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Fredrik Bjørvik

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NTNU
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Department of Energy and Process Engineering

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Design of a hybrid absorption/compression high temperature heat pump test rig

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Fredrik Bjørvik

Mechanical Engineering
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Supervisor: Trygve M. Eikevik
Co-supervisor: Ignat Tolstorebrov, Armin Hafner and Marcel U. Ahrens

Norwegian University of Science and Technology
Department of Energy and Process Engineering

Norwegian University of Science and Technology Engineering

Faculty of Engineering Department of Energy and Process

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MASTER THESIS

for

student Fredrik Bjørvik

Spring 2019

Design of a hybrid absorption/compression high temperature heat pump test rig

Designe en hybrid absorpsjons/kompresjons høytemperatur varmepumpe testrigg

Background and objective

The heat pump market has so far mainly focused on residential heat pumps for space heating and domestic hot water production. Less focus has been on heat pumps for higher temperature applications and industrial use, due to high initial investment costs, competition with alternative investments, and non-mature or non-existing technologies for the applications. New developments in compact high-pressure components, e.g. compressors, ejectors and heat exchangers for CO_2, ammonia and hydrocarbon heat pump systems, are important drivers to change this situation.

This master thesis will concentrate on the design of a high temperature hybrid heat pump for production of hot water from surplus heat. During this year there will be build a prototype test rig in the laboratory of NTNU-EPT. The thesis will be focused on the design of the test rig. This will implies the design and simulation of the operatonal conditions the test rig should operate under. The thesis will give deep insight in both design and modelling of the system for optimal operational condition. The test rig should have the focus on the high temperature side, the absorber, to be able to test different concepts of the heat exchanger and also different compressor arrangements.

The following tasks are to be considered:

1. Literature review of compression absorption heat pump with focus on the absorber and compressor.
2. Theoretical description of the absorption/compression HT-process
3. Make a description of the test rig and its optimal operational conditions.
4. Improve the simulation tool from the project work based on the test rig.
5. Make a scientific paper based on the result from this thesis.
6. Make proposal for further work.

Preface

This master thesis has been written for the Department of Energy and Process Engineering at NTNU during January 2019 to June 2019. The thesis is about designing an absorption/compression heat pump delivering high temperature heat using ammonia/water mixture as working fluid. Some of the work in this thesis is based on the work from Bjørvik (2018), which was a pre-work for this thesis and could be find at the Department of Energy and Process Engineering at NTNU.

I would like to thank my supervisor Trygve M. Eikevik for being available and very helpful during the work on this thesis.

I would also give a great thanks to my co-supervisors Ignat Tolstorebrov, Armin Hafner and Marcel U. Ahrens for a good discussion and great guidance.

Trondheim, Norway, 11th June 2019

Fredrik Bjørvik

Summary

Today the energy consumption worldwide is increasing, and the manufacturing sector have a big share of the world energy use (EIA, 2017). 38% of the global energy use in the manufacturing sector was from steam systems. There a big amount of the surplus heat from this steam system is not utilized, since the heat is often of too low temperatures and cannot be utilize directly in an industrial process (Banerjee et al., 2012). Hybrid absorption/compression heat pump (HACHP) is one of the best ways to utilize the surplus heat and to elevate the temperature, while reducing the use of primary energy for heating of the steam (Brunin et al., 1997). The HACHP is a combination of vapour-compression heat pump and absorption heat pump using ammonia/water as working fluid. With use of a binary fluid it is easier to achieve capacity control due to the extra degree of freedom, and temperature glide in the desorber and absorber will occur, which will reduce the irreversibility of the system. Moreover, it is easier to achieve higher temperature than for vapour compression heat pump using ammonia at relative low pressures.

Five different HACHP models with different system configuration were made to evaluate, which of them was the best option to use as a test rig. The models are further development of the simulation model made by Bjørvik (2018). All configurations were teste in four different cases with different input parameters there the working fluid absorb heat from water at 50 ^{0}C in all cases and the inlet sink temperature was also 50 ^{0}C. The injection from lean solution was considered as the best solution for the test rig, achieving to heat the sink from 50 ^{0}C up to 109.5 ^{0}C in case 3 with a coefficient of performances (COP) on 3.28. Injection from the lean solution consist of a screw compressor, which get liquid injection from the lean solution to lubricate, seal and cool down the compressor, since one of the main constrain in the heat pump is the compressor discharges temperature.

Moreover, the same type of screw compressor with two different volume ratios at 3.65 and 5.80 were tested in the simulation model with injection from lean solution to figured out, which of them suits the test rig best. The conclusion was that the one with a volume ratio at 3.65 was the best option. Furthermore, the simulation model with the injection from lean solution and the compressor with a volume ratio at 3.65 were teste with different inputs values for different input parameters to find the optimal operational condition. The optimal injection ratio was concluded to be from 0.07 to 0.12, while the optimal circulation ratio for the test rig was from 0.55 to 0.60. With an injection ratio at 0.1 and a circulation ratio at 0.57 the outlet sink temperature were 97.55 ^0C with an COP at 3.74. To validate the models, they were compared with results from other research. In the comparison with Nordtvedt (2005) the deviation in the achieved outlet sink temperature was at 0.32%, while the COP had a deviation on 2.79%.

Sammendrag

Energiforbruket i verden i dag er økende og produksjonssektoren har en stor andel av energiforbruket (EIA, 2017). 38% av det globale energiforbruket i produksjonssektoren stammer fra dampsystemer. Mye av overskuddsvarmen fra dampsystemene blir ikke brukt siden varmen fra disse er av for lav temperatur til å bli brukt direkte i en industriellprosess (Banerjee et al., 2012). En av de beste måtene å utnytte overskuddsvarmen på og øke temperaturen på den er ved hjelp av en absorpsjons/kompresjon varmepumpe, som også vil minke det primære energiforbruket som går med til oppvarming av dampen (Brunin et al., 1997). Absorpsjons/kompresjon varmepumpe er en kombinasjon dampkomprimerings-varmepumpe og absorbsjonsvarmepumpe som bruker en blanding av ammoniakk og vann som arbeidsmedium. Ved bruk av en binær blanding så er det lettere å oppnå kapasitetskontroll på grunn av den ekstra frihetsgraden i systemet. I tillegg vil det oppstå en gildene temperatur i absorberen og desorberen som vil føre til reduksjon i irreversibiliteten til systemet. Det vil også være lettere å oppnå høyere temperaturer for et relativt lavt trykk enn med en dampkomprimerings-varmepumpe som bruker ammoniakk.

Fem forskjellige absorpsjons/kompresjon varmepumpemodeller med forskjellige systemkonfigurasjoner ble laget for å undersøke hvilken av de som var best til å bli benyttet som en testrigg. Modellene er en videreutvikling av modellen som ble laget av Bjørvik (2018). Alle de ulike modellene ble testet i fire forskjellige simuleringer med forskjellige inngangsparametere. Fast for alle simuleringene var at varmekilden hadde en temperatur på 50 ^{0}C og at varmesluket også hadde en inngangstemperatur på 50 ^{0}C. Den beste konfigurasjonen å bruke som en testrigg var den med innsprøytning fra løsningskretsen og den oppnådde en utgangstemperatur i varmesluket på 109,5 ^{0}C i simulering nummer 3 med en effektfaktor (COP) på 3.28. Denne konfigurasjonen inneholder en skruekompressor som får innsprøytning av væske fra løsningskretsen for å smøre, tette og kjøle ned kompressoren. Siden en av hoved begrensningene til varmepumpen er kompressorens trykkgasstemperatur.

Videre ble samme type kompressor med to forskjellige volumforhold på henholdsvis 3.65 og 5.80 testet i simuleringsmodellen for å finne ut hvilken av de som passet testriggen best. Konklusjonen ble at den som hadde et volumforhold på 3,65 var den beste løsningen. Så ble

simuleringsmodellen med innsprøytning av væske fra løsningskretsen og kompressoren med et volumforhold på 3,65 testet med forskjellige verdier for inngangsparameterne. Dette var for å finne de optimale driftsforholdene. Det optimale innsprøytnings forholde ble funnet til å være fra 0,07 til 0,12, mens det optimale sirkulasjonsforholdet for testriggen ble funnet til å være fra 0,55 til 0,60. Med et innsprøytnings forhold på 0,1 og et sirkulasjonsforhold på 0,57 ble den oppnådde temperaturen i varmesluket ble på 97,55 ^{0}C, med en COP på 3,74. For å validere modellen ble resultater fra andre studier sammenlignet med resultatene fra modellene. I sammenligningen med Nordtvedt (2005) ble avviket for oppnådd temperatur i varmesluket på 0,32%, mens COP hadde et avvik på 2,79%.

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Nomenclature

Latin letters

$\dot{C}$	Capacitances rate	kW/k
C_p	Specific heat capacity	kJ/kg*K
CR	Circulation ratio	-
h	Specific enthalpy	kJ/kg
K	Correction factor	-
$\dot{m}$	Mass flow rate	Kg/s
P	Pressure	bar
PR	Pressure ratio	-
q	Vapour quality	-
$\dot{Q}$	Rate of heat transfer	kW
s	Entropy	kJ/kg*K
T	Temperature	K or ^{0}C
$\overline{T}$	Averages temperature	K or ^{0}C
u	Internal energy	kJ/kg
v	Specific volume	m^3/kg
V	Volume	m^3
$\dot{V}$	Volumetric flow	m^3/s
W	Injection ratio	-
$\dot{W}$	Work	kW
x	Ammonia mass fraction	-

Greek letters

γ	Isentropic exponent	-
Δ	Difference	-
ε	Thermal efficiency	-
η	Efficiency	-
η_{II}	Second law efficiency	-

Subscripts

4	State point 4	-
5	State point 5	-

Abs	Absorber	-
Co	Cold	-
Comp	Compressor	-
Des	Sources flow in desorber	-
Deso	Desorber	-
dsh	Desuperheater	-
ho	Hot	-
HP	Absorber pressure	-
in	Inlet	-
innj	Injection	-
isen	Isentropic	-
liq	Lean solution	-
LM	Logarithmic mean	-
Lorenz	Lorenz cycle	-
LP	Desorber pressure	-
max	Maximum	-
Min	Minimum	-
MP	Injection pressure/intermediate pressure	-
Out	outlet	-
Pump	Solution pump	-
rec	Reciprocating compressor	-
rich	Rich solution	-
screw	Screw compressor	-
swept	Swept volume	-
vapo	Vapour	-
volu	Volumetric	-
vp1	Inlet compressor	-
WD	Rich solution in desorber	-

Abbreviations

CFC	Chlorofluorocarbon	-
COP	Coefficient of performances	-

EES	Energy equation solver	-
HACHP	Hybrid absorption/compression high temperature heat pump	-
HCFC	Hydrochlorofluorocarbon	-
IHEX	Internal heat exchanger	-
NTNU	Norwegian University of Science and Technology	-
VHC	Volumetric heating capacity	-
VI	Volume ratio	-

1 Introduction

1.1 Background

Today the energy consumption worldwide is increasing, and the energy increase is expected to grow with 28% from 2015 to 2040 (EIA, 2017). The manufacturing sector has a big share of the total energy consumption and in 2005, 38% of the global energy use in the manufacturing sector was from steam system (Banerjee et al., 2012). One way to reduce the energy consumption is to utilize the surplus heat from this steam system to heat up working fluid, which can elevate the temperature or to heat up other applications directly. This will lead to both smaller operating costs and lower greenhouse gas emission.

Surplus heat is often of to low temperatures and cannot be utilize directly in an industrial process, which often need high temperatures (Nordtvedt, 2005). Today a lot of the high temperatures processes in the industry is generated from primary energy and not the surplus heat, but the surplus heat is good to use as a heat source for an industrial heat pump system. However, so far, the industrial heat pump has had less focus than the residential heat pumps, because of the high investment cost and more complex specification, which is one of the reasons that surplus heat have not be utilized. The latest years new technologies with compact high-pressure components have been develop and this will probably change the situation and most likely more focus will be on industrial heat pump in the further. Whatever, if the investment cost of the industrial heat pump is high, the usages time of an industrial heat pump compared to a residential heat pump is often much higher, so the payback time could be lower.

Global warming and the increase of the temperature on the earth have had a lot of focus the last decades. This have led to restrictions in the heat pump industry, like the Montreal protocol in 1987 where the phase out of the CFCs and HCFCs refrigerant start. After the Montreal protocol was introduce, some other restriction has been implemented to do the heat pump more environmentally friendly. To handle the restriction, new technologies have to be developing to make the heat pump market more economical beneficial and suitable for high temperatures (Calm, 2008).

One of the best ways to handle the restriction of CFCs and HCFCs according to Brunin et al. (1997) where to use the HACHP with ammonia/water mixture as working fluid. Some of the benefits Brunin et al. (1997) propose for the HACHP is the high temperature lift with relatively small pressure ratios. The smaller pressure ratio will reduce the compressor work and the efficiency of the system will increase.

1.2 Objectives

The objectives for this thesis are to design a high temperature HACHP test rig for heating of water from surplus heat, there a prototype will be built at NTNU this year. The thesis should have focus on the optimal operational condition, design, and to make a simulation tool for the test rig. This master thesis should also have focus on the high temperature side to test different concept of compressor and absorbers arrangement in the future test rig. In this thesis these tasks are going to be answer:

- Carry out a literature review of previous HACHP with focus on compressor and absorber.
- Describe how the HACHP works.
- Set the optimal operational condition, decide the heat pump configuration and describe the test rig.
- Make a simulation tool for the planned test rig based on the work from the project thesis.
- Make a scientific paper with highlights from this thesis.
- Make suggestion for further work.

1.3 Outline of thesis

Chapter 2 gives a fundamental explanation of the HACHP with the advantages and constrains. The characteristics of a zeotropic is fluid is also explained

Chapter 3 present the current status of an HACHP with focus on the absorbers and compressors.

Chapter 4 describe the setup of the simulation model.

Chapter 5 explained how to use the simulation tool.

Chapter 6 present, validate and discuss the result. In addition, the setup of the different simulation is explained.

Chapter 7 gives the conclusion and suggestion for further work

2 Absorption/compression heat pump

Absorption/compression heat pump is a vapour compression heat pump in combination with an absorption heat pump using a zeotropic working fluid. The HACHP have some advantages compared with a normal vapour compression heat pump and is especially good for high temperature lift. (Brunin et al., 1997) (Nordtvedt, 2005).

2.1 Zeotropic fluid

A zeotropic fluid is two or more components in a mixture with different volatilities (Sweeney and Chato, 1996). In azeotropic fluid, the boiling temperature is constant and decided by the pressure, but in a zeotropic fluid the boiling temperature is varying, and it is decided composition of the components and pressure. In a zeotropic mixture, the component with the highest boiling point is known as the absorbent and the component that is most volatile is named working fluid. A common mixture is ammonia/water their ammonia has the highest volatility and therefore is the working fluid and water is the absorbent (Jensen, 2015).

With using a zeotropic fluid, it is possible to achieve higher temperatures than for vapour compression heat pump using ammonia, since the condensing pressure get lower for an ammonia/water mixture. As seen in Figure 2.1 ammonia (R717) condense on 79.4 ^{0}C when the pressure is 41 bar. With an ammonia concentration on 10 weigh-% and 20 bar pressure the saturation temperature is 180 ^{0}C if the other component is water (R718) (Jensen, 2015), (Alefeld and Radermacher, 1993), (Nordtvedt, 2005).

Another advantage Nordtvedt (2005) is mentioning by using zeotropic fluid is capacity control. The capacity control with use of a non-zeotropic fluid is with chancing the pressure and the mass flow rate, while the zeotropic fluid could in addition achieve capacity control with adjusting the ammonia concentration in the absorber and desorber. As seen in Figure 2.1 a reduction of ammonia concentration will reduce the temperature for a constant pressure. By chancing the ratio of fluid going through the pump and compressor, the ammonia concentration will increase or decrease and a capacity changes will occur.

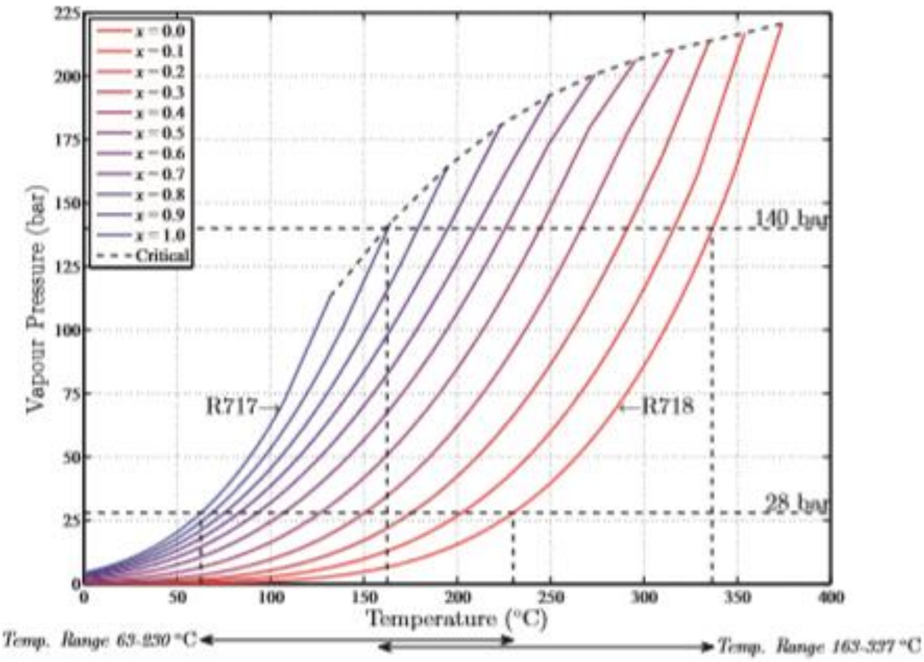

Figure 2.1: Vapour pressure, temperature, concentration diagram for ammonia/water (Jensen, 2015).

Furthermore, the last benefit Nordtvedt (2005) mentioning is that the phase changes in the desorber and absorber are non-isothermal, because the composition of the mixture that have not evaporate are changing in the desorber, which leads to a different boiling point through the process. Therefore, if the temperature glide in the absorber and desorber are matching the temperature glide in the heat sink and heat sources it will reduce the losses in the heat exchangers and increase the COP of the system. A vapour compression heat pump has an isothermal process of evaporation and condensation. Therefore, the vapour compression heat pump approaches a Carnot cycle, but with Zeotropic fluid as mentioned above the process is non-isothermal and it will approach the Lorenz cycle (Jensen et al., 2015). The differences between a Lorenz cycle and Carnot cycle is show in Figure 2.2. In addition, it is possible to see the definition of temperature lift, temperature lift process and the temperature glide.

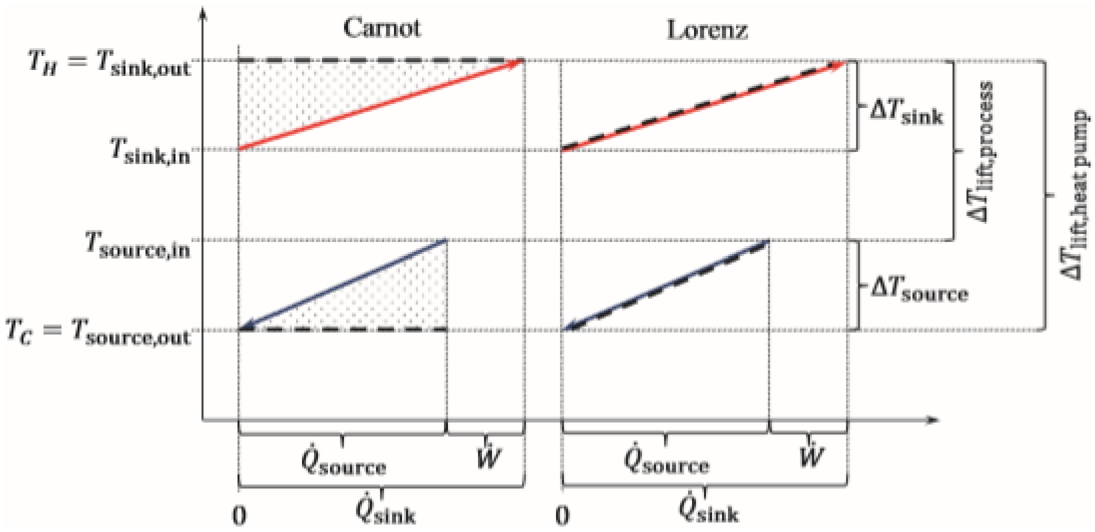

Figure 2.2: The differences between Carnot and Lorenz cycle, where the black dotted lines follows this two ideal approach (Jensen, 2015).

To understand the behaviour of zeotropic fluid it is easy to see on a temperature-concentration curve. Figure 2.3 show a temperature-concentration curve for an ammonia water combination at a constant pressure (Ganesh and Srinivas, 2011). There the dew point line is where the first liquid droplets is formed when the solution is cooled down from superheated heated vapour at a given pressure and the bubble point line is where the first vapour bubble is made from the subcooled area. Between those two lines, it is a two-phase area where it is a combination of liquid and vapour. Where X equal 0 is a pure solution of water and the point where the two lines crosses each other on the left-hand side is the saturation temperature for water ($T_{W,Sat}$) for a given pressure. On the other hand, where X equal 1 it is pure ammonia and $T_{A,Sat}$ is the saturation temperature for ammonia at a given pressure.

In Figure 2.3 it is possible to see the heating process for a zeotropic fluid with a constant pressure starting with X weight-% of ammonia in the subcooled area and a temperature correspond to point 1, which is lower than T_{Bubble}. When heating the fluid, it will reach the boiling point after a certain time and the concentration of ammonia in the vapour is X_1^V. Further heating leads to more of the solution evaporates and the temperature increase, which is different from a pure component evaporation. When the heating reach point 2 the weight-% of ammonia in the liquid is X_2^L, which is smaller than the concentration at the start of the evaporation. Therefore, the boiling point chances and the concentration in the vapour is X_2^v. Further, it will reach the dew point temperature and the evaporation process is completed. From Figure 2.3 the concentration of ammonia in liquid when the first droplet start to condensate is X^L. Adding more heat to the solution, it will end up in the superheated area at point 3.

