^{-17} \cdot P_{in}^3 - 2,68613690 \times 10^{-23} \cdot P_{in}^4 + 5,90088078 \times 10^{-30} \cdot P_{in}^5 - 5,57661611 \times 10^{-37} \cdot P_{in}^6 \tag{2}$$

$$T_{out} = 271,184983 + 0,000147704853 \cdot P_{out} - 1,13999686 \times 10^{-10} \cdot P_{out}^2 + 6,91983866 \times 10^{-17} \cdot P_{out}^3 - 2,68613690 \times 10^{-23} \cdot P_{out}^4 + 5,90088078 \times 10^{-30} \cdot P_{out}^5 - 5,57661611 \times 10^{-37} \cdot P_{out}^6 \tag{3}$$

$$TempLossR600 = T_{out} - T_{in} \tag{4}$$

$$\text{end} \tag{5}$$

TEMPLOSSR600A

Equations

```
$UnitSystem SI MASS RAD PA K J
```

```
$Tabstops 0.2 0.3 0.6 3.5 in
```

function $TempLossR600a(P_{in};\ P_{out})$ (1)

T=266,768389 + 0,000117658699*P - 6,89653400E-11*P^2 + 3,19442289E-17*P^3 - &
9,50215605E-24*P^4 + 1,60255038E-30*P^5 - 1,16497777E-37*P^6

$$T_{in} = 266,768389 + 0,000117658699 \cdot P_{in} - 6,89653400 \times 10^{-11} \cdot P_{in}^2 + 3,19442289 \times 10^{-17} \cdot P_{in}^3 - 9,50215605 \times 10^{-24} \cdot P_{in}^4 + 1,60255038 \times 10^{-30} \cdot P_{in}^5 - 1,16497777 \times 10^{-37} \cdot P_{in}^6 \quad (2)$$

$$T_{out} = 266,768389 + 0,000117658699 \cdot P_{out} - 6,89653400 \times 10^{-11} \cdot P_{out}^2 + 3,19442289 \times 10^{-17} \cdot P_{out}^3 - 9,50215605 \times 10^{-24} \cdot P_{out}^4 + 1,60255038 \times 10^{-30} \cdot P_{out}^5 - 1,16497777 \times 10^{-37} \cdot P_{out}^6 \quad (3)$$

$$TempLossR600a = T_{out} - T_{in} \quad (4)$$

end (5)

GENERATELOOKUPTABLE

Equations

Function for making lookuptable of saturated temp vs pressure

```
function $max_{tubeloss}(R\$)$                                    (1)
$Arrays On
    $T_1 = \text{ConvertTemp}(C;\ K;\ 0\ [\text{C}])$                    (2)
    $P_1 = \text{P}\,(R\$;\ \text{T} = T_1;\ \text{x} = 1)$               (3)
    $T_{crit} = \text{T}_{\text{crit}}\,(R\$)$                          (4)
    $N = \text{Round}(T_{crit} - T_1) - 1$                            (5)
    duplicate $i = 2;\ N$                                            (6)
        $T_i = T_{i-1} + 1$                                          (7)
        $P_i = \text{P}\,(R\$;\ \text{T} = T_i;\ \text{x} = 1)$           (8)
        $max_{tubeloss} = T_1$                                       (9)
    end                                                             (10)
```

So far so good, we get the values we want into an array

```
    $i$                                                             (11)
     $= 0$                                                          (12)
    repeat                                                          (13)
        $i$                                                         (14)
         $= i + 1$                                                  (15)
        Lookup('R600' ; $i$; 1) $= T_i$                               (16)
    until $(i \geq N)$                                              (17)
    $j = 0$                                                         (18)
    repeat                                                          (19)
        $j$                                                         (20)
         $= j + 1$                                                  (21)
        Lookup('R600' ; $j$; 2) $= P_j$                               (22)
    until $(j \geq N)$                                              (23)
```

We now have the values in a the lookup table

```
end                                                                 (24)
```

Prices for a M110 LFX plate heat exchanger for various plate numbers.

Prices given by Schløsser Kulde AS.

Plate dimensions	
Width [m]	0,318
Length [m]	0,69
Area per plate {m^2]	0,21942

Number of plates	Price [NOK]	Area {m^2]	Price per area [NOK/m^2]
30	45707	6,5826	6943,608908
40	48863	8,7768	5567,29104
50	52281	10,971	4765,38146
60	58428	13,1652	4438,063987
70	64657	15,3594	4209,604542
80	71690	17,5536	4084,062528
100	85046	21,942	3875,945675
120	99092	26,3304	3763,406557

Prices for copper tubes given by Schløsser Kulde AS

Outside diameter [inches]	Outside diameter [mm]	Outside diameter diameter [m]
3/8	9,53	0,009525
1/2	12,70	0,0127
5/8	15,88	0,015875
3/4	19,05	0,01905
7/8	22,23	0,022225
1 1/8	28,58	0,028575
1 3/8	34,93	0,034925
1 5/8	41,28	0,041275
2,125	53,98	0,053975
2 5/8	66,68	0,066675
3 1/8	79,38	0,079375
4 1/8	104,78	0,104775

Wall thickness [mm]	Inside diameter [m]	Cross sectional area [m^2]
0,76	0,008005	5,03283E-05
0,89	0,01092	9,36559E-05
1,02	0,013835	0,000150331
1,07	0,01691	0,000224583
1,14	0,019945	0,000312434
1,27	0,026035	0,00053236
1,4	0,032125	0,000810543
1,52	0,038235	0,001148186
1,78	0,050415	0,001996225
2,03	0,062615	0,003079262
2,29	0,074795	0,004393747
2,79	0,099195	0,007728041

Price per meter [NOK/m]	Weight per meter [kg/m]	Ratio outside/inside
38	0,19	1,19
58	0,3	1,16
82	0,43	1,15
99	0,54	1,13
119	0,68	1,11
173	0,97	1,10
234	1,32	1,09
303	1,7	1,08
483	2,61	1,07
883	3,69	1,06
1046	4,96	1,06
1747	7,99	1,06

ρ_copper @100celsius [kg/m	Area copper {m^2]	Weight per meter pipe [kg/m]
8924	2,09274E-05	0,186756165
	3,3021E-05	0,294679103
	4,76017E-05	0,424797803
	6,04398E-05	0,539365172
	7,55141E-05	0,673888261
	0,000108942	0,972199372
	0,000147451	1,315849611
	0,000189839	1,694122381
	0,000291876	2,604703769
	0,000412269	3,679090001
	0,000554569	4,948969683
	0,000893903	7,97719003

Prices for a Bitzer Open Drive Reciprocating compressor. Prices given by Schløsser Kulde AS

Displacement volume [m^3/h]	Price [NOK]	
19,7	33985	2-cylinders
28	36666	
39,4	44193	4-cylinders
47,1	51041	
56,1	53094	
73,7	76436	
84,6	77806	
110,5	97724	6-cylinders
126,8	100027	
151,6	105249	