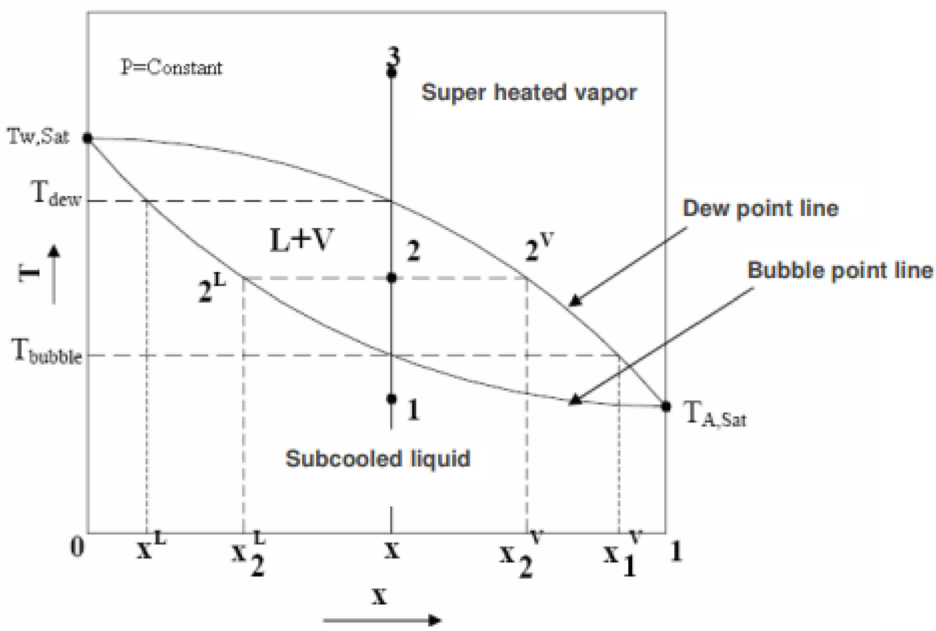

Figure 2.3: Temperature-concentration curve for ammonia water (Ganesh and Srinivas, 2011).

2.2 Absorption/compression heat pump process

HACHP process is based on the Osenbrück (1895) cycle from 1895 where the evaporation and condensation process is substituted with desorption and absorption process as seen in Figure 2.4 (Jensen et al., 2015).

In the desorber the zeotropic fluid is supplied heat from a heat source and starts evaporating, at the end of the desorber it is a two-phase mixture of saturated vapour and saturated liquid in thermal equilibrium. In an ammonia/water combination, the vapour is mainly consisting of ammonia, but at low pressure the water content could be significantly. The liquid (lean solution) contains of water and ammonia. From the desorber the mixture goes through a separator before the vapour goes to a compressor to increase the pressure and temperature (point 1 to 2) and the liquid is going through a solution pump to elevating the pressure (point 3 to 4). After the pump the lean solution enter the heat exchanger at point 4 to increase the temperature of the lean solution. Then the vapour and lean solution is mixing at the entrances of the absorber at respectively point 2 and 5. There the vapour condense, and the lean solution absorbs the vapour, while it realises heat to the sink. Through the absorber (point 2/5 to 6) the absorption process take place at gliding temperatures and the ammonia concentration in the liquid is increasing through the process to a rich solution. After the absorber the rich solution is entering the internal exchanger (point 6-7) and the rich solution is cooled down by the lean solution. Further, the rich solution passes through the expansion valve (point 7 to 8) to decrease the pressure and temperature before it entering the desorber and ready again to absorb heat from the sources (Jensen et al., 2015), (Nordtvedt, 2005).

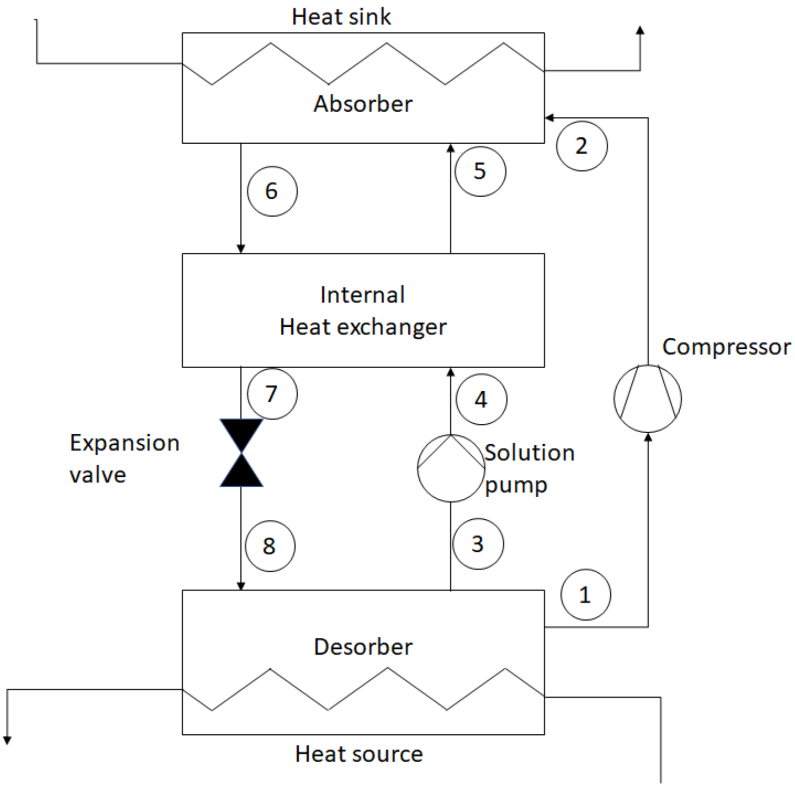

Figure 2.4: Osenbrück (1895) cycle.

To sum up chapter 2, the HACHP is a good heat pump to achieve high sink temperatures in an efficient way because of the advantages with using a zeotropic fluid.

3 Literature review

3.1 Two stage absorption/compression heat pump

Stene (1993) imply that one of the limitations for the HACHP is the compressor discharge temperature, because of the solubility of the lubricant in the compressor. One solution to solve the high discharged temperature is to use two compressors instead of one. This will reduce the pressure ratio in each of the compressor, which will reduce the discharged temperature, if the vapour is cool down between the compressors (Jensen, 2015). Nordtvedt (2005) made a research how the pressure ratio is influencing the discharged temperature and as seen in Figure 3.1 the small increase in pressure ratio could have a big impact on the discharge temperatures.

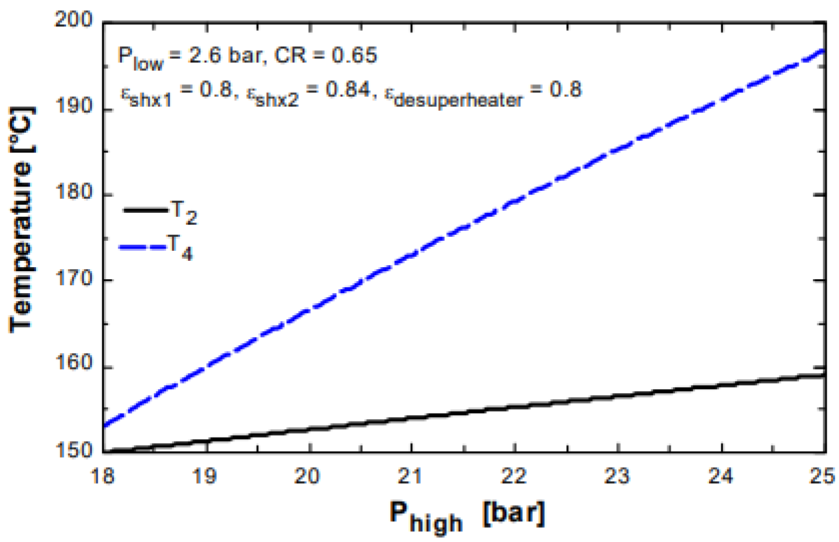

Figure 3.1: Discharge temperatures versus absorber pressure with constant desorber pressure. There T2 is the discharges temperature for the first compression and T4 is the discharges temperature for the second compression (Nordtvedt, 2005).

Jensen (2015) show some different ways to implementing a two stages compression and the two stages method, which gave the highest COP according to Jensen (2015) is a two-stage HACHP with two internal heat exchanger as showed in Figure 3.2. The two stages compression is often more efficient than one stages regardless, which method is used.

The HACHP with internal heat exchanger have two heat exchangers there one of the heat exchangers (component 9) deliver heat from the rich solution to the lean solution, before the lean solution entering the other heat exchanger, which deliver heat from the vapour to the lean solution. This heat exchanger is mainly to cool down the vapour before it enters the second compressor, but also to heat up the lean solution to reduce the entropy losses when mixing liquid and vapour in the mixer. Another improvement is the desuperheater (gas cooler), which cool down the vapour before entering the mixer and at the same time deliver heat to the sink. This is to lower the temperature difference in the mixer between the vapour and the lean solution. Bergland (2015) shows that the difference by using a desuperheater and not using a desuperheater was small when it comes to the temperature out of the sink, if the vapour either was cool down to the saturation temperature or the temperature of the lean solution.

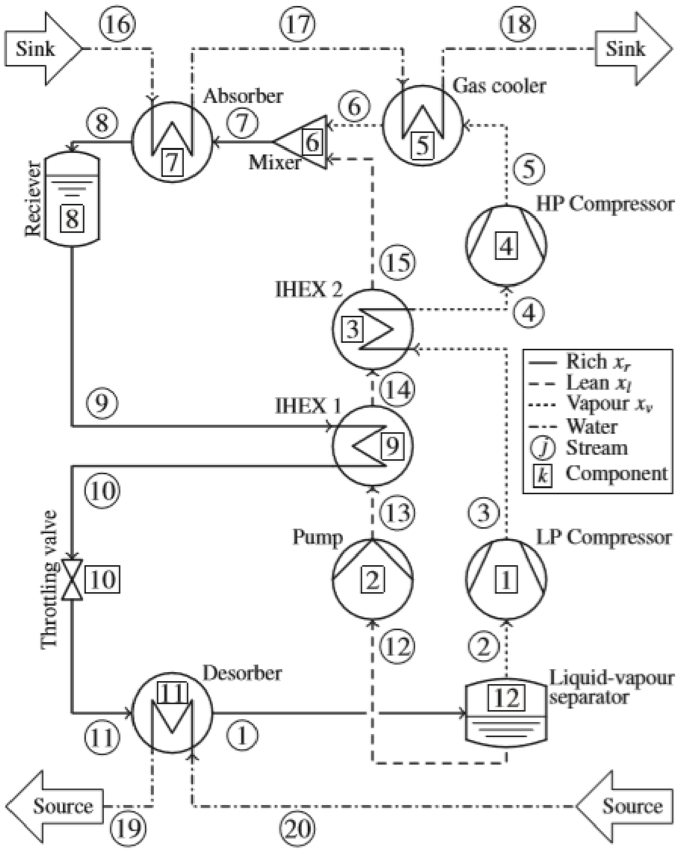

Figure 3.2: Two-stages HACHP with internal heat exchanger (Jensen, 2015).

Jensen (2015) was also considering some other option for a two-stage compressor system and one of the other solutions was to place the heat exchanger to the vapour before the heat exchanger to the rich solution when following the flow of the lean solution. Another suggestion was to have a two stages compression system with a bubble through inter-cooler, to cool down the vapour to saturation temperature before the second compressors. The last solution was to have liquid injection from the rich solution to cool down the vapour to saturation temperature between the compressors. In the simulation Jensen (2015) had a fixed temperature lift and fixed temperature glide in both the sink and the sources with the same value for all the configuration and it was simulated for every value of the ammonia concentration. The conclusion was that bubble through inter-cooler and the configuring with the heat exchanger between vapour and lean solution first when following the lean solution flow was the worst options to improve the COP of the heat pump. As mentioned above, the most efficient solution was the system showed in Figure 3.2, but the liquid injection has a lower investment cost and it is more efficient than the other two solution for almost every values of the ammonia concentration. The lowest discharges temperature was also achieved by the compressor arrangement shown in Figure 3.2 and the liquid injection from the rich solution had the second lowest discharges temperature.

3.2 Previous absorption/compression heat pump

After Osenbrück in 1895 introduce the Osenbrück cycle the where no one working on the HACHP before Altenkirch (1950) in the 1950s did a theoretical study on the subject. However, it was first in the 1970s a real effort was put into studying the HACHP system, by named as Zigler, Åhlby, Itard and Bruin (Nordtvedt, 2005).

Brunin et al. (1997) made a prototype of an HACHP to check if the HACHP system with ammonia water mixture could replace the vapour compression heat pump with CFCs and HCFCs fluid as a high temperature heat pump. To check if the system could be replaced some constrains was set to secure an economical and technologic possible solution. Some of the constrains is that the low pressure had to be higher than 1 bar and the high pressure should be lower than 20 bar. The economical limitation involves a COP on 4 and a VHC on 2 MJ/m^3. Brunin et al. (1997) conclude that the HACHP with a water ammonia mixture is the only heat pump, which can replace the vapour compression heat pump with CFCs and HCFCs fluids as a high temperature heat pump.

More recently, Nordtvedt (2005) investigated the different between a steady state mathematical model and experimental results. In the mathematical model made by Nordtvedt (2005), the heating of water was from 50 to 96 ^{o}C in the sink and the cooling of water in the sources was from 50 to 6 ^{0}C and it gave a COP on 3.41. While in the experimental setup it gave a COP on 2.47, when the water was heated from 50 to 95 ^{o}C and the water where cooled from 50 to 17 ^{o}C. Nordtvedt (2005) say that the different in the model and the test result was because of sub cooling of the lean solution before the absorber, larger losses in the compressor, losses in the electrical motor, bigger losses in the absorber and desorber than expected and pressure drop at the high-pressure side, because of resistances.

As mentioned in chapter 3.1 one of the limitations for an HACHP is the discharges temperatures of the compressor. Nekså et al. (1998) said that discharger temperatures under 180 ^{o}C should be achievable to maintain the lubricant in the compressor. Jensen (2015) shows with a maximum discharge's temperature of 170 ^{0}C that it is feasible to achieve a sink temperature out at 150 ^{0}C with a source's temperature in at 90 ^{0}C and a two-stage compression. Both Jensen (2015) and Brunin et al. (1997) analysis is based on economics and to decide the economical COP they use the net present value method. Jensen (2015) also showed HACHP is the best heat pump for sink temperatures above 80 ^{0}C considering the economical solution whit the same constrains as from the Jensen (2015) study above. Except from the discharged temperature the other main constrain in the study was the high pressure and the vapour ammonia mass fraction going through the compressor. Since this is only a numerical study the result is a bit more uncertain than with an experimental study as Nordtvedt (2005) showed.

Today some HACHP is in industrial use and Hybrid energy AS have develop some of the heat pumps. Hybrid energy AS promise they can deliver HACHP systems, which deliver temperatures up to 120 ^{0}C. So far, they have delivered a two stages compression HACHP system, which had a sink temperature out at 110 ^{0}C to a wastewater treatment plant in Norway with a COP at 2.4. There the inlet sources temperature was 20 ^{0}C and the inlet sink temperature were 75 ^{0}C. Another system they have made is to a company in Denmark, which deliver district heating with help of solar power. The COP on that system was 4.3 and the deliver temperature were 100 ^{0}C with an inlet sink and sources temperature at 35 ^{0}C (HybridEnergyAS, 2016).

As mentioned in chapter 2.1 the HACHP get an extra degree of freedom compared to a vapour compression heat pump, because it can vary the ammonia mass fraction in the absorber and desorber. So far, it has been some studies to optimise the different parameters and the focuses have been on the effect of changing the circulation ratio and how the changes are influencing the COP, temperature lift and the temperature out of the sink. The circulation ratio is given in equation 3.1 using Jensen (2015) approach, where the circulation ratio is the ratio between the mass flow rate of the lean solution ($\dot{m}_{liq}$) and the mass flow of the rich solution($\dot{m}_{rich}$). It is two different approach of the circulation ratio, where Nordtvedt (2005) using the ratio between the mass flow rate of the lean solution and the mass flow rate of vapour ($\dot{m}_{vapo}$) as seen in equation 3.2.

$$CR = \frac{\dot{m}_{liq}}{\dot{m}_{rich}} \tag{3.1}$$

$$CR = \frac{\dot{m}_{liq}}{\dot{m}_{vapo}} \tag{3.2}$$

Nordtvedt (2005) figured out through simulation that a circulation of 0.65 was the best considering the highest COP as shown in Figure 3.3, which correlate to an ammonia concentration of 0.73. However, the highest temperature lift was achieved with a circulation ratio at 0.95 and the highest temperature is achieved with a circulation ratio between 1.1 and 1.5 as seen in Figure 3.4. The COP values vary with 0.09 with a circulation ratio vary between 0.4-1.5 and the temperature lift in the same range of circulation ratio vary with 90 K. Jensen (2015) showed that the circulation ratio and the ammonia concentration affect the desorber pressure, but the absorber pressure is only affected by high circulations ratios and high ammonia concentration. Jensen (2015) also study how different circulation ratio and ammonia concentration affect the COP of the system. With a temperature glide in the heat sink and heat source at 30 K the COP was relative stable for ammonia concentration between 0.15-0.9. However, with a temperature glide at 10 K the ammonia concentration had an impact on the COP where the highest COP was with ammonia concentration at 0.1 or 0.9. Jensen (2015) conclude that the impact of the ammonia concentration and circulation ratio vary with the type of system and its operating conditions.

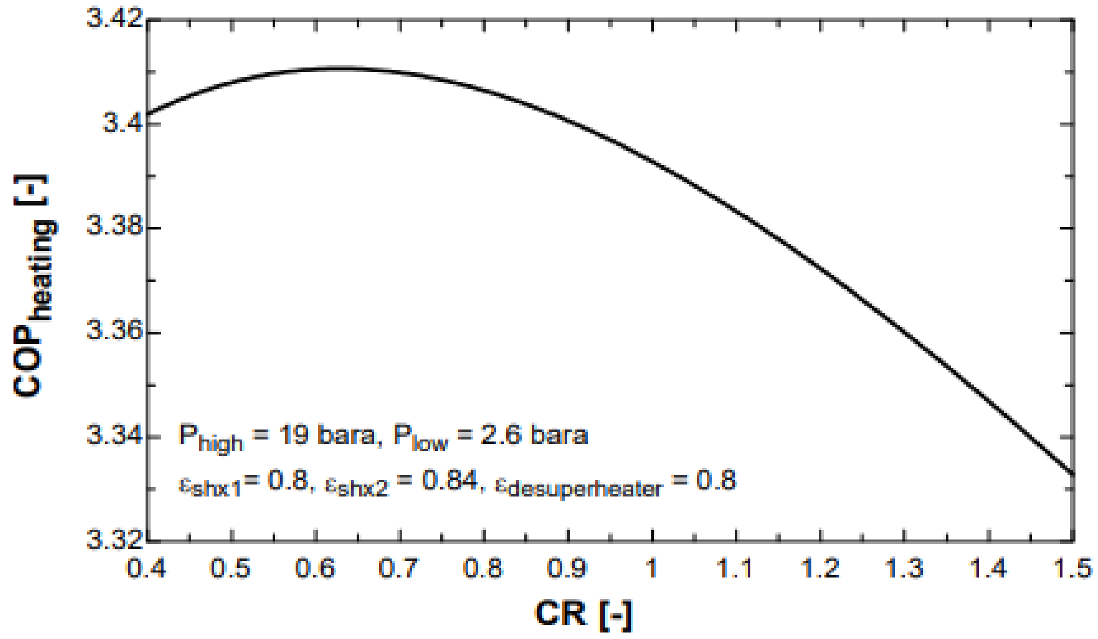

Figure 3.3: COP versus circulation ratio from Nordtvedt (2005) model

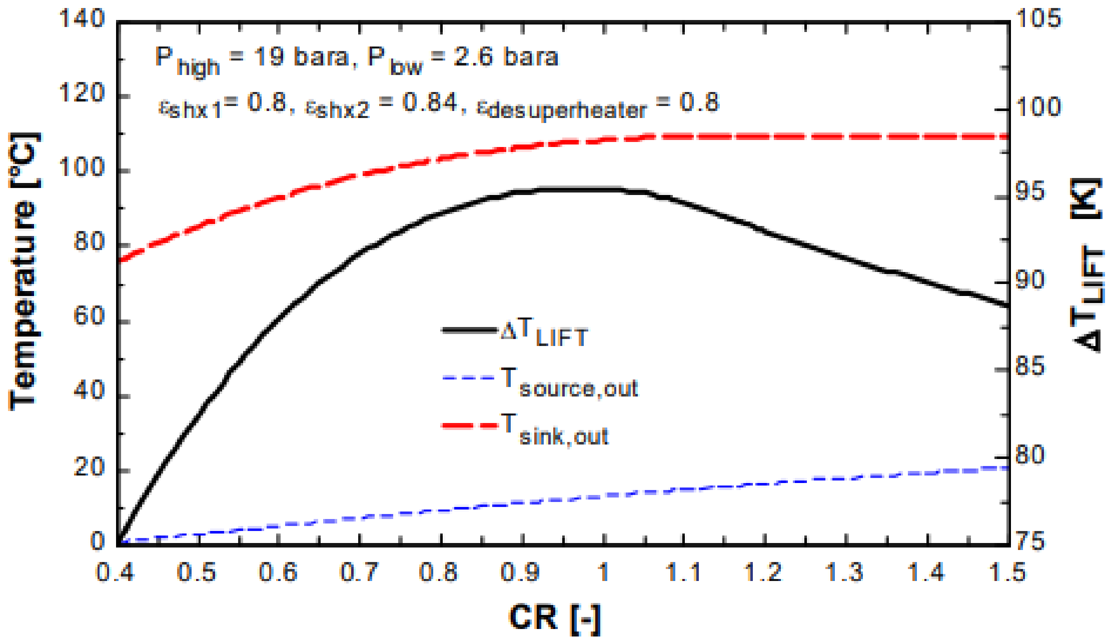

Figure 3.4: Temperature out of the sink, temperature lift, temperature out of source as function of circulation ratio for Nordtvedt (2005) model.

3.3 Absorbers

One of the most important component in the HACHP is the absorber and the absorber have a huge impact of the COP for the system (Jung et al., 2014). The absorber process in an

HACHP is more complex than the evaporation process in a conventional heat pump system, since the mass transfer as well as the heat transfer must be taken into consideration, when designing the absorber.

Bubble, falling film, adiabatic and membrane mode is four different modes that can be used in an absorbers to ensure mass and heat transfer (Ibarra-Bahena and Romero, 2014). Falling film and bubble mode is most used today in the HACHP and it have been a few experiment compering these two modes (Jung et al., 2014).

Lee et al. (2002) tested three different plate heat exchangers to compare the falling film mode and the bubble mode for an ammonia-water mixture at low pressure and with different flow rate for the vapour and the liquid. It was found that the bubble mode was much better when it comes to mass transfer performances and it generated more heat than the falling film mode. Kang et al. (2000) did a parametrical analysis of bubble and falling film mode in a plate heat exchanger to investigate the heat and mass transfer for the absorber in an absorption heat pump using ammonia-water mixture as a working fluid. From this research, the conclusion was that the bubble mode was the best to use in an absorber, since the local absorber rate for the bubble mode was always higher than for falling film mode. Because of this, the heat exchangers could be 48.7 % smaller with bubble mode. Another findings Kang et al. (2000) figured out is that the falling film mode have wettability problem and therefore need a great liquid distribution at the liquid flow inlet. The falling film mode could be seen in figure 3.6

One of the disadvantages with bubble mode compared with falling film mode is the pressure drop thought the absorber is higher, but the problem is biggest for low-pressure system. (Lee et al., 2002). Bubble mode need vapour distribution that is easier to achieve than liquid distribution, which liquid film use. In vapour, distributions the vapour is driven through a pool of liquid as seen in figure 3.5 and this require a pressure different on the vapour side (Lee et al., 2002).

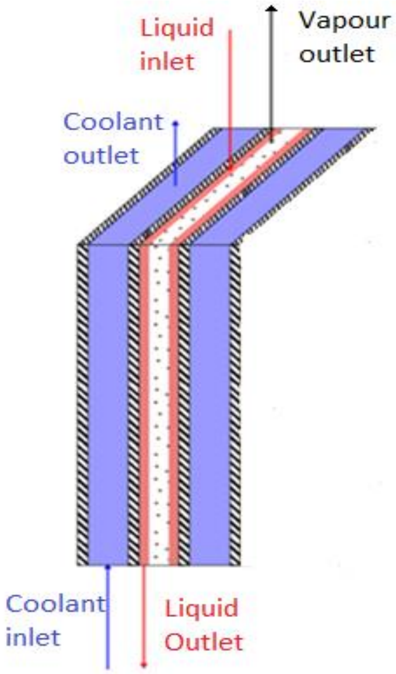

Figure 3.6: Falling film mode absorber in a plate heat exchanger (Triché et al., 2017)

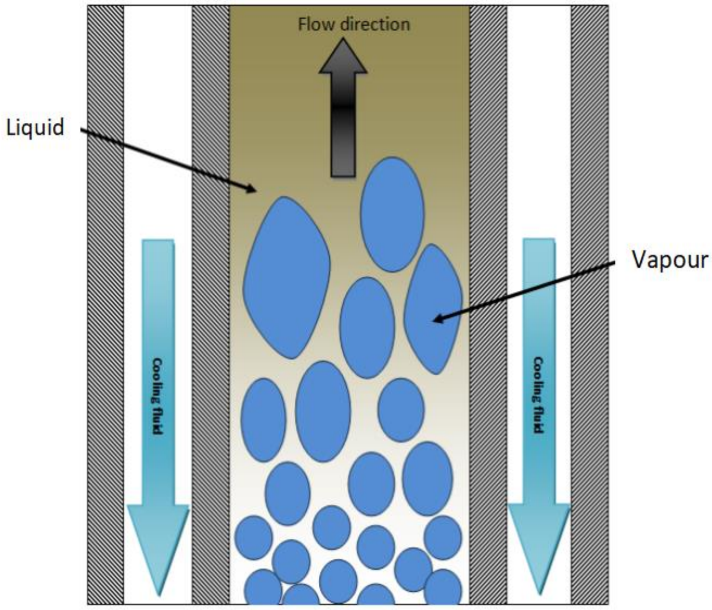

Figure 3.5: Bubble mode absorber in a plate heat exchanger (Ibarra-Bahena and Romero, 2014).

Moreover, the heat exchanger type has to be considered when choosing the absorber and the two main type that is used as an absorber is plate heat exchangers and Shell/tube heat exchangers. Nordtvedt (2005) compared different earlier absorption/compression cycles and figured that the most preferred heat exchanger type used as an absorber has been shell/tube heat exchangers. However, when the 21 century began more and more heat exchangers in the absorption/compression cycles where plate heat exchangers and Nordtvedt (2005) used a plate heat exchangers in the experimental setup. Plate heat exchanger is more compact, provide high heat transfer coefficient, easy to adjust the heat transfer and easy to maintain. The main disadvantages is that the plate heat exchanger cannot handle big temperature differences, pressure drops, and it have problem with handling very high pressure and temperatures (Lee et al., 2002). Alfa Laval have the later years developed some new plate heat exchangers, which can handle higher temperatures and today Alfa Laval have some heat exchangers that can handle temperatures up to 250 ^{0}C and pressure at 26.8 bar (AlfaLaval, 2016).

3.4 Liquid injection into compressor

As mentioned in chapter 3.1 the main issues with the compressor is the high discharges temperatures and in the early research of the screw compressors one way to solve the problem was to inject oil into the compressor. In Addison, the oil was used to seal the clearance's spaces between the rotors and used as a lubricant. The problem with injecting oil into the compressor was the large amount of oil that was needed and a big oil separator where required. In Addison some power was needed to pump the oil into the compressor (Xu et al., 2011). Therefore, to reduce the space and power consumption Moody and Hamilton (1975) suggested to inject liquid into the compressor instead of oil. This where possible now since the clearances volume had been smaller with new technology. The liquid injected into the compressor is not only for cooling, but also to lubricate and seal the clearances volume (Stosic et al., 2005). The liquid where taken from the condenser, so an additional pump where not needed. However, one major risk with injecting liquid into the compressor is slugging and it is biggest for a reciprocating compressor but could also occur in screw compressors. A higher amount of superheat will reduce the chances for slugging, so it is hard to estimate the maximum amount of liquid that could be injected into the compressor since it is dependent on a lot of parameters in the heat pump and type of compressor (Duncan, 1999). However, with high amount of superheat the compressor length and the compressor speed will also affect how much of the liquid that will evaporated inside the compressor and this will influence the compressor discharges temperature. Another disadvantages is that the compressor in itself is more expensive than a compressor without injection, since it need an injection port, but it only need one compressor compared with a two stages system with two compressors (Lee et al., 2015). Moreover, Zaytsev (2003) investigated the isentropic efficiency for different compressor and figured out that the screw compressor normally achieves an isentropic efficiency from 0.5 to 0.8, while the reciprocating compressor have an isentropic efficiency between 0.6 to 0.85.

Another problem with injecting liquid into the compressor is that the efficiency of the compressor falls with increased liquid content in the compressor. Bakken et al. (2018) showed that the polytropic efficiency decreased with a bigger amount of liquid injected into the inlet of a centrifugal compressor. With 10% mass fraction of liquid into the compressor the polytopic efficiency decrease around 10 % compared with a liquid mass fraction at 0%.

Lee et al. (2015) study experimentally the different between liquid and vapour injection for a refrigeration system with high pressure ratio. Both the injection medium and working medium is R22 and the compressor is a scroll compressor which lift the pressure from around 4.8 bar to 24 bar and had the injection in front of the compressor. The maximum COP for the liquid injection system was achieved at 10% injection ratio as seen in Figure 3.7 and the discharges temperature almost decreased linearly with increasing the injection ratio. Moreover, the injection with liquid is superior to the vapour injection when it comes to low discharges temperatures.

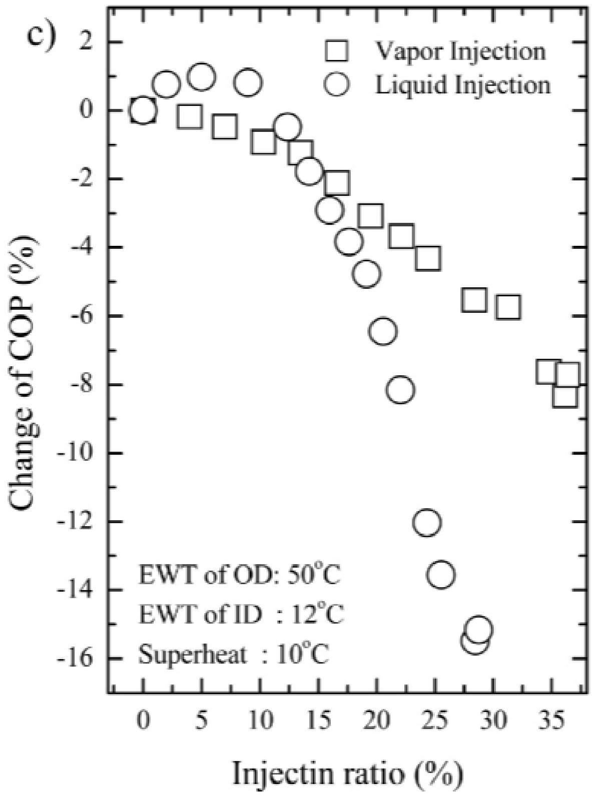

Figure 3.7: Changes of COP as a function injection ratio (Lee et al., 2015)

To sum up chapter 3, the same parameters could be affected differently dependent on the type of system and the operating condition. The biggest limitation with HACHP is the absorber pressure and the discharged temperature, but new and better components in the market could remove or reduces this limitation.

4 Simulation model

4.1 General

To find the optimal operational condition, to figured out which compressor arrangement that suits the test rig best and to make a simulation tool, five simulation models where made in EES. EES is an equation solving simulation tool, which solving non-linear equation. In Addition, EES has an inbuilt procedure giving thermodynamic properties for different working fluid, which is build-up on Ibrahim and Klein (1993) correlations. For zeotropic working fluid three out of eight properties, which is possible to get out of the procedure, must be given. The code to get the properties for ammonia/water is:
CALL NH3H2O(XYZ;Inx;Iny;Inz:T;P;x;h;s;u;v;q). XYZ is three number that correlated to which input parameter that need to be used. In respectively order, the parameters with their units is (FChartSoftware, 2019):

T= Temperature [K]

P= Pressure [bar]

x= Ammonia mass fraction [-]

h= Enthalpy [kJ/kg]

s= Entropy [kJ/kg-K]

u= Internal energy [kJ/kg]

v= Specific volume [m^3/kg]

q= Vapour quality [-]

All the models include a desorber, compressor, desuperheater, absorber, internal exchanger, expansion valve and a solution pump. A schematic diagram of the model with injection from the lean solution can be seen in Figure 4.1, while the rest of the models can be seen in appendix C. As seen from Figure 4.1 the rich solution is absorbing heat from the sources in the desorber and mostly ammonia is starting to evaporate from the ammonia-water mixture, which entering the separator after the desorber. From the separator the vapour entering the compressor that compress the vapour to a higher pressure, while the lean liquid is fed into the pump to elevate the pressure. After the pump the lean solution splits into two streams there one is going through an expansion valve to adjust the pressure before it is injected into the compressor to reduce the temperature inside the compressor. The other liquid stream is going through a heat exchanger there it absorbs heat from the rich solution, before it mixes at the inlet of the absorber with the vapour that have been cooled down by the desuperheater. In the absorber, heat is rejected to the sink while the liquid is absorbing the vapour. After the absorber the rich solution is entering the heat exchanger, which is giving heat to the lean solution, while cooling down the rich solution. After the heat exchanger the rich solution is entering the expansion valve there the pressure and temperature drops. Again, the rich solution is ready to absorb heat from the source in the desorber.

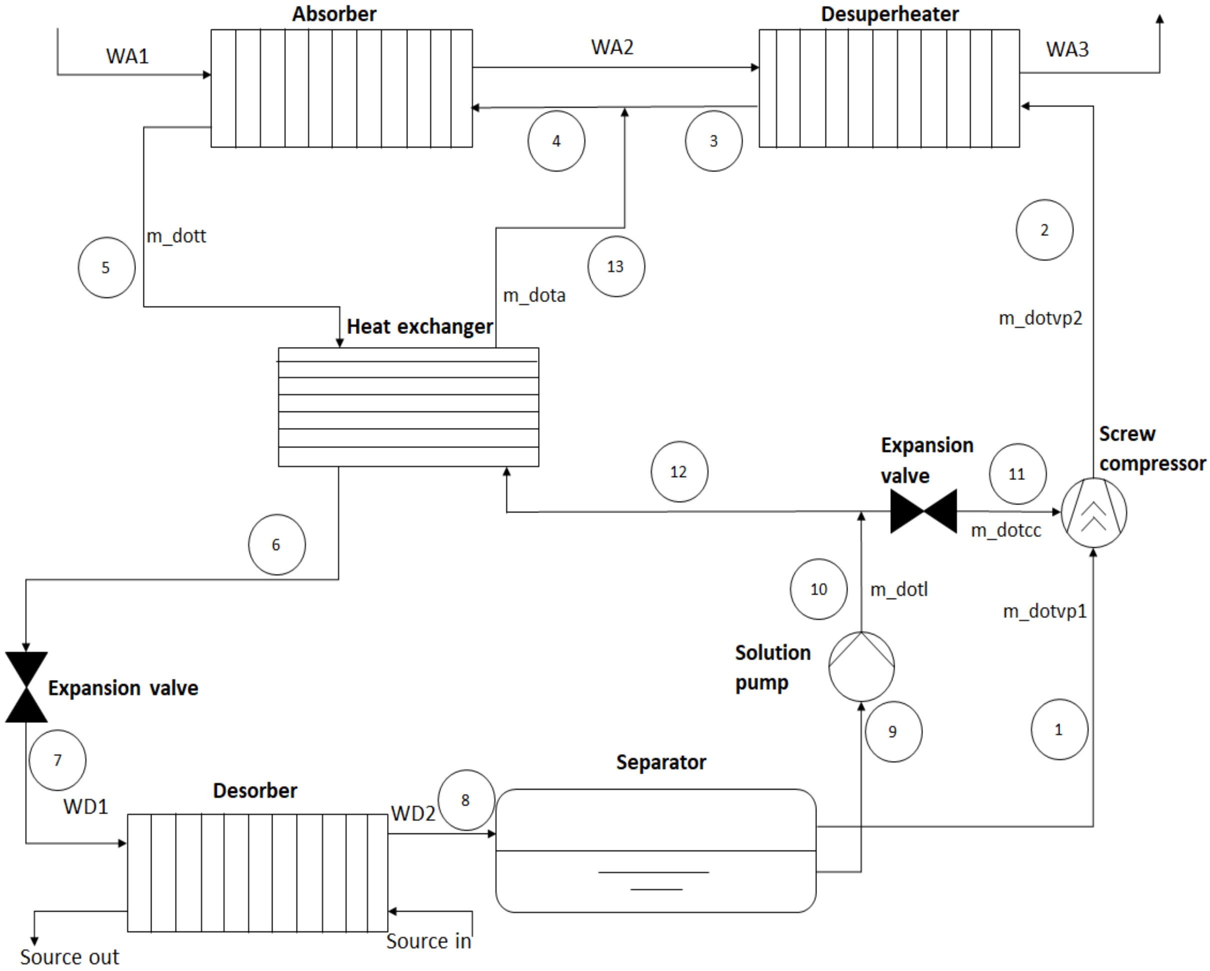

Figure 4.1: Schematic diagram of the simulation model with injection from lean solution. The names on the differences flows in the figure is the same names that is used in the script.

As seen from appendix C the different between the injection from lean solution and injection from the rich solution is that the injection line is from the rich solution after the heat exchanger. Also, here the pressure is adjusted with an expansion vale to the intermediate pressure before injection. Moreover, comparing the injection from lean solution and IHEX option 1, the IHEX option 1 have replaced the screw compressor with two reciprocating compressor and removed the injection line to the compressor. In Addison, IHEX option 1 have added one extra heat exchanger between the compressors in the vapour circuit and lean solution after the other heat exchanger when following the lean solution flow. The different between IHEX option 1 and 2 is that the heat exchanger have changes order, so the heat exchangers between the vapour and lean solution is before the other when follow the lean solution flow. Last but not a least, the different between the IHEX option 1 and one-stages is that the heat exchanger between the vapour and lean solution is removed, so it is almost like the Osenbruck cycle in Figure 2.4.

The calculation is based on thermodynamics equations and equation 3.1 is used to calculate the circulation ratio using Jensen (2015) approach. The ammonia mass fractions in the rich solution is calculate from the mass balances equation showed in equation 4.1 and the mass flow rate in the vapour, rich and lean solution is calculated from the mass balances equation 4.2. In addition, the energy equations 4.3 and 4.4 is utilized to calculate the thermodynamics properties in the models. The calculation method can be seen in appendix D. To find the optimal operating condition for the models and which compressor arrangement that suits the test rig best, some input parameters must be given, and some outputs parameters have to be evaluated. Inputs and outputs parameter for the calculation is listed in Table 4.1, but in some calculation the vapour mass flow rate is given as an input parameter then the heat transfer rate to the sink is an output parameter.

$$\sum(\dot{m} * x)_{in} = \sum(\dot{m} * x)_{out} \tag{4.1}$$

$$\sum \dot{m}_{in} = \dot{m}_{out} \tag{4.2}$$

$$\sum(\dot{m} * h)_{in} = \sum(\dot{m} * h)_{out} \tag{4.3}$$

$$\sum(\dot{m} * h)_{in} + \dot{W} = \sum(\dot{m} * h)_{out} \tag{4.4}$$

Table 4.1: Model inputs and outputs parameter

Inputs	Outputs
▪ Sink and sources pressure ▪ Sink and Sources inlet temperature ▪ Heat transfer rate to the sink ▪ Circulation ratio ▪ Absorber and desorber Pressure ▪ Minimum temperature difference in absorber and desorber ▪ Efficiency of pump, heat exchanger and compressor. ▪ Correction factor ▪ Injection ratio*	▪ Sink, sources, rich, vapour and lean solution mass flow rates ▪ System performance ▪ Thermodynamics state points ▪ Temperature out of the sink and sources ▪ Heat transfer rate from the absorber, desuperheater and internal heat exchanger ▪ Compressor and pump work ▪ Efficiency of desuperheater

*Only for the models with liquid injection.

To simplify the calculation and to avoid a too complex model some assumption has been made. The main assumption, which has been made is:

- Friction in the system is neglected, so the pressure drop due to losses in pipes and heat exchangers is assumed to be zero.
- The solution pump is assumed to be a perfect one, so the isentropic efficiency is 100%.
- After the absorber all the vapour have condensed, and the rich solution is at the saturation temperature.
- Mixing of the lean solution and the vapour is assumed to be adiabatic, so no heat exchanges with the surroundings.
- Vapour and liquid in both desorber and absorber are assumed to be in thermodynamic equilibrium.
- Other heat losses to the ambient is also neglect.
- The liquid in the separator is in thermodynamic equilibrium with the vapour at the inlet of the compressor.

- All the liquid injected into the compressor is assumed to be for cooling of the vapour and at the intermediate stages in the compressor the vapour and the injected liquid is assumed to be in thermodynamic equilibrium.

4.2 Compressor

The compressors use the isentropic efficiency to find the enthalpy out of the compressor there equation 4.5 where used and $h_{out,isen}$ is the enthalpy out of the compressor for an isentropic process. Nordtvedt (2005) used equation 4.7 to calculate the isentropic efficiency for the reciprocating compressor. The isentropic efficiency is often given from the compressor manufactures, so to use the model as a simulation tool equation 4.7 must be suited to the compressor that are selected. The isentropic efficiency for the screw compressor is given in equation 4.8 and to calculate the work out of the compressor the motor efficiency of the compressor was set to 0.9. For both compressors type the volumetric efficiency is given by equation 4.10 and for all models except the one-stages, equation 4.6 is used to determine the intermediate pressure, where the correction factor must be given. For the liquid injection compressors equation 4.9 is used to determine the mass flow rate of liquid injected into the compressor and W is the injection ratio, which is the Percent of liquid injected into the compressor compared with mass flow of vapour entering the compressor. The intermediate pressure for the liquid injection compressors is the same as the injection pressure and the liquid injection model first compress the vapour up to the intermediate pressure there the mixing occurs before it compresses the mix up to the absorber pressure. In both compression stages it corrects for the isentropic efficiency with the pressure as the ratio between the absorber pressure and desorber pressure

$$h_{out} = \frac{\left(h_{out,isen} - h_{in}\right) + (\eta_{isen} * h_{in})}{\eta_{isen}} \tag{4.5}$$

$$P_{MP} = K * \sqrt{P_{LP} * P_{HP}} \tag{4.6}$$

$$\eta_{isen,rec} = 0.9051 - 0.0422 * PR \tag{4.7}$$

$$\eta_{isen,screw} = 0.9051 - 0.0222 * PR \tag{4.8}$$

$$\dot{m}_{Innj} = \dot{m}_{vp1} * W \tag{4.9}$$

$$\eta_{volu} = 1.0539 - 0.0788 * PR \tag{4.10}$$

4.3 Absorber

The absorber is modeled as a countercurrent heat exchanger and use the energy equation as seen in equation 4.3. To ensure heat transfer from the mixture to the sink through the whole absorber a minimum temperature difference between the two fluids is fixed. Therefore, the absorber is divided into 50 segments, where the energy equation is applied on each part. To regulate the temperature differences inside the absorber the mass flow rate in the sink is adjust, so if the temperature difference is too small the mass flow rate will increase. As mentioned, it is assumed that all the vapour has condensed at the exit of the absorber, so the vapour quality at the outlet is equal to zero.

4.4 Desorber

The main differences between modelling the absorber and desorber is that the minimum temperature differences in the desorber for an ammonia/water mixture is located at the outlet or the inlet of the heat exchanger, while in the absorber it is located somewhere in the middle of the heat exchanger (Nordtvedt, 2005). The temperature glide for the rich solution and the sources is set to be the same through the desorber with adjusting the mass flow rate in the sources. Equation 4.11 is used to calculate the outlet temperature for the rich solution, which is the differences between the inlet temperature of the sources and the minimum temperature differences. To calculate the outlet temperature of the sources equation 4.12 is used where $\mathrm{T}_{WD,in}$ is the inlet temperature for the rich solution and to find the sources mass flow rate equation 4.3 is applied.

$$T_{WD,out} = \mathrm{T}_{Des,in} - \Delta T_{Deso} \tag{4.11}$$

$$T_{Des,out} = \mathrm{T}_{WD,in} + \Delta T_{Deso} \tag{4.12}$$

4.5 Single-phase heat exchanger

In the single-phase heat exchanger model the flows are countercurrent. Since EES do not provide a procedure for the specific heat capacity, equation 4.13 is used to calculate it. To calculate the heat transfer rate, equation 4.15 is used and the minimum heat capacitance rate is calculated from equation 4.14, while the efficiency is calculated from equation 4.16. Both the internal heat exchangers and the desuperheater is calculated as a single-phase heat exchanger, but the desuperheater is not utilize in the simulation except the simulation there the results are compared with Nordtvedt (2005) results. The reason that the desuperheater is not utilized are that Bergland (2015) concluded the effect of the desuperheater was small, but it is included in the script as it can be utilized in the simulation tool for the test rig.

$$C_p = \frac{\Delta h}{\Delta T} \tag{4.13}$$

$$\dot{C} = \dot{m} * C_p \tag{4.14}$$

$$\dot{Q} = \varepsilon * \dot{C}_{min} * (T_{ho,in} - T_{co,in}) \tag{4.15}$$

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}} \tag{4.16}$$

4.6 Pump

The pump lifts the pressure in the liquid isentropic from a low pressure to a higher pressure and the only thing in the pump that is not assumed to be perfect is the motor efficiency, which was set to 0.9.

4.7 Expansion valve

In the simulation model, the expansion valve is an isenthalpic process, which expand the fluid, reduces its pressure and temperature, which leads the fluid into a vapour/liquid mixture.

4.8 Separator

The separator in this model, separates the rich solution in to vapour and the lean solution. In this model, it is assumed that the lean solution consists of pure saturated liquid after the separator and the vapour only consist of pure saturated vapour.

4.9 System performances

To determine how efficient the heat pump system is both the COP heating and Lorenz COP is calculated from equations 4.18 and 4.21. The Lorenz COP is the same as the Carnot COP for a vapour compression heat pump, which is the COP for a reversible cycle and the second law efficiency is calculated from equation 4.23, which is a good measure of the efficiency of the system (Cengel and Boles, 2015).

$$\dot{Q}_{abs} = (h_4 - h_5) * \dot{m}_{rich} \tag{4.17}$$

$$COP_{heating} = \frac{\dot{Q}_{dsh} + \dot{Q}_{abs}}{\dot{W}_{pump} + \dot{W}_{comp}} \tag{4.18}$$

$$\dot{W}_{comp} = \frac{(h_{out} - h_{in}) * \dot{m}_{vapo}}{\eta_{comp}} \tag{4.19}$$

$$\dot{W}_{pump} = \frac{(h_{out} - h_{in}) * \dot{m}_{liq}}{\eta_{pump}} \tag{4.20}$$

$$COP_{Lorenz} = \frac{\bar{T}_{LM,sink}}{\bar{T}_{LM,sink} - \bar{T}_{LM,source}} \tag{4.21}$$

$$\bar{T}_{LM} = \frac{T_{in} - T_{out}}{\ln\left(\frac{T_{in}}{T_{out}}\right)} \tag{4.22}$$

$$\eta_{II} = \frac{COP_{heating}}{COP_{Lorenz}} \tag{4.23}$$

5 Simulation tool

The simulation tool is based on the schematic drawing in Figure 4.1 and the simulation code could be seen in appendix D. In Table 4.1 the input values and outputs value of the simulation tool are listed, but some of them is possible to changes. For example, could the heat transfer rate to the sink be an output parameter, if one of the mass flow rates in the solution circuit instead are given. When the compressor type is determined it is recommended to set the mass flow rate at the inlet of the compressor as an input parameter based on equation 5.1. There $\dot{V}_{swpt}$ is the swept volume and η_{volu} is the volumetric efficiency, both are often given by the compressor manufactures and $v_{in}$is the specific volume. As mentioned in chapter 4.2 the isentropic efficiency in equation 4.8 also need to be changes based on, which compressor is chosen. In Addison, could the absorber pressure be calculated by equation 5.2, which is the optimal pressure ratio for a screw compressor. The isentropic exponent (γ) is set to 1.36 (EngineeringToolBox, 2018), which is the isentropic exponent for pure ammonia, since the flow in the compressor consist of almost pure ammonia.

$$\dot{m}_{vp1} = \frac{\dot{V}_{swpt} * \eta_{volu}}{v_{in}} \tag{5.1}$$

$$\frac{P_{out}}{P_{in}} = \left(\frac{V_{in}}{V_{out}}\right)^{\gamma} \tag{5.2}$$

All the input values are given under “input parameters” in the script, except the temperature after desuperheater (T[3]), which can be determined under “Loop” and section three, there it must be decide how much the vapour is going to be cooled down. This could be used to check the effect of the desuperheater for the system. The input parameters must be given in the SI unit system with this units:

- Mass: kg
- Temperature: Kelvin
- Pressure: bar
- Energy: kJ

It is two ways to adjust the accuracy of the simulation and both is in the absorber. The first is to increase the number of segment that the absorber is divided into and the second one is to reduce the amount that the mass flow rate in the sink is adjusted with when the differences in the absorber is not big enough. For a more detail description on how the different component in the simulation tool is modelled see chapter 4.

If the simulation doing above 250 iteration it will not solve the problem, so a solution to this could be to give some of the values an initial guess. The way to do this is under the option button, click on variable info button and then choose a parameter to give a guess value. If it is still some problem set the lower limitation for all the mass flow rate to zero and the upper and lower limitation for the ammonia mass fraction respectively to 1 and 0. This is usually not a problem for the simulation tool, but another problem that could occur is that the simulation will not solved and in the calculations progress box with red letters it will be written absorber as seen in Figure 5.1. The way to solved this is to either increase the absorber pressure or decrease the inlet sink temperature. Because as mentioned in chapter 4.1 the solution leaving the absorber is at saturation temperature for the given pressure and ammonia concentration. Therefore, if the differences between the saturation temperature and the inlet sink temperature is less than the minimum temperature in the absorber the script cannot solved the problem.

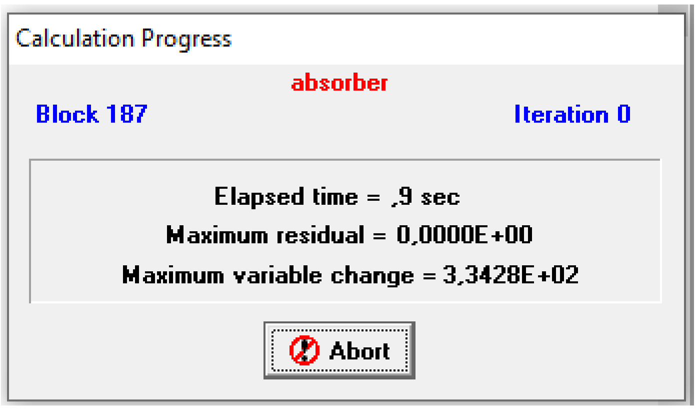

Figure 5.1: Calculation in progress problem with the simulation tool

6 Result and discussion

6.1 Heat pump configuration

As mentioned one of the main scope for the task was to decide the compressor arrangement that is going to be used as a test rig. Therefore, different heat pump configuration has been tested with different input parameter in four cases.

6.1.1 Simulation setup and results

Case1

The aim for case 1 was to evaluate the different configuration with a compressor discharged temperature at 170 ^{0}C. The input parameters for this case is listed in Table 6.1.

Table 6.1: Input parameter for case 1.

Parameter	**Value**
Sink pressure	10 bar
Sources pressure	1.6 bar
Efficiency of internal heat exchangers	0.9
Efficiency of desorber	0.9
Compressor and pump motor efficiency	0.9
Minimum temperature difference in absorber	5K
Minimum temperature difference in desorber	5K
Heat transfer rate to the sink	250 kW
Sources inlet temperature	50 ^{0}C
Sink inlet temperature	50 ^{0}C
Injection ratio*	0.1
Desorber pressure	2.2 bar
Circulation ratio	0.5
Compressor discharges temperature**	170 ^{0}C

*This is just for the cases with liquid injection to the compressor.
**It is not possible to achieve a discharged temperature at 170 ^{0}C for the one-stages model (see chapter 5). The discharged temperature for this case were at 271.05 ^{0}C

There are several parameters, which must be evaluated to decide the type of compressor arrangement and the most important parameters is the COP heating, compressor discharged temperature and temperature out of the sink. In Addison to the important parameters some other parameters that are affecting those important parameters are listed in Table 6.2 for case 1. In appendix B several other results for the four cases are given.

Table 6.2: Result case 1

Parameters	One-stages	Injection lean	Injection rich	IHEX option 1	IHEX option 2
$COP_{heating}$	3.246	3.293	3.527	3.237	2.446
η_{II}	0.2656	0.4489	0.4250	0.4798	0.3795
Temperature out of the sink [^{0}C]	62.65	106.55	92.85	117.85	131.85
Absorber pressure [bar]	13.5	16.45	14.81	19.51	29.47
Isentropic efficiency*	0.6461	0.7391	0.7557	0.7652 0.7922	0.7669 0.7325

*For the two stages compression the isentropic efficiency for the first compression is listed first in the table and the second compression last.

From Table 6.2 with a maximum compressor discharged temperature at 170 ^{0}C the highest heat sink temperature were 131.9 ^{0}C obtained with the IHEX option 2, while the one-stages had the lowest heat sink temperature. The compressor arrangements with injection from the rich solution had the highest COP heating, but the second lowest temperature out of the sink. IHEX option 1 had the highest second law efficiency, while the injection from lean solution had the second highest second law efficiency and COP heating with temperature out of the sink at 106.55 ^{0}C. Moreover, the IHEX option 2 had without a doubt the highest absorber pressure with 29.5 bar.

Case 2

Case 2 is going to evaluate the COP heating and the compressor discharged temperature when the temperature out of the sink is the same for all the compressor arrangement. Therefore, the main difference between case 1 and case 2 is that the temperature out of the sink is set to 100 ^{0}C and the compressor discharged temperature is not fixed for case 2. The result from this case can be seen in Table 6.3.

Table 6.3: Result case 2

Parameters	One-stages	Injection lean	Injection rich	IHEX option 1	IHEX option 2
$COP_{heating}$	2.955	3.453	3.359	3.751	3.665
η_{II}	0.3815	0.4452	0.4325	0.4834	0.4655
Compressor discharged Temperature [^{0}C]	295.45	160.05	180.15	147.25	134.15
Absorber pressure [bar]	15.02	15.46	15.76	15.65	16.02
Isentropic efficiency*	0.6171	0.7491	0.7561	0.7892 0.7958	0.8038 0.7771

*For the two stages compression the isentropic efficiency for the first compression is listed first in the table and the second compression last.

In case 2 as in case 1 the highest second law efficiency was obtained by the IHEX option 1 compressor arrangement and in case 2 it also had the highest COP. However, the differences in COP between the different compressor arrangement is small compared to case 1 except the one-stages configuration. The lowest compressor discharged temperatures were achieved by the IHEX option 2, IHEX option 1 and injection from lean solution with respectively temperatures at 134.2 ^{0}C, 147.3 ^{0}C and 160.1 ^{0}C. As seen from Table 6.3 the differences in the absorber pressure is just 1 bar between the highest and lowest pressure.

Case 3

To evaluated how the circulation ratio is affecting the different compressor arrangement, case 3 is made. In this case the input parameters are the same as for case 1 except the circulation ratio, which is set to 0.6 and the result is enumerate in Table 6.4

Table 6.4: Result case 3

Parameters	One-stages**	Injection lean	Injection rich	IHEX option 1	IHEX option 2
$COP_{heating}$	3.245	3.279	3.469	3.398	2.198
η_{II}	0.3926	0.4197	0.4212	0.4411	0.2974
Temperature out of the sink [^{0}C]	102.65	109.45	103.65	111.45	128.05
Absorber pressure [bar]	13.5	16.45	15.13	17.5	29.47
Isentropic efficiency*	0.6461	0.7391	0.7524	0.7679 0.8019	0.7669 0.7325

*For the two stages compression the isentropic efficiency for the first compression is listed first in the table and the second compression last.
**The discharged temperature for the one-stages compressor arrangement is 271 ^{0}C

As seen from Table 6.2 and Table 6.4 the increase in circulation ratio led to a decrease in the COP heating for the compressors with injection and IHEX option 2, while the COP heating for the IHEX option 1 increased. The sink temperature increased for the compressors with injection but decreased for the compressor arrangements with two internal heat exchangers. However, the temperature is still highest for the compressor arrangement with two internal heat exchangers.

Case 4

This case is going to evaluate the important parameters when the compressor discharged temperature and the injection ratio are increased. The input parameters that is different from case 1 is listed in Table 6.5.

Table 6.5: Input parameters for case 4 that is changes from case 1

Parameter	**New value**	**Old value**
Heat transfer rate to the sink	100 kW	250 kW
Injection ratio**	0.2	0.1
Desorber pressure	3 bar	2.2bar
Circulation ratio	0.4	0.5
Compressor discharges temperature*	220 ^{0}C	170 ^{0}C

* The discharged temperature for the one-stages compressor arrangement is 265 ^{0}C
** Only applied by the compressor with injection

Table 6.6: Result case 4

Parameters	One-stages	Injection lean	Injection rich	IHEX option 1	IHEX option 2
$COP_{heating}$	3.123	1.980	2.226	2.358	1.507
η_{II}	0.3601	0.3738	0.3810	0.4295	0.2902
Temperature out of the sink [^{0}C]	86.65	157.05	138.75	150.3	179.25
Absorber pressure [bar]	18.00	40.94	34.57	39.90	78.4
Isentropic efficiency*	0.6519	0.6021	0.6493	0.7127 0.7820	0.7109 0.6654

* For the two stages compression the isentropic efficiency for the first compression is listed first in the table and the second compression last.

From Table 6.6 the COP heating and the second law efficiency is highest for the IHEX option 1, if looking away from the one-stages compressor. However, in this case the injection from lean solution achieve higher temperature out of the sink than IHEX option 1, while the highest temperature is achieved by IHEX option 2 with a temperature at 179.3 ^{0}C. Another think to pay attention to, is the absorber pressure for the IHEX option 1 is a lot higher than for the other configurations.

6.1.2 Summary and discussion

As seen from the four different cases the injection from rich solution have the highest COP for case 1 and case 3, while the IHEX option 1 have the highest COP for the rest of the cases. However, the different in COP between IHEX option 1, injection from lean solution and injection from rich solution have a maximum difference at 0.4 for all cases. IHEX option 2 have the lowest COP for all cases except case 2, if the one-stages is not considered. The main reason for this is that the temperature between the compressor is cooled down more for the IHEX option 2 than the other configuration, so to lift the discharges temperature to the same value as the other compressor the pressure ratio must be bigger and with a higher-pressure ratio the compressor work gets bigger. As seen from case 1,3 and 4 the absorber pressure is higher for the IHEX option 2 than the other configuration and the desorber pressure is the same for all, so the pressure ratio is bigger for IHEX option 2. This is also the same reason why the Injection from rich solution achieved higher COP than the injection form lean solution in all the cases except case 2.

On the other hand, IHEX option 2 have the highest temperature out of the sink except from case 2 there it has the lowest discharges temperature. For case 1 and case 3 the IHEX option 1 have the second highest temperature out of the sink and injection from the rich solution have the lowest temperature, if the one-stages is not considered. The second highest sink temperature for case 4 is achieved by injection from lean solution. The main reason for the options with two IHEX achieves higher temperatures is that the lean solution gets heated by the vapour in addition to the cooling effect in the vapour. Moreover, the reason for IHEX option 2 achieve higher sink temperature with the same compressor discharges temperatures than IHEX option 1 is that the cooling effect on the vapour is bigger for IHEX option 2. Because the lean solution for IHEX option 1 is heated by the rich solution before cooling down the vapour. Therefore, IHEX option 1 have a higher enthalpy in the lean solution when the heat transfer between the vapour and lean solution occur, so the vapour in IHEX option 2 is colder between the compressors and therefore can achieve higher pressure and temperatures for the same discharge’s temperatures. This is the same principle as the injection from lean solution get higher sink temperatures for the same discharge temperatures than injection from the rich solution, since the lean solution consist of higher water content than the rich solution and water have a higher heat capacity than ammonia. In Addison, the injection temperature between the two configurations is not big. Therefore, the injection from lean solution gives a higher cooling effect. One thing to be aware of if choosing IHEX option 2 is that the cooling effect is not too big, so the vapour will condensate before the second compressor stages and as mentioned in chapter 3.4 the chance for slugging is bigger for a reciprocating compressor than a screw compressor.

When considering which compressor arrangement is going to be chosen price is an important factor to evaluate. As mentioned in chapter 3.1 the liquid injection has a lower investment cost than having two compressors and two IHEX. In Addison, it reduces the need of spaces, since it only need one IHEX and one compressor, but the compressor is often more expensive than a compressor without a liquid port. Mayekawa have come with an offer on a MYCOM 125 L screw compressor with a volume ratio at 3.65 or 5.80 and with liquid injection (MAYEKAWA, 2009). Mayekawa recommended that the liquid injected into the compressor consist of more water than ammonia, so injection from the lean solution suites this compressor better than injection from rich solution, since lean solution consist of more water.

As seen from Table 6.6 the injection from lean solution achieved higher temperature than IHEX option 1 and the main reason for this is that the injection flow into the compressor is increased, so the compressor discharges temperature will be lower for the same pressure ratio. The biggest problem with increasing the injection flow into to the compressor is that the efficiency of compressor will be lower as mentioned in chapter 3.4. Another problem will be to evaporate all the liquid before the end of the compressor, if not all the liquid is evaporated this will lead to a higher discharges temperature than assumed from the simulation model. This will be described in more detail in chapter 6.2.2.

As seen from Table 6.3 it looks like the absorber pressure have the biggest effect on the temperature out of the sink, but the small differences in absorber pressure also show that the discharges temperature have a small effect on the temperature as Bjørvik (2018) also concluded with. From Table 6.2 and Table 6.4 a change in the circulation ratio lead to a change in the COP heating and the temperature out of the sink for all the compressor arrangement, so to optimize the different compressor arrangement most of the input parameter should be evaluated for each special heat pump with their operational condition.

All in all, when considering price, spaces, performance of the heat pump and interest value of research, it seems like the injection from the lean solution is the best option for this test rig with good COP heating as well as a relative high temperature out of the sink. In Addison, lower investment cost and the require space to set up the test rig is small. The injection from lean solution have not been studying much, so the interest value of research is high.

6.2 Optimal operational condition

To optimize the test rig most of the input parameters must be evaluated and in this chapter some of the input parameter mentioned in Table 4.1 are going to be evaluated with the injection from lean solution simulation model.

6.2.1 Simulation setup and results

The first parameter that is going to be evaluated is the injection ratio and the input parameters which is used in this simulation is showed in Table 6.7. They are customized to the MYCOM 125 L screw compressor with a volume ratio at 3.65 from Mayekawa and the specification for the compressor could be seen in appendix F. From equation 5.2 the optimal pressure ratio for a compressor with VI at 3.65 is 5.82

Table 6.7: Input parameter for the simulation of optimal operational condition

Parameter	**Value**
Sink pressure	10 bar
Sources pressure	1.6 bar
Efficiency of internal heat exchanger	0.9
Efficiency of desorber	0.9
Compressor and pump motor efficiency	0.9
Minimum temperature difference in absorber	5K
Minimum temperature difference in desorber	5K
Mass flow rate at inlet to the compressor	0.132 kg/s
Sources inlet temperature	50 ^{0}C
Sink inlet temperature	50 ^{0}C
Desorber pressure	3 bar
Absorber pressure	17.5 bar
K	1
Circulation ratio	0.5

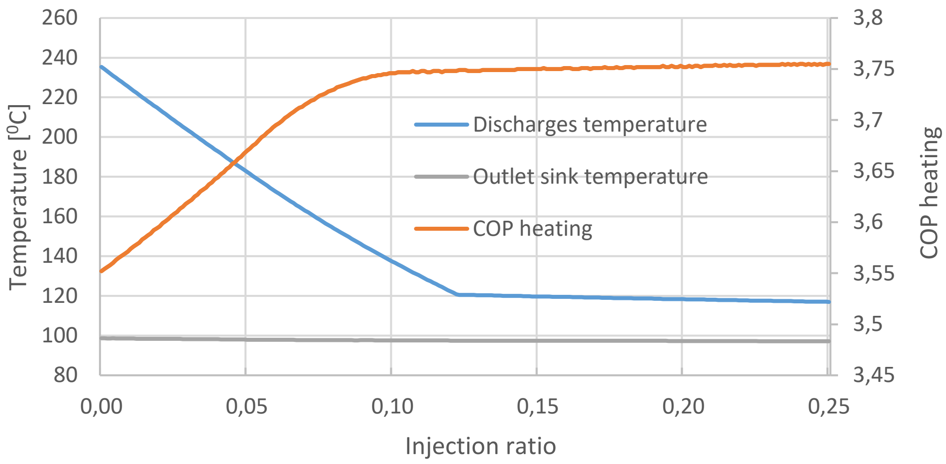

Figure 6.1: Compressor discharges temperature, outlet sink temperature, COP as a function of injection ratio for the VI 3.65 compressor

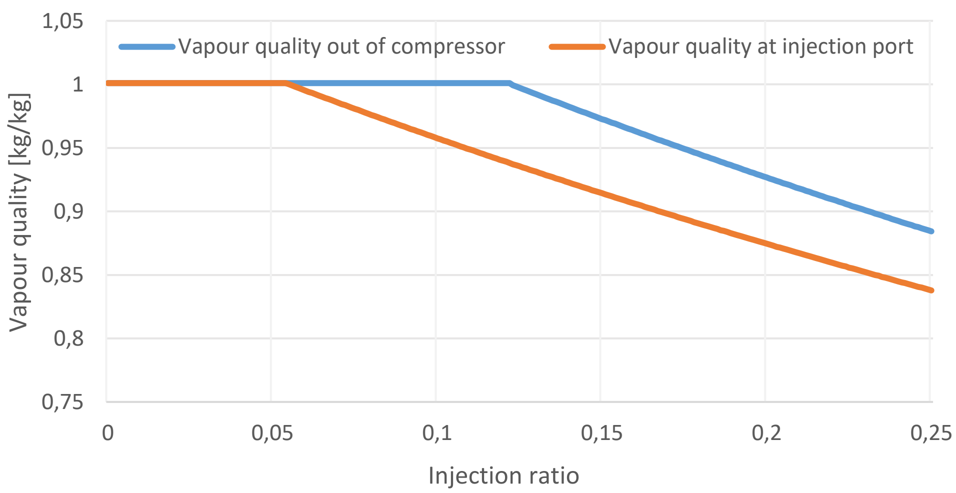

Figure 6.2: Vapour quality out of compressor and at the injection stages as a function of injection ratio for the VI 3.65 compressor

From Figure 6.1 the cop heating is raising rapidly with increased injection ratio before it starts to flat out around an injection ratio on 0.075. From approximately an injection ratio on 0.1 and up to 0.25 the COP is almost constant with a value at 3.75. The compressor discharges temperature is decreasing fast with growing injection ratio, before it starts to flat out with an injection ratio at 0.12. To achieve a discharges temperature at 180 ^{0}C the injection ratio must be 0.053. Moreover, the temperature out of the sink is slowly falling when the injection ratio is increasing, and the temperature is between 97 ^{0}C to 99 ^{0}C in the whole ranges of injection ratio in Figure 6.1. As seen from Figure 6.2 it will be some liquid in the solution leaving the compressor if the injection ratio is bigger than 0.12 and the vapour and liquid is in thermodynamic equilibrium. With an injection ratio of 0.056 or bigger there will be some liquid in the compressor at the liquid injection stages.

Another option Mayekawa suggest was the same compressor with an inbuilt volume ratio at 5.80 and the results from this compressor could be seen in Figure 6.3 and Figure 6.4. The only input parameter, which is changes from the VI 3.65 simulation is the absorber pressure that is set to a value on 32.88 bar, since the optimal pressure ratio for the VI 5.80 compressor is 10.96

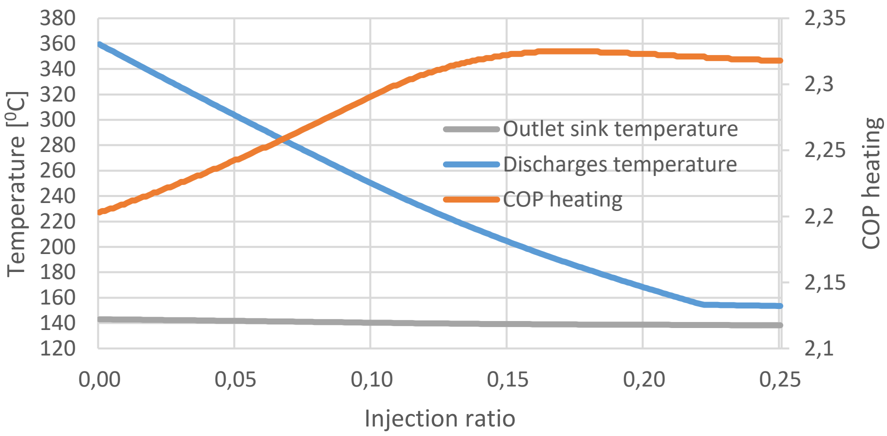

Figure 6.3: Compressor discharges temperature, outlet sink temperature, COP as a function of injection ratio for the VI 5.80 compressor

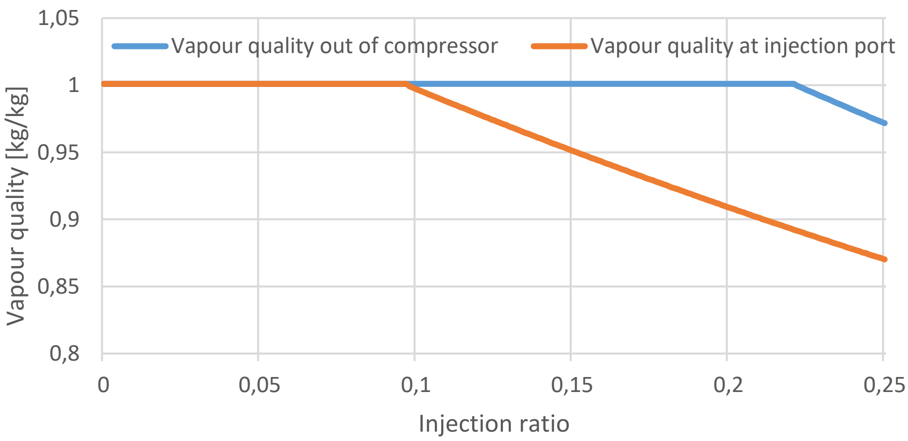

Figure 6.4: Vapour quality out of compressor and at the injection stages as a function of injection ratio for the VI 5.80 compressor

As seen from Figure 6.3 the COP heating is also increasing with raising injection ratios for the VI 5.80 compressor, but from an injection ratio on 0.17 its start slowly to decrease and the highest COP heating achieved was at 2.325. The discharges temperature for the VI 5.80 compressor follow almost the same pattern as for the VI 3.65 compressor, but here the discharges temperature will not be below 180 ^{0}C before an injection ratio at 0.18. The temperature out of the sink vary from 138 ^{0}C to 143 ^{0}C in the ranges of the injection ratio showed in Figure 6.3, with the highest temperature for the lowest injection ratios. Moreover, from Figure 6.4 it will form some liquid at the liquid injection stages with an injection ratio higher than 0.1 and liquid will occur at the outlet of the compressor if the injection ratio is higher than 0.22.

To evaluate the circulation ratio a simulation model with the VI 3.65 compressor have been made with varying the circulation ratio. The injection ratio is set to 0.1 and the rest of the input parameters is the same as in Table 6.7.

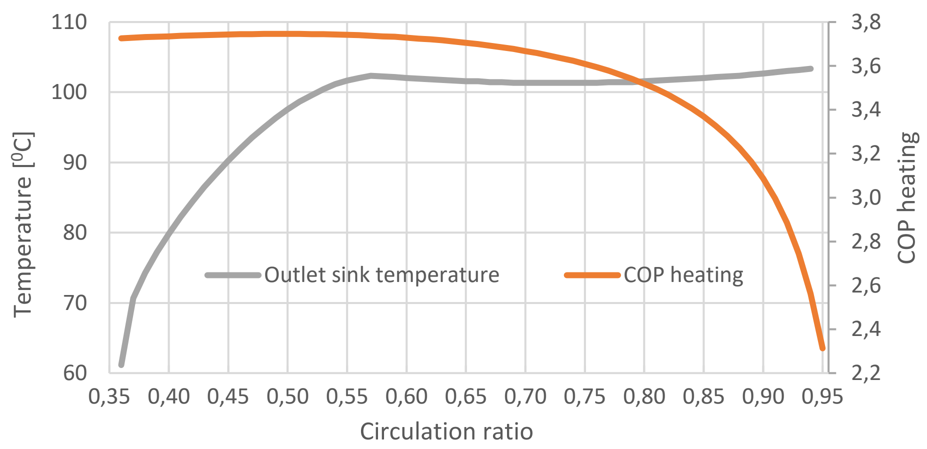

Figure 6.5: Outlet sink temperature and COP as a function of the circulation ratio for the VI 3.65 compressor

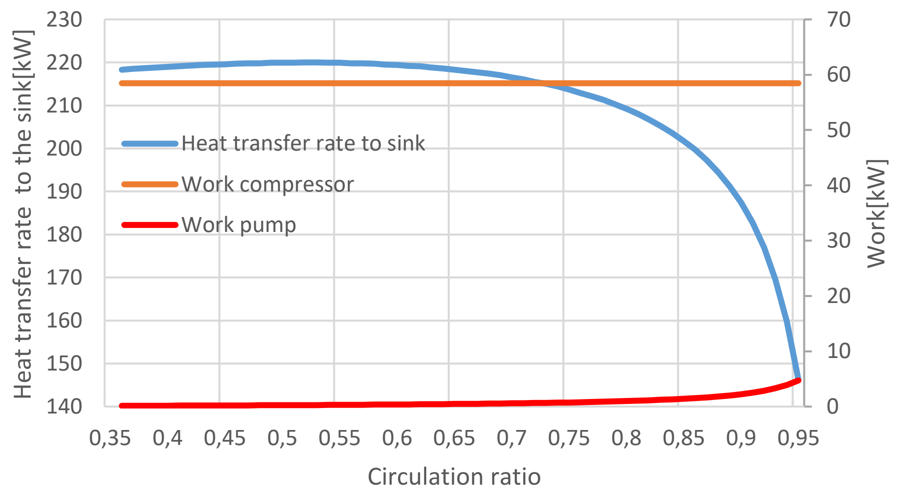

Figure 6.6: Heat transfer rate to the sink, compressor and pump work as a function of circulation ratio for VI 3.65 compressor

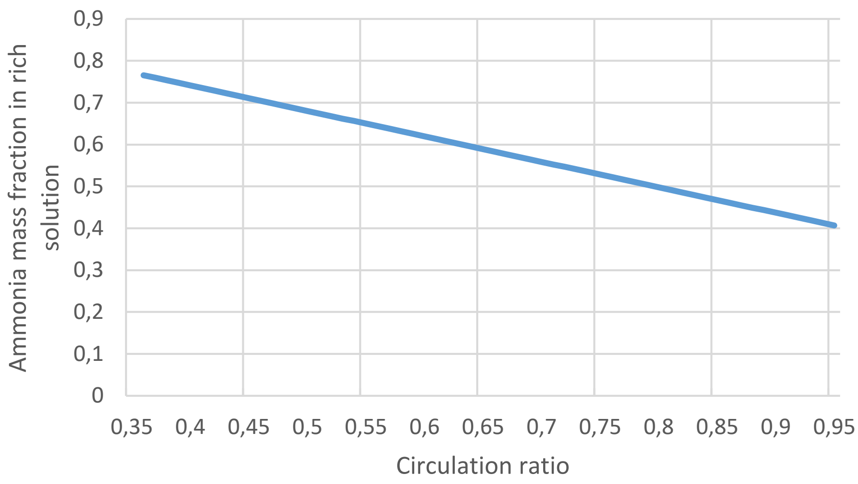

Figure 6.7: Ammonia mass fraction for changing circulation ratio for the VI 3.65 compressor

From Figure 6.5 the COP heating is highest for a circulation around 0.5 with a COP on 3.746 and the COP is starting to fall drastically after a circulation ratio at 0.7. Moreover, the temperature out of the sink is lowest with low circulation ratios and with a circulation ratio at 0.57 and 0.94 the highest temperature is achieved with a temperature at 102.4 ^{0}C. As seen from Figure 6.6 the heat transfer rate to the sink is highest with a circulation ratio at 0.52 and then decrease drastically at high circulation ratio, while the work from the solution pump is increasing with increased circulation ratio and the work from the compressor is kept constant for all circulation ratio. Moreover, from Figure 6.7 the ammonia mass fraction from the rich solution is reduce linearly with increased circulation ratio.

Evaluation of the desorber pressure is important to optimize the heat pump performance. Therefore, a simulation with varying the desorber pressure is made. The parameters which is changes from Table 6.7 is the injection ratio that is set to 0.1 and the compressor inlet mass flow rate, which is given by equation 5.1 there the swept volume is the averages swept volume of the MYCOM 125 L screw compressor with a volume ratio at 3.65 as seen in appendix F. The absorber pressure will also be changes and varying dependent on the desorber pressure. The optimal pressure ratio for the VI 3.65 compressor is 5.82 given by equation 5.2.

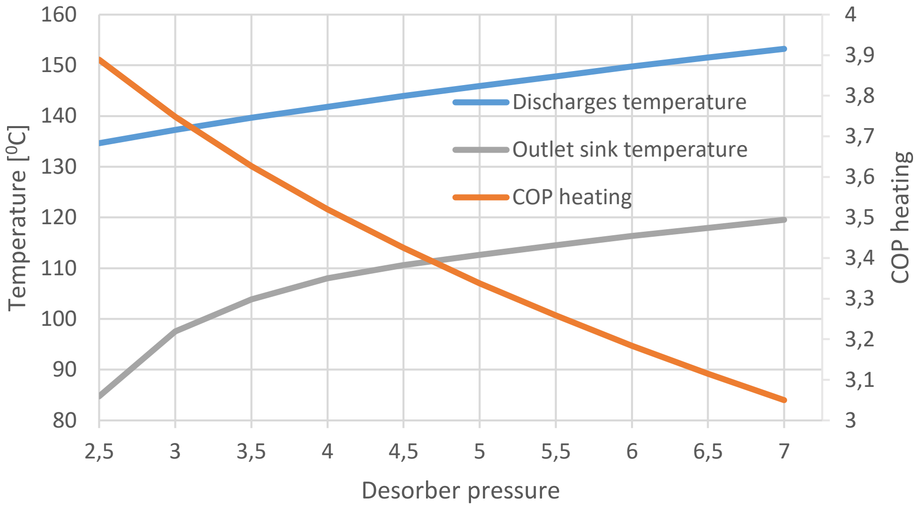

Figure 6.8: Compressor discharges temperature, outlet sink temperatures and COP as a function of the desorber pressure for the VI 3.65 compressor

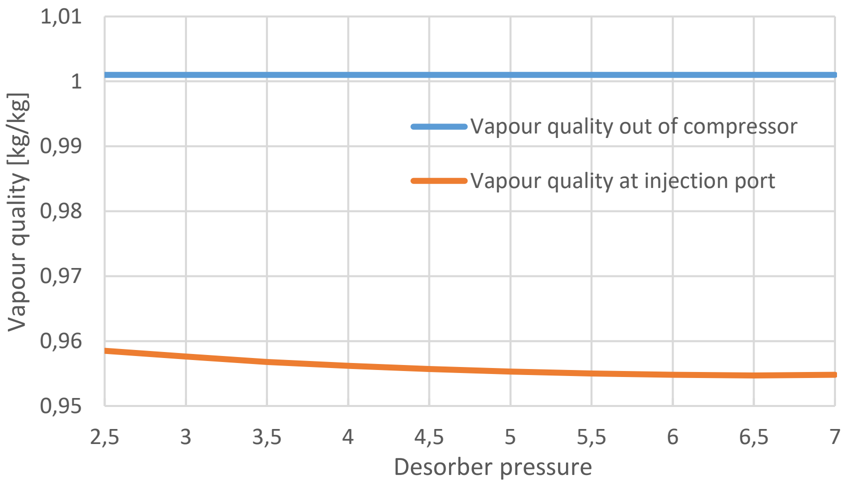

Figure 6.9: Vapour quality at the injection stages and at the outlet of the compressor as a function of desorber pressure for the VI 3.65 compressor

Higher desorber pressure will reduce the COP but increased the discharges temperature and the temperature out of the sink as showed in Figure 6.8. The highest COP is 3.89, while the highest temperature out of the sink is 119.6 ^{0}C and the maximum discharges temperature is 153.3 ^{0}C. From Figure 6.9 the fluid leaving the compressor will be pure vapour for every value of the desorber pressure, while the fluid after the mixing will consist of a small amount of liquid. The vapour quality is slowly reduced with increased desorber pressure and the reduction is only 0.0037 from a desorber pressure at 2.5 bar to 7 bar.

6.2.2 Summary and discussion

Injection ratio.

From Figure 6.1 and Figure 6.3 the lowest discharges temperature and the highest COP is obtained whit high injection ratios, while the temperature out of the sink is not affected too much with the changes in injection ratio. A reason that the COP is highest with high amount of liquid injection is that the changes in enthalpy per changes in pressure is bigger further away from the saturation curve in p-h diagram and the injection brings the vapour closer to the saturation point. From equation 4.4 a bigger change in enthalpy leads to a higher work if the mass flow rates are constant. However, the mass flow rate is getting bigger with increased injection ratio. Therefore, will these two reasons equalize each other out, after a given injection ratio and flatting out the COP, before it decreases. The heat transfer rate to the sink will also affect the COP but it is decrease slowly with increased injection ratio. The problem with injecting too much liquid is showed in Figure 6.2 and Figure 6.4, there too much liquid will lead to a high liquid mass fraction in the compressor and at the outlet of the compressor. As mentioned in chapter 3.4 there are several problems with a big amount of liquid in the compressor such as slugging, but there are hard to estimate an exact amount of liquid that can be injected before slugging occur, since it dependent on more parameters as mentioned in chapter 3.4. Therefore, an experiment should be done with each compressor to investigate when slugging will occur for that specific compressor.

Another problem with the simulation model is that the liquid, which is injected into the compressor is assumed to be in thermal equilibrium with the vapour immediately after entering the compressor, this could lead to more liquid in the compressor than assumed from the simulation. The reason for this is that the compressor velocity is high, and the compressor is short, so the amount of time it takes for the vapour to go through the compressor is short. Therefore, the vapour will most likely not have enough time to evaporate all the liquid injected into the compressors still if it is enough energy in the vapour to do it, so the amount of liquid will most likely be higher in the compressor than shown in Figure 6.2 and Figure 6.4. This will especially apply for high injection ratios and as mentioned in chapter 3.4 the compressor efficiency will decrease with increased liquid fraction in the compressor and this is not taken into account in the simulation model. This will lead to a lower COP than the results show, and it will have higher impact on high injection ratios.

Another problem that will occur if less liquid than assumed in the simulation evaporates is that the discharges temperature will be higher than assumed, since the liquid and vapour will not be in thermodynamic equilibrium and the liquid have not taken up all the possible energy from the vapour. On the other hand, as mentioned in chapter 3.4 the liquid injection is going substitute the oil in the compressor, so a small amount of liquid is needed to seal the clearances and to lubricate, so this will also affect the discharges temperature.

To sum up, the decision of how much liquid that is going to be injected into the compressor is hard and a lot of uncertainty parameter is involved, but based on Figure 6.1, Figure 6.2 and the discussion above the injection ratio should be between 0.07 and 0.12. Because this will lead to the best heat pump performances with use of the VI 3.65 compressor and it will achieve a compressor discharges temperature below 180 ^{0}C without too much liquid in the compressor. To get a more accurate calculation of how much liquid is evaporating in the compressor some experiment must be done.

Circulation ratio

As mentioned the highest COP is obtained with a circulation ratio at 0.5 for the VI 3.65 compressor. Since the mass flow rate entering the compressor, injection ratio, absorber pressure and desorber pressure is constant for all circulation ratio, the compressor work is constant. The pump work is increasing with rising circulation ratio, since the mass flow rate in the liquid are increasing, but the pump work is small compared to the compressor work as seen in Figure 6.6. Therefore, the heat transfer rate to the sink have the biggest impact on the COP. This could be seen from Figure 6.6 there the heat transfer rate to the sink almost follow the same pattern as the COP in Figure 6.5. The heat transfer rate to the sink is determined by equation 4.17 and with increased circulation ratio the mass flow rate in the rich solution raise, since the mass flow entering compressor are constant, so based on that the heat transfer should increase. On the other side, with more liquid entering the absorber, the inlet enthalpy gets lower and the saturation enthalpy for the mixture just have a small change compared with changes in the inlet enthalpy. Therefore, based on the changes in the enthalpies and mass flow rate, the heat transfer rate to the sink will achieve a maximum before it decreases and in this case, it is at a circulation ratio on 0.52.

As seen from Figure 6.5 the temperature out of the sink is almost constant with higher circulation ratio than 0.55 and vary only with 3 ^{0}C up to circulation ratio at 0.95. This is a similar pattern as the temperature in Bjørvik (2018), but the temperature in Bjørvik (2018) is flatting out at lower circulation ratio and is a One-stages system. A reason that the temperature is lower for low circulation can be that the rich solution contains higher ammonia concentration as seen in Figure 6.7 and the absorber is modeled with increasing the mass flow rate in the sink when the temperature differences is less than 5 K. Since the ammonia mass fraction is higher the temperature leaving the absorber will be lower, because of the assumptions about pure liquid is leaving the absorber. Therefore, the temperature near the outlet of the absorber will often be lower than 5 K, if not the mass flow rate in the sink is high and this will again lead to a reduction in the sink temperature as seen from equation 4.3.

Based on Figure 6.5 the optimal circulation ratio for the VI 3.65 compressor should be between 0.55 to 0.6. There both the COP and outlet sink temperature is high and from Figure 6.6 the heat transfer rate is also high at these values. From Figure 6.5 it is possible to see that the circulation ratio has a huge impact on the performances of the system, just with a small change in the circulation ratio both the COP and outlet sink temperature can changes drastically. Moreover, from Table 6.2 and Table 6.4 it is possible to see that the optimal circulation ratio varies a lot dependent on the configuration of the system and input parameters.

Pressure

From Figure 6.1 and Figure 6.3 it is possible to see that the discharges temperature is much higher for the VI 5.8 compressor than the VI 3.65 compressor. The main reason for this is the pressure ratio is higher for the VI 5.8 compressor and as mentioned in chapter 3.1 the pressure ratio has a huge influence on the discharge's temperature. As seen from Figure 6.8 the discharges temperature is increased a bit when the desorber pressure are increasing, but it cannot be compared with the big increase when the pressure ratio getting bigger. Moreover, the high discharges temperature for the VI 5.8 compressor could be a problem, since the injection ratio need to be higher than 0.18 in this simulation model to achieve a discharges temperature under 180 ^{0}C, which is recommended in chapter 3.2. The problem with the time of evaporation as mentioned above will also be a problem here, so it could be a challenge to get the discharges temperature for the VI 5.80 compressor under 180 ^{0}C for any injection ratios values. As seen from Figure 6.4 this will lead to a high liquid content in the compressor and some other problem in the compressor could occur, which have been explained above. However, since the temperature of the oil in the compressor was the main constrains and this is substituted with liquid injection it is hard to tell what the allowable discharges temperature could be. One thing to be awarded of, is to not evaporate all the liquid, since this is going to be used as the lubricate without the oil. Another constrain in the compressor could be the material that are selected, but that is something the compressor company must set the restriction for.

From Figure 6.2 and Figure 6.4 it is possible to see that the VI 5.8 compressor have lower liquid content for the same injection ratio than the VI 3.65 compressor. The reason for this is that the pressure ratio is bigger and therefore more superheat, which could evaporate the liquid, but as mentioned above the compressor length and speed will also affect the evaporation process.

To achieve a highest possible temperature out of the sink, Bjørvik (2018) conclude that both the absorber pressure and pressure ratio should be as high as possible and this could also be seen from Figure 6.1, Figure 6.3 and Figure 6.8. There the increase in the pressure ratio between the VI 3.65 and VI 5.80 compressor lead to a big raise in the temperature out of the sink, while from Figure 6.8 the increase in desorber pressure also led to a raise in the temperature but achieve a lower temperature than the VI 5.80 compressor, which had a lower absorber pressure, but higher pressure ratio. As mentioned above the pressure ratio is what influence the discharges temperature most. Therefore, the desorber pressure should be as high as possible, and this implicate that the absorber pressure also needs to be high. From Figure 6.8 higher desorber pressure reduce the COP, while increasing the temperature out of the sink. Another problem with increasing the pressure is often that the compressor and heat exchanger have a maximum pressure they could operate under and this should not be exceeded. From chapter 3.3 AlfaLaval (2016) had a heat exchanger that can handle pressure up to 26.8 bar and a temperature at 250 ^{0}C this could be a good indication for what the absorber pressure could be. With increasing the desorber pressure without increasing the temperature in the sources this will lead to an increase in the ammonia mass fraction in all the flows.

To sum up, it is important to reduce the pressure ratio to achieve an acceptable discharges temperature and have a high as possible absorber pressure to achieve the highest temperature out of the sink, but this will affect the COP negative. The VI 3.65 compressor looks like it will suit the test rig better than VI 5.80 with lower and acceptable discharges temperatures at low injection ratios. However, the allowable discharges temperature is something the compressor company must figure out.

6.3 Absorber discussion

As mentioned in chapter 3.3 the absorber is an important component and the decision about which absorber is going to be used in the test rig is complex, since both the absorber mode and heat exchanger type must be chosen. From chapter 3.3 the two main modes used in the absorber is the bubble mode and the falling film mode and the advantages with use of the falling film mode is the low pressure drop. The advantages with using bubble mode is the high heat transfer rate and good mass transfer performances because of the use of vapour distribution, while the main disadvantages are the pressure drop through the absorber. However as Lee et al. (2002) mentioned the pressure drop problem will be biggest for low-pressure system. Therefore, the pressure drop will not be so big for the absorber in a HACHP, since the pressure are high. Moreover, the disadvantages with using the falling film mode was wettability problem. Therefore, good liquid distribution at the at the liquid flow inlet is required.

Moreover, as mentioned in chapter 3.3 the two main heat exchangers, which have been used as an absorber is shell/tube heat exchanger and plate heat exchanger. The advantages with the shell/tube heat exchanger is that they can handle high temperatures and pressure. In Addison, the pressure drop is small through the heat exchanger, but the main disadvantages are that the plate heat exchanger could be more compact because of the high heat transfer rate. Moreover, the plate heat exchangers are easy to regulate and that they are easy to maintain. On the other side the disadvantages are the pressure drop, problem with handling high temperature differences, and problem with very high pressure and temperature. The problem with the temperature differences will most likely not be a big problem in the absorber in an HACHP, since as mentioned in chapter 2.1 the benefit with HACHP cycle is the temperature glide in the absorber, which will reduce the temperature differences between the sink and the working fluid. As mentioned in chapter 3.3, AlfaLaval (2016) have the latest year developed new plate heat exchanger, which could handle temperatures up to 250 ^{o}C and pressure up to 26.8 bar. This will reduce the problem with high pressure and temperatures in plate heat exchanger, since this will most likely be in the operating ranges for the test rig.

To sum up, the plate heat exchanger with bubble mode seems as the best option, still if most of the research is done for low pressure system, but there is some indication that it will suit higher pressure as well.

6.4 Uncertainties in the model.

As mentioned in chapter 4.2 the compressor is modeled so that all the liquid is mixed in the intermediate stages and the evaporation process takes places there. Another uncertainty than the evaporation process and the amount of liquid used for cooling instead of sealing and lubrication, is that the vapour is corrected for the isentropic efficiency in the intermediate stages before the mixing. Furthermore, the injected liquid and vapour are compressed and again corrected for the isentropic efficiency, so the vapour injected into the compressor are adjusted two times for the isentropic efficiency and both time the pressure ratio that is used is for the whole compression. This led to a bit lower COP, but higher compressor discharges temperature and temperature out of the sink. Jensen (2015) did a two stages compression with mixing between the compressor, so the main differences are that the isentropic efficiency is calculated for a lower pressure ratio. On the other hand, as mentioned above the compressor is not adjusted for the efficiency reduction with the liquid in the compressor, which will lead to a higher COP in the simulation and lower discharges temperature and this will especially have a bigger effect for high liquid injection.

Since the suggestion for type of compressor was not given before 15 of May the isentropic efficiency and volumetric efficiency is not adapted to the compressor from Mayekawa (MAYEKAWA, 2009). However, from Table 6.2 to Table 6.6 the isentropic efficiency is lowest for the injection compressor arrangements in most of the cases, which use a screw compressor instead of reciprocating compressor. As mentioned in chapter 3.4 the reciprocating compressor should have a bit higher isentropic efficiency than the screw compressor and the screw compressor should have an isentropic efficiency between 0.5-0.8, which it has in all the cases. This may indicate that the estimation of the isentropic efficiency is accurate. All the other assumptions in the model is mentioned in chapter 4.1

6.5 Validation

One way to validate the models is to compare the results with result from other papers and in appendix A the IHEX option 1 is compared with Nordtvedt (2005) simulation results. Instead of using the circulation ratio and heat transfer rate to the sink, Nordtvedt (2005) used the mass flow rates in the lean solution and vapour as input parameters. Therefore, to achieve the same input parameters the circulation ratio and heat transfer to the sink is adjusted. As seen from the appendix A Nordtvedt (2005) get about 0.1 lower COP than the simulation model in this thesis, which is a deviation at 2.79%. The reason for this could be the different use of the desuperheater and that the model in this thesis do not contain any cooling in the compressor. Nordtvedt (2005) achieve a higher temperature out of the sink with 1.2 ^{0}C and a reason for this could be that Nordtvedt (2005) utilizing the desuperheater more. However, 1.2 ^{0}C is a small temperature difference and as mentioned in chapter 3.1, the utilizing of the desuperheater did not have a big effect on the system something this shows. 1.2 ^{0}C result in a deviation at 0.32 %.

As mentioned in chapter 3.1 Jensen (2015) compared some different compressor arrangements. The IHEX option 1 and 2 in this thesis is almost identical with two of Jensen (2015) compressor arrangement. Unfortunately, Jensen (2015) do not have a compressor arrangement with injection from the lean solution, but the injection from the rich solution in this thesis is similar to the liquid injection from Jensen (2015) and the differences is mentioned in chapter 6.4. Since the temperature lift is the same for all the compressor arrangement in Jensen (2015) simulation model, it is best to compared it with the results in case 2. Because that is the only case with the same temperature lift process for all the compressor arrangement. From Table 6.3 and Jensen (2015) simulation model both have the IHEX option 1 as the most efficient system, while Jensen (2015) had the IHEX option 2 as the system with the lowest efficiency and from case 2 the IHEX option 2 is the system with the second highest COP. An explanation for this could be the differences in the isentropic efficiency there Jensen (2015) had a fixed value for all the compressor arrangements, but in Jensen (2015) thesis all the compressors are of the same type. Moreover, as mentioned in chapter 6.4 the differences in the compressor calculation will lead to a small decrease of the COP for the injection compressors in this thesis and the differences in the injection ratio between the two theses will also affect the COP as seen from Figure 6.1.

As mentioned in chapter 3.1, Jensen (2015) figured that the compressor arrangement giving the highest discharges temperature was IHEX option 2, while from Table 6.3 the IHEX option 2 is the compressor arrangement giving the lowest discharges temperature. In IHEX option 2 the internal heat exchanger between the vapour and lean solution is before the internal heat exchanger between the rich and lean solution wen following the lean solution flow. As mentioned in chapter 6.1.2 the cooling effect to the vapour should be bigger for IHEX option 2 than IHEX option 1.

In Chapter 3.4 Lee et al. (2015) compared the injection of vapour and liquid into a compressor and injection of liquid was superior when it comes to low discharges temperatures. As mentioned in chapter 6.1.2 this is the same principle as for the IHEX options. There the cooling effect is best when the vapour is cooled down with the liquid that is coldest, if the fluids have the same composition. Therefore, the temperature between the compressor should be lowest for IHEX option 2 as mentioned in chapter 6.1.2 and the discharges temperature as well, if the simulations has the same pressure ratio.

From Figure 3.7 and Figure 6.1 it is possible to see the differences between how the COP is influences by the injection ratio between Lee et al. (2015) study and the VI 3.65 compressor from Mayekawa. Lee et al. (2015) defined the injection ratio different then this thesis with the ratio between the mass flow rate of injection and the mass flow rate leaving the compressor. This will give a small influence on the result. Lee et al. (2015) obtained the highest COP with an injection on 0.1 before it drastically declines, while the VI 3.65 compressor obtain the highest COP at injection ratio at 0.25, but it is almost constant from an injection at 0.1-0.25. The main reason for the differences is that the isentropic efficiency in this thesis is not corrected for the liquid in the compressor. As mentioned in chapter 6.2.2. If the isentropic efficiency had been adjusted the COP in Figure 6.1 also had decreased after a given injection ratio and it had been steeper for high injection ratios as in Figure 3.7.

All in all, it can be said that the result from the simulation models in this thesis is similar to other reports with the assumption made except the result from Jensen (2015), but the systems and input parameters is different, so some differences in the result are expected.

7 Conclusion and Further work

Conclusion

From chapter 5 and 6 the most important result is:

Simulation tool

- A simulation tool with explanation for the planed test rig is made, but it needs an upgrade for the isentropic efficiency and volumetric efficiency dependent on the compressor that are chosen.

Absorber

- The trend is that bubble mode is the best option for the absorber in low pressure system and that plate heat exchanger are mostly used, but the are some indication that it will suit the temperature and pressure range for the HACHP as well.

Heat pump configuration

- Injection from the lean solution is the heat pump configuration that suits the planed test rig best with relatively high outlet sink temperatures and COP. In Addison, to the low investment cost and spaces requirement.
- In the simulation, IHEX option 2 is the best heat pump configuration to achieve high sink temperature with low compressor discharges temperatures. With a maximum compressor discharges temperature at 170 ^{0}C, the IHEX option 1 achieve an outlet sink temperature at 131.9 ^{0}C in case 1 with a COP on 2.446.
- For the same temperature lift process for all configuration the IHEX option 1 is the most efficient system with an COP at 3.751 for case 2

Heat pump optimization

- The VI 3.65 compressor from Mayekawa suits the test rig better than the VI 5.8 compressor with lower compressor discharges temperatures and higher COP, but lower sink temperatures. With an injection ratio of 0.053 the compressor discharges temperature in the simulation is 180 ^{0}C for the VI 3.65 compressor with an COP at 3.677 and an outlet sink temperature at 98 ^{0}C

- For the VI 3.65 compressor the optimal injection ratio from the simulation model is 0.07-0.12 and with an injection ratio at 0.1 the COP is 3.746, while the outlet sink temperature was 97.55 ^{0}C.
- The heat pump is very sensitive to changes in circulation ratio and the optimal circulation ratio for the VI 3.65 compressor in this simulation model is between 0.55 and 0.60. A circulation ratio on 0.57 gave a COP on 3.738 and temperature out of the sink at 102.4 ^{0}C. However, the optimal circulation ratio is varying a lot between different heat pump configurations and input parameters.
- The desorber pressure in the simulation with the VI 3.65 compressor have a huge impact on the system. The outlet sink temperature rise with increased desorber pressure, while the COP decrease. The limitation with high desorber pressure is that the absorber pressure gets higher, so the pressure limit by the compressor, heat exchanger and absorber will be the limitation. With a desorber pressure at 4 bar the temperature out of the sink were 108.5 ^{0}C and the COP were 3.52.

Validation

- The simulation result from this thesis is compared with other research and the results are similar to some of them, but the results from Jensen (2015) had some deviation in results. Compared to Nordtvedt (2005) the result for the IHEX 2 had just small deviation and the COP deviate with 2.79%, while the temperature out of the sink with 0.32%.

Further work

From the work with the master thesis this is the suggestion for further work:

- Optimizing the simulation tool:
    - When the compressor for the test rig is chosen, insert the volumetric and isentropic efficiency for the compressor in the tool, which is often given by the compressor manufactures.
    - One of the biggest uncertainties in the simulation model is the evaporation process in the compressor. Therefore, experiment measurement should be done to find correlation that suit the process and implementing into the script.

    - A correlation for how the isentropic efficiency is affected by the liquid in the compressor should be obtained from experimental values.
- Measured values from the test rig should be compared with the value from the simulation to find some improvement in the simulation tool
- Both with the simulation tool and the test rig the optimal places of injecting liquid into the compressor and the injection pressure should be find.
- An interesting optimization of the test rig could be to use an ejector to mix the vapour and lean solution at the inlet of the absorber and this may reduce the compressor work.
- Investigated the compressor limitation:
    - Since the compressor is with liquid injection, which can substitute the oil it has been interesting to investigate the compressor allowable discharges temperature and how this is influencing the system, if this is not given by the compressor manufacturer.
    - Figured out the maximum injection ratio before slugging occur.

8 Reference list

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Appendix

A Comparison with Nordtvedt

Table A. 1: Inputs parameters to compare Nordtvedt and IHEX option 1

Input parameters	Nordtvedt input	IHEX Option 1 input
Inlet temperature sink [^{0}C]	50	50
Inlet temperature sources [^{0}C]	50	50
Minimum temperature differences desorber [K]	5	5
Minimum temperatures differences absorber [K]	5	5
Thermal efficiency of IHX1	0.84	0.84
Thermal efficiency of IHX2	0.84	0.84
Absorber pressure[bar]	19	19
Intermediate pressure [bar]	7.82	7.823
Desorber pressure[bar]	2.6	2.6
Mass flow rate in the lean solution [kg/s]	0.0246	0.02464
Mass flow rate in the rich solution [kg/s]	0.0378	0.03786

Table A. 2: Outputs parameters to compare Nordtvedt and IHEX option 1

Outputs parameters	Nordtvedt input	Simulation output	Deviation [%]
Input parameters	0.0624	0.0625	0.16
Inlet temperature sink [^{0}C]	0.344	0.3534	2.66
Inlet temperature sources [^{0}C]	0.98	0.9804	0.04
Minimum temperature differences desorber [K]	0.73	0.7332	0.44
Minimum temperature differences absorber [K]	3.41	3.508	2.8
Thermal efficiency of IHX1	57.5	58.2	1.2
Thermal efficiency of IHX2	5.4	4.339	24.45
Absorber pressure[bar]	92.3	92	0.06
Intermediate pressure [bar]	96.3	95.1	0.33

B Additional results

Table B. 1: Additional results from case 1

Parameters	One-stages	Injection lean	Injection rich	IHEX option 1	IHEX option 2
Lorenz COP [-]	12.22	7.336	8.299	6.746	6.445
Mass flow rate inlet compressor [kg/s]	0.132	0.1379	0.1383	0.1407	0.1584
Mass flow rate outlet compressor [kg/s]	0.132	0.1517	0.1522	0.1407	0.1584
Mass flow rate outlet absorber [kg/s]	0.2639	0.2757	0.2905	0.2813	0.3196
Mass flow rate sink [kg/s]	4.754	1.1055	1.394	0.8772	0.7256
Mass flow rate sources [kg/s]	0.9581	0.9868	0.9969	0.9936	1.025
Injection/intermediate pressure [bar]	-	6.016	5.708	7.292	7.207
Outlet temperature sources [^{0}C]	9.55	10.25	9.85	10.95	16.85
Compressor work [kW]	76.84	75.67	70.66	76.91	101.65
Pump work [kW]	0.1898	0.2501	0.222	0.3099	0.5498
Ammonia mass fraction inlet compressor [-]	0.975	0.975	0.975	0.975	0.975
Ammonia mass fraction inlet pump [-]	0.3278	0.3278	0.3278	0.3278	0.3278
Ammonia mass fraction outlet compressor [-]	0.6514	0.6514	0.6514	0.6514	0.6514

Table B. 2: Additional results from case 2

Parameters	One-stages	Injection lean	Injection rich	IHEX option 1	IHEX option 2
Lorenz COP [-]	7.746	7.756	7.767	7.759	7.872
Mass flow rate inlet compressor [kg/s]	0.1292	0.1387	0.1375	0.1422	0.1489
Mass flow rate outlet compressor [kg/s]	0.1292	0.1525	0.1512	0.1422	0.1489
Mass flow rate outlet absorber [kg/s]	0.2584	0.2773	0.2886	0.2844	0.2979
Mass flow rate sink [kg/s]	1.194	1.194	1.194	1.194	1.194
Mass flow rate sources [kg/s]	0.931	0.997	0.986	1.022	1.046
Injection/intermediate pressure [bar]	-	5.832	5.888	6.044	5.284
Outlet temperature sources [^{0}C]	9.95	10.05	10.15	10.05	11.35
Compressor work [kW]	84.36	72.17	74.18	66.4	67.96
Pump work [kW]	0.2108	0.2340	0.2372	0.2435	0.2620
Ammonia mass fraction inlet compressor [-]	0.975	0.975	0.975	0.975	0.975
Ammonia mass fraction inlet pump [-]	0.3278	0.3278	0.3278	0.3278	0.3278
Ammonia mass fraction outlet compressor [-]	0.6514	0.6514	0.6514	0.6514	0.6514

Table B. 3: Additional results from case 3

Parameters	One-stages	Injection lean	Injection rich	IHEX option 1	IHEX option 2
Lorenz COP [-]	8.264	7.819	8.237	7.704	7.390
Mass flow rate inlet compressor [kg/s]	0.1318	0.1383	0.1384	0.1407	0.1759
Mass flow rate outlet compressor [kg/s]	0.1318	0.1521	0.1522	0.1407	0.1759
Mass flow rate outlet absorber [kg/s]	0.3296	0.3456	0.3598	0.3518	0.4397
Mass flow rate sink [kg/s]	1.132	1.002	1.111	0.969	0.762
Mass flow rate sources [kg/s]	1.194	1.239	1.245	1.257	1.431
Injection/intermediate pressure [bar]	-	6.016	5.769	7.154	7.207
Outlet temperature sources [^{0}C]	17.45	18.35	18.05	18.55	27.85
Compressor work [kW]	76.76	75.87	71.72	73.19	112.85
Pump work [kW]	0.2844	0.3762	0.3416	0.411	0.9156
Ammonia mass fraction inlet compressor [-]	0.975	0.975	0.975	0.975	0.975
Ammonia mass fraction inlet pump [-]	0.3278	0.3278	0.3278	0.3278	0.3278
Ammonia mass fraction outlet compressor [-]	0.5867	0.5867	0.5867	0.5867	0.5867

Table B. 4: Additional results from case 4

Parameters	One-stages	Injection lean	Injection rich	IHEX option 1	IHEX option 2
Lorenz COP [-]	8.672	5.297	5.843	2.358	5.192
Mass flow rate inlet compressor [kg/s]	0.0566	0.0547	0.0552	0.0590	0.0758
Mass flow rate outlet compressor [kg/s]	0.0566	0.0656	0.0663	0.0590	0.0758
Mass flow rate outlet absorber [kg/s]	0.0943	0.0911	0.1031	0.0985	0.1263
Mass flow rate sink [kg/s]	0.6522	0.2209	0.2673	0.2363	0.1789
Mass flow rate sources [kg/s]	0.3661	0.3186	0.3267	0.3438	0.3440
Injection/intermediate pressure [bar]	-	11.08	10.18	13.68	13.80
Outlet temperature sources [^{0}C]	8.25	11.55	10.85	11.35	24.85
Compressor work [kW]	31.95	50.33	44.77	42.23	65.86
Pump work [kW]	0.0734	0.1783	0.1508	0.1884	0.4935
Ammonia mass fraction inlet compressor [-]	0.9842	0.9842	0.9842	0.9842	0.9842
Ammonia mass fraction inlet pump [-]	0.3763	0.3763	0.3763	0.3763	0.3763
Ammonia mass fraction outlet compressor [-]	0.7411	0.7411	0.7411	0.7411	0.7411

C Schematic diagram of the different configuration

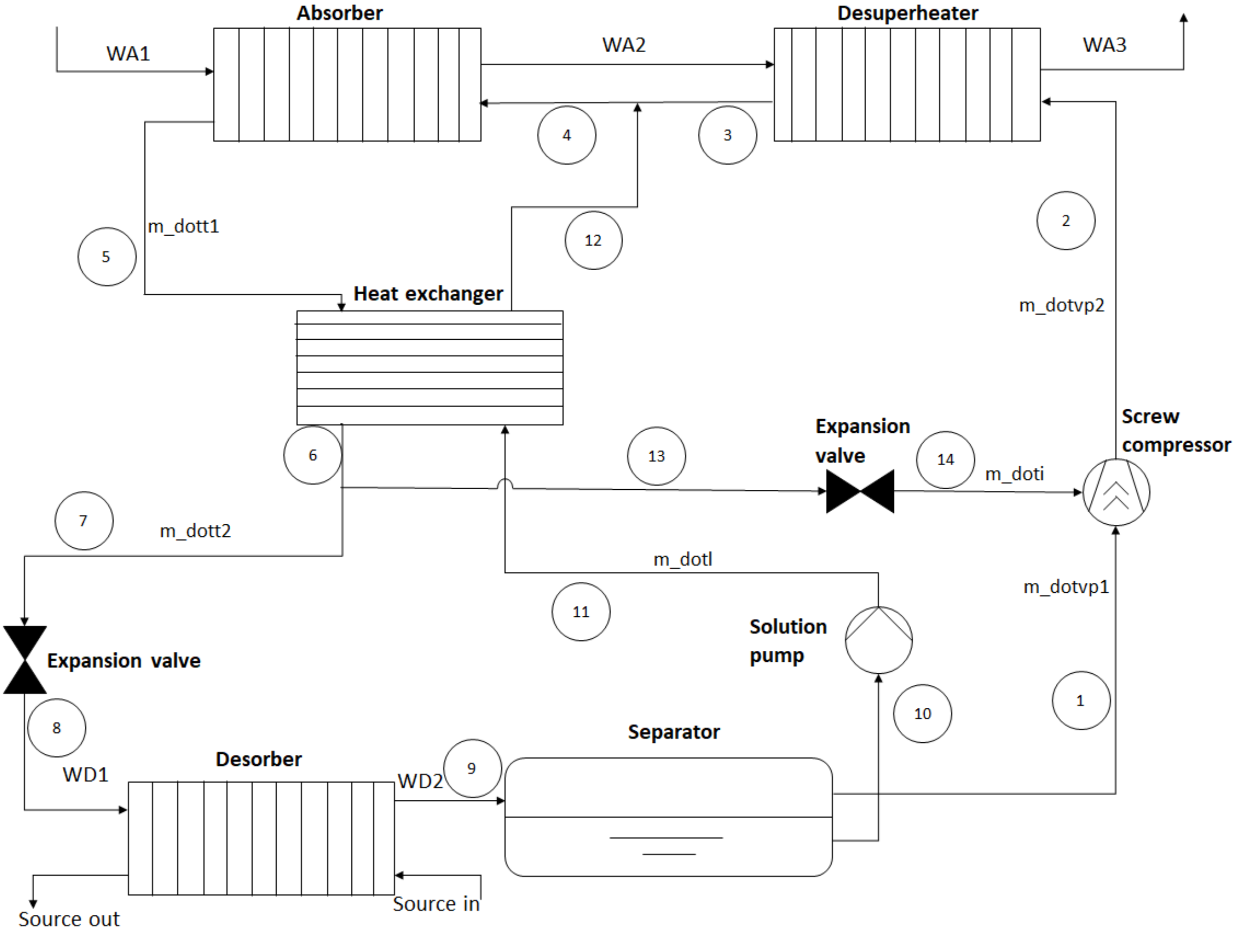


Figure C.1: Schematic diagram of the simulation model with injection from rich solution

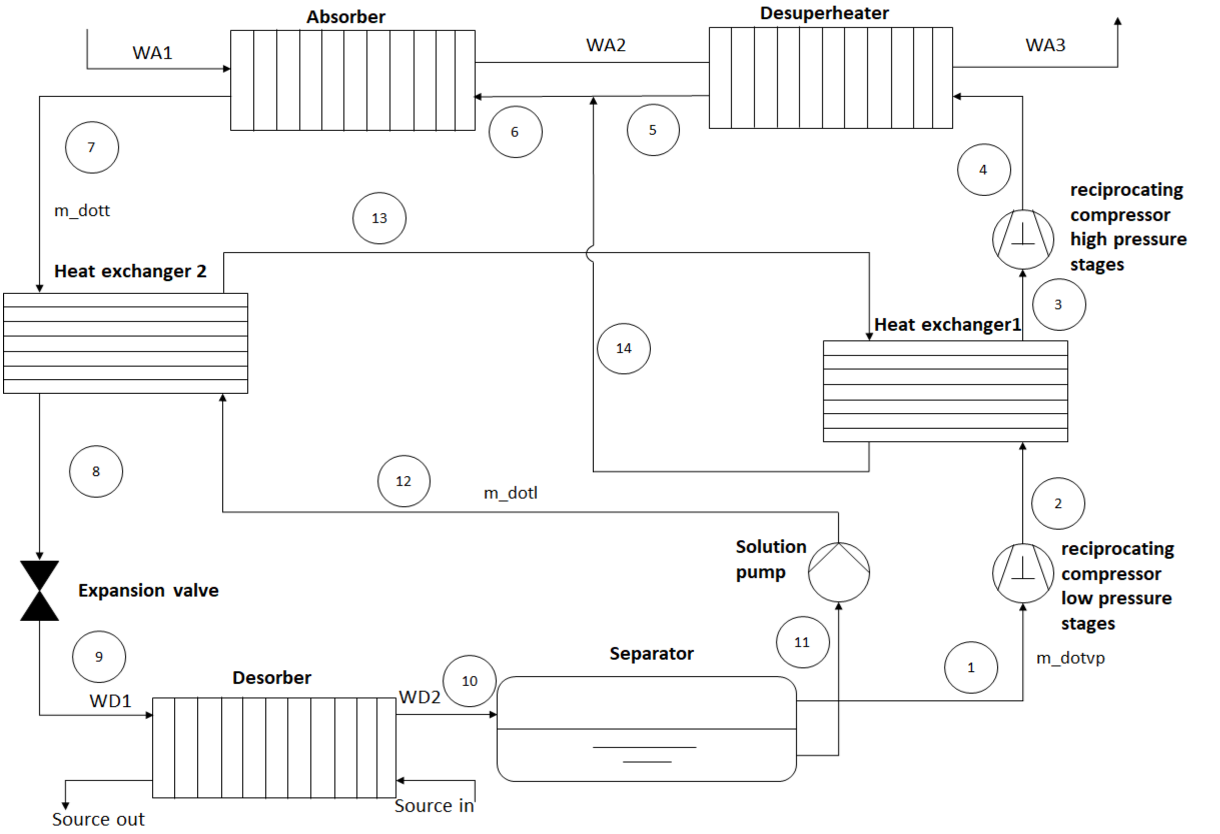


Figure C.2: Schematic diagram of the simulation model of IHEX option 1

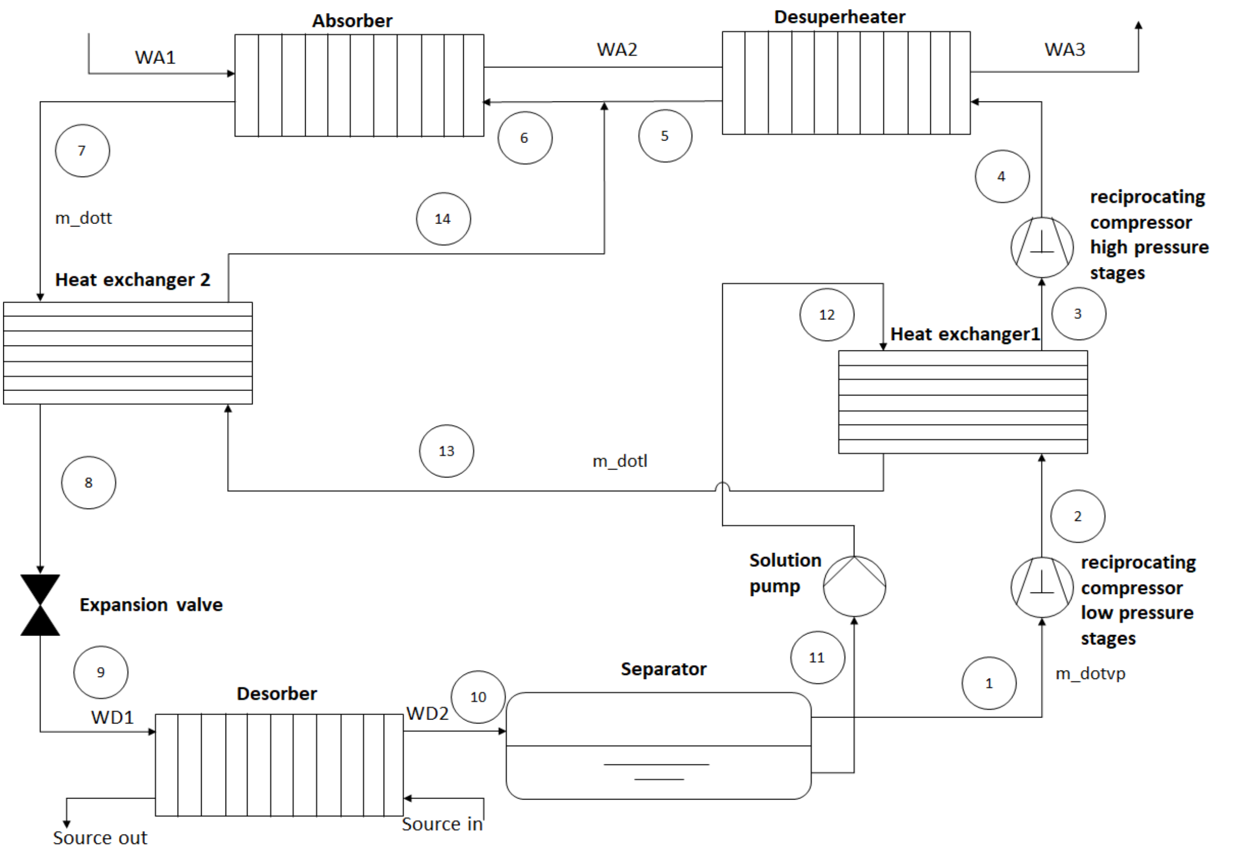


Figure C.3: Schematic diagram of the simulation model of IHEX option 2

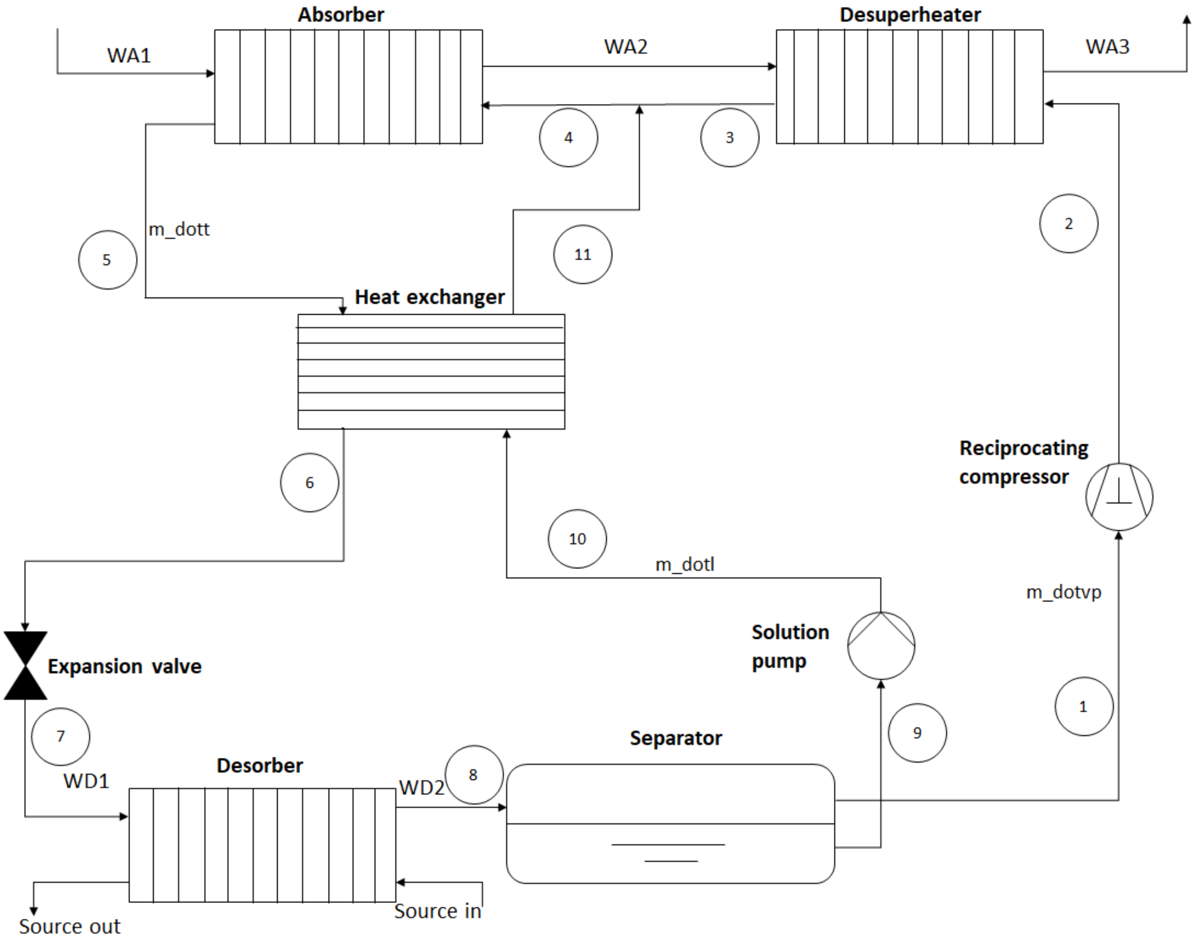


Figure C.4: Schematic diagram of the simulation model of one-stages

D EES code/simulation tool for injection from lean solution

The simulation model of injection from the lean solution, which is the simulation tool will be presented in this appendix

```
$UnitSystem SI K bar mass rad

{Procedures}
Procedure ihx(T;K;O;Z;X;M;N;E:R1;R2) {Procedure for IHEX}
i:= 1
Call NH3H2O(123;T;O;X:T[i];P[i];X[i];h[i];s[i];u[i];v[i];q[i]) {T[5], P_HP, ZZ}
i:= i +1
Call NH3H2O(123;K;O;X:T[i];P[i];X[i];h[i];s[i];u[i];v[i];q[i]) {T[12], P_HP, ZZ}
qh:= (h[1] - h[2]) * M {qh- ideal heat exchange from 5-6. M-Vapor flow rate}
i:= i + 1
Call NH3H2O(123;T;O;Z:T[i];P[i];X[i];h[i];s[i];u[i];v[i];q[i]) {T[5], P_HP, x_liquid}
i:= i + 1
Call NH3H2O(123;K;O;Z:T[i];P[i];X[i];h[i];s[i];u[i];v[i];q[i]) {T[10], P_HP, x_liquid}
qc:= (h[3] - h[4]) * N {qc- ideal heat exchange from 12-13.}
qihx:= 0
if(T<K) Then Goto 10
qmax:= min(qh;qc) {ideal heat exchange in IHEX}
qihx = qmax * E {actual heat exchange in IHEX}
10: R1:= h[1] - (qihx / M) {Return value for h[6]}
R2:= h[4] + (qihx / N) {Return value for h[13]}
End



Function dsh(T;K;F;P;Z;M) {Function to determine thermal efficiency of the desuperheater.}
i:= 1
Call NH3H2O(123;T;P;Z:T[i];P[i];x[i];h[i];s[i];u[i];v[i];q[i]) {T[2], P_HP, x_vapour2}
i:= i +1
Call NH3H2O(123;K;P;Z:T[i];P[i];x[i];h[i];s[i];u[i];v[i];q[i]) {T[3], P_HP, x_vapour2}
i:= i + 1
Call NH3H2O(123;F;P;Z:T[i];P[i];x[i];h[i];s[i];u[i];v[i];q[i]) {T_wa2, P_HP, x_vapour2}
q:= (h[1] - h[2]) * M {actual heat exchange in dsh}
qmax:= (h[1] - h[3]) * M {ideal heat exchange in dsh}
epsilon:= q / qmax {dsh efficency}
dsh:= epsilon
End


Procedure absorber(Z[50..100];A[50..100];O;M;N;T_WA1;Y:C1;C2)
K[50]:= T_WA1 {Heat sink inlet temperature}
N:=N - 0,001 {Heat sink mass flow rate}{Adjust the accuracy}
10: i:=50 {Number of segments}
N:=N + 0,001 {Heat sink mass flow rate}{Adjust the accuracy}
r:= enthalpy('Water';P=O;T=K[50]) {Enthalpy of heat sink at the inlet absorber, O = P_A}
e[i] := r
D[i]:= Z[i]-K[50] {Temperature difference between mixed solution and heat sink }
Repeat
i:=i+1
q:=(A[i]-A[i-1])*M {Heat exchange between two points in the absorber}
e[i] := e[i-1]+(q/N) {indexed eNthalpy of the solution}
N:= (((A[i]-A[50])*M)/(e[i]-r)) {approxiMate heat sink mass flow rate}
L:= temperature('Water';P=O;h=e[i]) {Indexed temperature of the mixed solution}
D[i] := Z[i]-L {Indexed temerpature difference between mixed soLution and heat sink}
W[i]:= L
if(D[i]<Y) Then Goto 10 {If the temperature difference is beLow minimum temperature differences go to 10}
Until (i=100)
C1:= N {Return value of heat sink mass flow}
C2:= W[100] {Return value of heat sink temperature}
End
```

```
{INPUT PARAMETERS}
T_Sourcein = 323,15 [K] {Inlet sources temperature}
P_HP = 17,5 [bar] {Absorber pressure}
P_LP = 3 [bar] {Desorber pressure}
P_A = 10 [bar] {Sink Pressure}
epsilon_ihx = 0,9 {Thermal efficency of IHX}
epsilon_dsh = 1 {efficency of desuperheater }
CR = 0,5  {Circulation ratio}
eta_motorc = 0,9 {Compressor motor efficency}
eta_motorp = 0,9 {Pump motor efficency}
T_mindesorber= 5[K] {Minimum temperature differences in absorber}
T_minabsorber=5[K] {Minimum temperature differences in desorber}
x_WA2=0 {Ammonia concertation in sink}
Q_sink = 100 {Heat transfer rate to the sink}
T_WA1=323,15 [K] {Inlet temprature sink}
P_D=1,6 {Pressure in sources}
epsilon_des=0,9 {Thermal efficency of desorber}
w=0,1 {How much precent going to the compressor for cooling}
k=1 {Correction factor}



{Function}
P_MP = k*((P_LP*P_HP)^0,5) {Injection pressure}
Qihx = (h[5]-h[6])*m_dott {Heat transfer rate in the internal heat exchanger}
ZZ= ((x_vapour1*m_v)+(x_liquid*m_l))/(m_t) {Amonia concentraison in point 4-8}
h_4=((h[3]*m_dotvp2)+(h[13]*m_dota))/(m_dott) {Enthalpy in point 4 mixing}
m_dotl = CR*m_dott {Liquid mass flow rate in point 9-10}
m_dott = Q_absorber/DELTAh_absorber {Mass flow rate in point 4-8}
x[9]= x_liquid {Ammonia mass fraction in liquid point 9-13}
m_dotvp1= m_dott-m_dotl {mass flow rate at the inlet of the compressor}
x[1] = x_vapour1 {Ammonia mass fraction in vapour at the inlet of the compressor}
T_GlideDesorber = T_Sourcein-T_Sourceout {Temperature glide in desorber}
T_Glideabsorber = T_WA3-T_WA1 {Temperature glide in absorber}
x_vapour2 = ((x_liquid*m_cc)+(x_vapour1*m_vp1))/(m_vp2) {Ammonia mass fraction in vapour at the outlet of the
compressor. Point 2 and 3}
h_2i = (((h_2s-h[1])+(eta_isentropic*h[1]))/eta_isentropic) {Enthalpy at the injection stages in compressor before mixing }
m_dotvp2=m_dotvp1+m_dotcc {mass flow rate at the outlet of the compressor. Point 2 and 3}
m_dota= m_dotl-m_dotcc {Liquid mass flow rate in point 12-13}
h_2_inj = ((h_2i*m_vp1)+(m_cc*h[11]))/m_vp2 {Enthalpy at the injection stages after mixing in the compressor}
m_dotcc= m_dotvp1*w {Mass flow rate injection. Point 11}
eta_isentropic= 0,9051-(0,0222*PR) {Isentropic efficency compressor}
eta_volu = 1,0539-(0,0788*PR) {Volumetric efficiency}
PR= P_HP/P_LP {Pressure ratio}
h_2 = (((h_2i_inj-h_2_inj)+(eta_isentropic*h_2_inj))/eta_isentropic) {Enthalpy out of the compressor}



{DESUPERHEATER}
DELTAh_desuperheater = h[2] - h[3] {Enthalpy difference through the dsh}
Q_desuperheater = DELTAh_desuperheater*m_dotvp2*epsilon_dsh {Heat transfer rate to the sink from desuperheater}
Call NH3H2O(123;T_WA2;P_A;x_WA2:T_WA2[1];P_A[1];x_WA2[1];h_WA2;s_WA2;u_WA2;v_WA2;q_WA2) {Water
properties in the sink before desuperheater point WA2 }
h_WA3 = h_WA2+(Q_desuperheater/m_dotWA) {Enthalpy after desuperheater in the sink. Point WA3}
T_WA3 = temperature('Water';P=P_A;h=h_WA3) {Heat sink temperature after the desuperheater. Point WA3}
Call NH3H2O(123;T[2];P_HP;x_vapour2:T[14];P[14];x[14];h[14];s[14];u[14];v[14];q[14]) {Ammonia/water properties at the
innlet of desuperheater. Point 2}
epsilondsh = dsh(T[14];T[3];T_WA2;P_HP;x_vapour2;m_dotvp2) {Efficency of desuperheater}


{Desorber}
T_WD2 = T_Sourcein-T_mindesorber {Temprature out of the desorber. Point WD2}
```

```
T_Sourceout = T[7]+T_mindesorber {Temperature out of the sources}
Call NH3H2O(123;T_Sourcein;P_D;0:T_Sourcein[17];P_Sourcein[17];x_Sourcein[17];h_Sourcein[17];s_Sourcein[17];
u_Sourcein[17];v_Sourcein[17];q_Sourcein[17]) {Inlet sources properties}
Call NH3H2O(123;T_Sourceout;P_D;0:T_Sourceout[18];P_Sourceout[18];x_Sourceout[18];h_Sourceout[18];s_Sourceout[18];
u_Sourceout[18];v_Sourceout[18];q_Sourceout[18]) {Outlet sources properties}
Q_des= (h[8]-h[7])*m_dott {Heat transfer rate from the sources}
m_dotwd= (Q_des*epsilon_des)/(h_Sourcein[17]-h_Sourceout[18]) {Mass flow rate in the sources}



{ABSORBER (Array [50]-[100])}
Call absorber(T[50..100];h[50..100];P_A;m_dott;m_dotWAApprox;T_WA1;T_minabsorber:m_dotWA;T_WA2)
DELTAh_absorber = h[4] - h[5] {Enthalpy difference through the absorber }
step_aa = DELTAh_absorber / 50 {Enthalpy difference through the absorber  in 50 steps}
A[50] = h[5] {Enthalpy of solution at point 5}
Duplicate i = 51;100
A[i] = A[i-1] + step_aa {Enthalpy of mixed solution trough the absorber}
End
Duplicate i =50;100
Call NH3H2O(234;P_HP;zz;A[i]:T[i];P[i];x[i];h[i];s[i];u[i];v[i];q[i]) {Properties in the absorber}
End

h_WA1 = enthalpy('Water';T=T_WA1;P=P_A) {Heat sink inlet enthalpy}
h_ApproxWA2 = enthalpy('Water';T=T[100] - T_mindesorber;P=P_A) {Guess value of heat sink outlet temperature}
m_dotWAApprox = Q_absorber/(h_ApproxWA2-h_WA1)  {Guess value of heat sink mass flow rate}



{Loop}
{---- 1 ----}
Call NH3H2O(128;T_WD2;P_LP;1:T[1];P[1];x[1];h[1];s[1];u[1];v[1];q[1]) {After separator vapour}
{---- 2 ----}
Call NH3H2O(235;P_MP;x_vapour1;s[1]:T_2s;P_2s;x_2s;h_2s;s_2s;u_2s;v_2s;q_2s) {Injection stages before correction of
isentropic efficiency and mixing}
Call NH3H2O(234;P_MP;x_vapour2;h_2_inj:T_2s_inj;P_2s_inj;x_2s_inj;h_2s_inj;s_2s_inj;u_2s_inj;v_2s_inj;q_2s_inj)
{Injection stages after mixing}
Call NH3H2O(235;P_HP;x_vapour2;s_2s_inj:T_2i_inj;P_2i_inj;x_2i_inj;h_2i_inj;s_2i_inj;u_2i_inj;v_2i_inj;q_2i_inj) {After
compression before correction of isentropic efficiency}
Call NH3H2O(234;P_HP;x_vapour2;h_2:T[2];P[2];x[2];h[2];s[2];u[2];v[2];q[2])  {After compressor}
{---- 3 ----}
Call NH3H2O(123;T[2];P_HP;x_vapour2:T[3];P[3];x[3];h[3];s[3];u[3];v[3];q[3]) {After desuperheter}
{---- 4 ----}
Call NH3H2O(234;P_HP;ZZ;h_4:T[4];P[4];x[4];h[4];s[4];u[4];v[4];q[4])  {After mixing}
{---- 5 ----}
Call NH3H2O(238;P_HP;ZZ;0:T[5];P[5];x[5];h[5];s[5];u[5];v[5];q[5]) {After absorber}
{---- 6 ----}
Call ihx(T[5];T[12];P_HP;x_liquid;ZZ;m_dott;m_dota;epsilon_ihx:h_6;h_13)
Call NH3H2O(234;P_HP;ZZ;h_6:T[6];P[6];x[6];h[6];s[6];u[6];v[6];q[6]) {After IHEX}
{---- 7 ----}
Call NH3H2O(234;P_LP;ZZ;h_6:T[7];P[7];x[7];h[7];s[7];u[7];v[7];q[7]) {After expansion valve}
{---- 8 ----}
Call NH3H2O(123;T_WD2;P_LP;ZZ:T[8];P[8];x[8];h[8];s[8];u[8];v[8];q[8]) {After desorber}
{---- 9 ----}
Call NH3H2O(128;T_WD2;P_LP;0:T[9];P[9];x[9];h[9];s[9];u[9];v[9];q[9]) {After separator liquid}
{---- 10 ----}
Call NH3H2O(235;P_HP;x_liquid;s[9]:T[10];P[10];x[10];h[10];s[10];u[10];v[10];q[10]) {After pump}
{---- 11 ----}
Call NH3H2O(234;P_MP;x_liquid;h[10]:T[11];P[11];x[11];h[11];s[11];u[11];v[11];q[11]) {Liquid injection}
{---- 12 ----}
Call NH3H2O(123;T[10];P_HP;x_liquid:T[12];P[12];x[12];h[12];s[12];u[12];v[12];q[12]) {After pump}
{---- 13 ----}
Call NH3H2O(234;P_HP;x_liquid;h_13:T[13];P[13];x[13];h[13];s[13];u[13];v[13];q[13]) {After IHEX}
```

```
{Calculaison of COP}
W_comp1 = (m_dotvp1*(h_2s-h[1]))/(eta_motorc* eta_isentropic) {Compressor work up to the injection stages}
W_comp2 = (m_dotvp2*(h_2i_inj-h_2s_inj))/(eta_motorc* eta_isentropic) {Compressor work after injection stages}

W_comp = W_comp1+W_comp2 {Compressor work}
W_pump = ((h[10] - h[9]) * m_dotl)/eta_motorp {Pump work}
COP_heating = Q_sink / (W_comp + W_pump) {COP heating}
Q_sink = Q_desuperheater + Q_absorber {Heat transfer rate to the sink}

T_LM_sink = (T_WA1-T_WA3)/(ln(T_WA1/T_WA3)) {Logarithmic mean temperature of sink}
T_LM_sources = (T_Sourcein-T_Sourceout)/(ln(T_Sourcein/T_Sourceout)) {Logarithmic mean temperature of sources}
COP_LORENZ = T_LM_sink/(T_LM_sink-T_LM_sources) {Lorenz COP}



{MISCELLANEOUS}
m_l= CR*m_t
m_t= 1
m_v= m_t-m_l

m_cc = m_vp1*w
m_vp1 = m_vp2-m_cc
m_vp2= 1
```

E Risk assessment

Unit: *Department of energy and process engineering* **Date:** 15.01.19

Line manager: Terese Løvås

Participants in the identification process (including their function)**:** Fredrik Bjørvik (Student)

Short description of the main activity/main process: Master project for student Fredrik Bjørvik. Design of a hybrid absorption/compression high temperature heat pump test rig

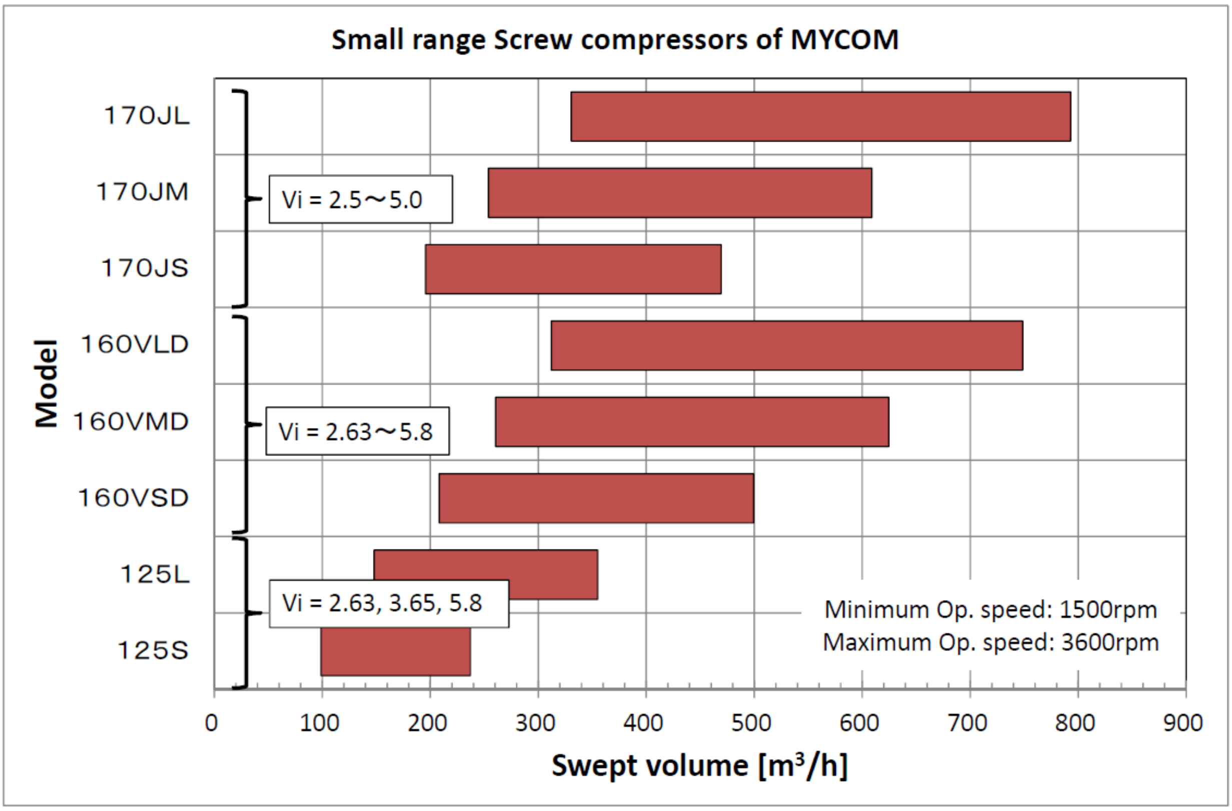
Is the project work purely theoretical? (YES/NO): YES

Signatures: *Responsible supervisor:* *Student:* Fredrik Bjørvik

ID nr.	Activity/process	Responsible person	Existing documentation	Existing safety measures	Laws, regulations etc.	Comment
	Simulation of a heat pump system in Energy equation solver (EES)	Fredrik Bjørvik				No safety massuared required

F Compressor data

Compressor data from Mayekawa, where compressor 125 L is used in this thesis (MAYEKAWA, 2009)



G Scientific paper

A draft of the scientific paper is made base on the result from this thesis.

DESIGN OF A HYBRID ABSORPTION/COMPRESION HIGH TEMPRATURE HEAT PUMP TEST RIG

Fredrik BJØRVIK[(a)]
[(a)]Norwegian University of Science and Technology, Kolbjørm Hejes vei 1D, Trondheim, 7034, Norway
frebj@stud.ntnu.no

ABSTRACT

Five different simulation models with different configuration of an absorption/compression heat pump was made to decide which one was the best to be used as a test rig. The five models were tested in four different simulation cases and the conclusion were that the injection from lean solution was the best option be to use as a test rig. Mayekawa had a compressor with a volume ratio on 3.65 and 5.8, which was tested in the simulation model for injection from lean solution and the one with a volume ratio at 3.65 suits the test rig best. Moreover, to decide the operational boundaries condition some input parameters were evaluated and the optimal circulation ratio should be between 0.55 and 0.6, which will give an outlet sink temperature at 102.3 ^{0}C with an COP at 3.74. The optimal injection ratio was found to be from 0.07 to 0.12.

1. INTRODUCTION

Today the energy consumption worldwide is increasing, and the energy increase is expected to grow with 28% from 2015 to 2040 (Doman, 2017). The manufacturing sector has a big share of the total energy consumption and in 2005, 38% of the global final energy use in the manufacturing sector was from steam system (Banerjee et al., 2012). From the steam system it is often generated surplus heat, which is not utilized, because the temperature is too low to be used directly in an industrial process. However, the surplus heat can be used as a heat sources in a heat pump to elevate the temperature. With the restriction of CFCs and HCFCs Brunin et al. (1997) says that the absorption/compression heat pump with ammonia/water mixture as working medium is the best heat pumps to utilize the surplus heat and to achieve high enough sink temperatures.
Absorption/compression heat pump process is based on the patent from Osenbrück (1895), which is a vapour compression heat pump in combination with an absorption heat pump. There are several advantages with using the Absorption/compression heat pump and one of the advantages is better capacity control than vapour compression heat pump. With varying the ammonia concentrations in the different solutions and varying the circulation ratio makes it possible to adjust the temperature and the heat transfer in a more efficient way. The circulation ratio is the ratio between the mass flow rate in the lean and rich solution as seen in equation 8. Since a zeotropic fluid is used, temperature glide in the absorber and desorber will occur, if the temperature glide is adapted to the temperatures in the source and sink, the heat exchanges losses will be reduced. The last main benefit is that the absorption/compression heat pump could achieve higher sink temperature at relatively low absorber pressure compared with vapour compression heat pump using ammonia (Nordtvedt, 2005). One of the constrains for the system is the compressor discharges temperature and Nekså et al. (1998) said that a discharger temperature under 180 ^{0}C should be achievable to maintain the lubricant in the compressor.
After Osenbrück (1895) introduce the absorption/compression heat pump no one put a real effort into study it before 1970s (Nordtvedt, 2005). Nordtvedt (2005) investigate how the circulation ratio affecting the system and it had a big impact on the system, especially the temperature lift and COP, where the highest COP were obtained at circulation ratio on 0.65 before the COP reduces. A circulation ratio at 0.65 from Nordtvedt (2005) definition is equivalent to a circulation ratio at 0.4 from Jensen (2015) definition. Jensen (2015) also investigate how the circulation ratio and ammonia mass fraction affecting the system and concluded that the optimal circulation ratio is dependent on the type of system and operating condition. Moreover, Bergland (2015) investigated the effect the desuperheater and concludes that the desuperheater have a small impact on the system.
To reduce the discharges temperature and increase the efficiency of the system a two stages compression with cooling between the compressor could be a solution. Jensen (2015) study different two-stages configuration with a fixed temperature lift for all the configuring and figured out that the IHEX option 1 as seen in Figure 3

was the best configuration. Both in terms of lower the compressor discharges temperature and the highest COP, while the injection from rich solution, which is almost the same as seen in Figure 2 obtained the second highest COP and second lowest discharges temperature. The different between the injection from rich solution in Figure 2 and Jensen (2015), is that Jensen (2015) is not injecting the fluid into the compressor instead injected between two compressors. However, as Jensen (2015) indicated the liquid injection might have a lower investment cost, since it only need one compressor and one heat exchanger.
Liquid injection into the compressor could be a substitute for the oil in the compressor with using the liquid as lubricate and for sealing (Moody and Hamilton, 1975) (Stosic et al., 2005). Bakken et al. (2018) investigated experimental how the liquid content into a centrifugal compressor affected the polytopic efficiency of the compressor and conclude that more liquid in the compressor lead to lower efficiency. Lee et al. (2015) also did an experimentally study on a refrigeration system with vapour and liquid injection in front of a scroll compressor and concludes that the highest COP is achieve with an injection ratio at 0.1, before it drastically decreases with higher injection ratio.

2. Simulation model and setup of simulations

Five model with different configuration is made in EES to investigate, which compressor arrangement is the best option for the test rig and to decide the boundaries condition. EES has an inbuilt procedure for ammonia/water mixture based on Ibrahim and Klein (1993) correlation giving eight thermodynamic properties, if three of them are given. A schematic diagram of the different configuration except the one-stages configuration can be seen in Figure 1 to Figure 4. There two of the configurations is with injection of fluid into a screw compressor and the other two is with a two-stages compression with reciprocating compressor.

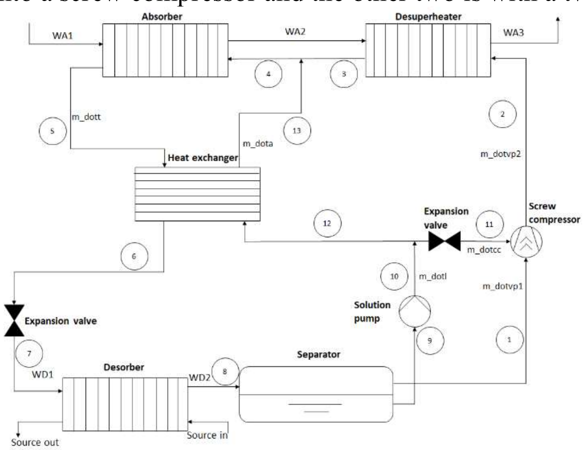


Figure 1:Schematic diagram of injection from lean solution

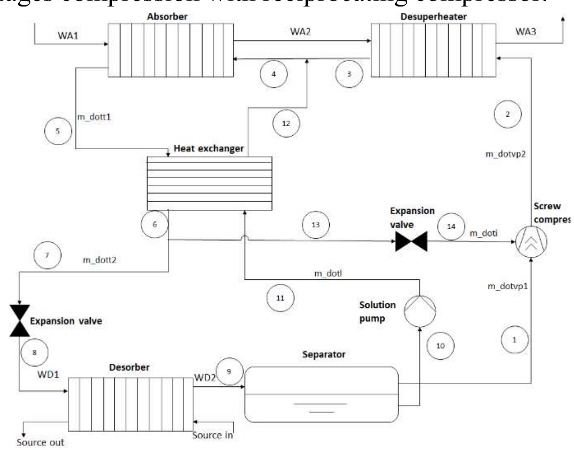


Figure 2:Schematic diagram of injection from rich solution

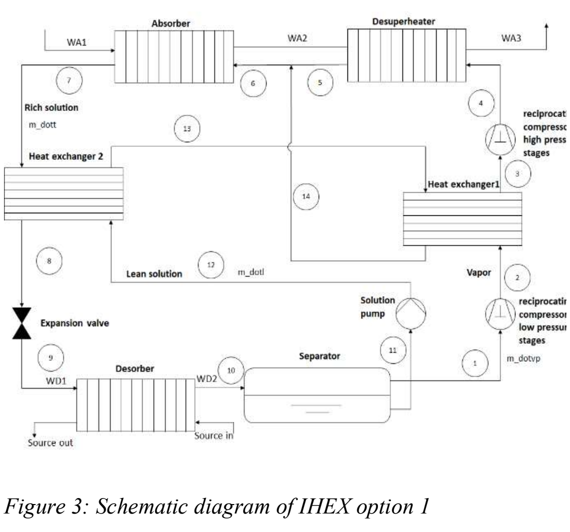


Figure 3: Schematic diagram of IHEX option 1

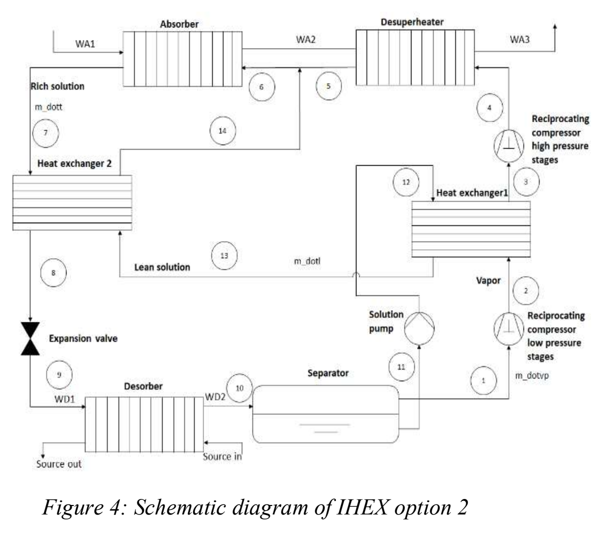


Figure 4: Schematic diagram of IHEX option 2

There are made several scenarios both to decide the compressor arrangement and the boundaries condition. All the inputs parameter that is the same for all the scenarios are listed in Table 1.

Table 1: Input parameters for all scenarios

Parameters	Value
p_{wa} [bar]	10
p_{sou} [bar]	1.6
ε_{ihex} [-]	0.9
ε_{des} [-]	0.9
$\varepsilon_{mot,com}$ [-]	0.9
$\varepsilon_{mot,pump}$ [-]	0.9
$\Delta T_{min,abs}$ [K]	5
$\Delta T_{min,des}$ [K]	5
$T_{sou,in}$ [^{0}C]	50
T_{WA1} [^{0}C]	50

2.1. Compressors

For the reciprocating compressor the volumetric efficiency and the isentropic efficiency is given by Nordtvedt (2005), which can been seen in equation 1 and 2. The screw compressor used the same volumetric efficiency as the reciprocating compressor and the isentropic efficiency for the screw compressor is given by equation 3. To use the simulation model as a simulation tool for the test rig the isentropic efficiency and volumetric efficiency, which is often given by the compressor manufactures must be suited to the compressor that are chosen. To decide the intermediate pressure equation 4 is used and in the configurations with liquid injection, the intermediate pressure is the injection pressure. In order to find the optimal amount of fluid injected into the compressor equation 5 is given and expressed with the injection ratio.

$$\eta_{volu} = 1.0539 - 0.0788 * PR\ [-] \tag{1}$$

$$\eta_{isen,rec} = 0.9051 - 0.0422 * PR\ [-] \tag{2}$$

$$\eta_{isen,screw} = 0.9051 - 0.0222 * PR\ [-] \tag{3}$$

$$p_{MP} = K * \sqrt{p_{LP} * p_{HP}}\ \text{[bar]} \tag{4}$$

$$\dot{m}_{Innj} = \dot{m}_{vp1} * W\ [kg/s] \tag{5}$$

2.2. Single-phase heat exchangers

In the single-phase heat exchanger, the desuperheater and the internal heat exchanger are included. The desuperheater is not used in the different simulation, since Bergland (2015) concluded that the desuperheater have a small impact on the system. However, since a simulation tool is going to be build it is included in the EES script. The single-phase heat exchanger is modelled with countercurrent flows and the heat transfer rate is calculated from equation 6 and 7

$$\dot{C} = \dot{m} * C_p\ \text{[kW/K]} \tag{6}$$

$$\dot{Q} = \varepsilon * \dot{C}_{min} * \left(T_{hot,in} - T_{cold,in}\right)\ [kW] \tag{7}$$

2.3. Absorber and Desorber

Both the absorber and desorber is also modelled as countercurrent heat exchanger. The pinch point in an absorber with ammonia/water as working fluid occur somewhere in the middle of the absorber (Nordtvedt, 2005). Therefore, to ensure heat transfer through the whole absorber, it is divided into 50 segments, where the energy equation is applied on each segment. To avoid that the temperature difference between the rich solution and sink is too small the mass flow rate in the sink is regulated. In the desorber the pinch point occurs either at the outlet or the inlet. Therefore, the desorber is modelled so that the minimum temperature differences in the desorber, which is given as an input parameter occurs both at the outlet and inlet of the desorber with adjusting the mass flow rate in the sources.

2.4. Other components

The solution pump is assumed to be perfect except from the motor efficiency. Moreover, the expansion valve reduces the pressure as an isenthalpic process. In the separator the fluid is separated into pure liquid and pure vapour and the ratio between the mass flow rate of liquid leaving the separator and the mass flow rate of fluid entering the separator is given in equation 8.

$$CR = \frac{\dot{m}_{liq}}{\dot{m}_{rich}} \text{ [-]} \tag{8}$$

3. Results

3.1. Compressor arrangement

To decide which compressor arrangement that suits the test rig best four different cases are made with different input parameters. Table 2 shows the results for case 1, which compare the different compressor arrangement with a maximum discharges temperature at 170 ^{0}C, except the one-stages that have a maximum discharges temperature at 271 ^{0}C and the reason for this, is that the absorber pressure get too low, if the discharges temperature at 170 ^{0}C should be achievable. Therefore, the temperature difference is too small in the absorber. The heat transfer rate to the sink is set to 250 kW and circulation ratio is 0.5, while the desorber pressure is set to 2.2 bar. For the injection configurations the injection ratio is set to 0.1. Once can observe from Table 2 that injection from rich solution obtain the highest COP, while the IHEX option 2 have the lowest COP but gets the highest sink temperature.

Table 2: Results for case 1

Parameter	One-stages	Injection lean	Injection rich	IHEX option 1	IHEX option 2
$COP_{heating}$[-]	3.246	3.293	3.527	3.237	2.446
T_{WA3} [^{0}C]	62.65	106.55	92.85	117.85	131.85
p_{abs} [bar]	13.5	16.45	14.81	19.51	29.47

In case 2 the compressor discharges temperature and COP is going to be evaluated with the same sink temperature for all configurations. Therefore, the different from case 1 is that the outlet sink temperature is set to 100 ^{0}C instead of the discharge's temperature. Table 3 show the results for case 2 there IHEX option 2 achieved the lowest discharges temperature, while IHEX option 1 obtained the highest COP and the absorber pressure is almost the same for all configuration.

Table 3: Results for case 2

Parameter	One-stages	Injection lean	Injection rich	IHEX option 1	IHEX option 2
$COP_{heating}$[-]	2.955	3.453	3.359	3.751	3.665
T_2 [^{0}C]	295.45	160.05	180.15	147.25	134.15
p_{abs} [bar]	15.02	15.46	15.76	15.65	16.02

Case 3 is going to evaluated how the circulation ratio is affecting the results. Therefore, the only input parameter, which is changes from case 1 is the circulation ratio and it is set to 0.6. From Table 4 the IHEX option 2 also this time achieve the highest sink temperature and the injection from the rich solution obtained the highest COP, while the IHEX option 2 had by far the highest absorber pressure.

Table 4: Results for case 3

Parameter	One-stages	Injection lean	Injection rich	IHEX option 1	IHEX option 2
$COP_{heating}$[-]	3.245	3.279	3.469	3.398	2.198
T_{WA3} [^{0}C]	102.65	109.45	103.65	111.45	128.05
p_{abs}[bar]	13.5	16.45	15.13	17.5	29.47

To see how the injection ratio and compressor discharges temperature is affecting the system case 4 is made. There the injection ratio is increased to a value at 0.2 and the discharges temperature is set to 220 ^{0}C. Moreover, compared to case 1 the circulation ratio is changes to 0.4 and the desorber pressure to 3 bar, while the heat transfer rate to the sink was set to 100 kW. As seen in table 5 the IHEX option 2 have the highest absorber pressure and sink temperature while, the one-stages have the highest COP but a discharges temperature at 265 ^{0}C. In this case injection from lean solution having the second highest sink temperature.

Table 5: Results case 4

Parameter	One-stages	Injection lean	Injection rich	IHEX option 1	IHEX option 2
$COP_{heating}$ [-]	3.123	1.980	2.226	2.358	1.507
T_{WA3} [^{0}C]	86.65	157.05	138.75	150.3	179.85
p_{abs} [bar]	18.00	40.94	34.57	39.90	78.4

3.2. Operational boundaries condition

The injection ratio, pressure ratio, desorber pressure and, circulation ratio is parameters that is going to be evaluate for the injection from lean solution compressor arrangement in this chapter. Mayekawa come with a suggestion on a MYCOM 125 L screw compressor with a VI at 3.65. This give an optimal pressure ratio at 5.83. When evaluating the injection ratio, the desorber pressure is 3 bar and the mass flow rate is 0.132 kg/s, while the circulation ratio is 0.5. Figure 5 show how the injection ratio influences the compressor discharges temperature, outlet sink temperature and COP for the VI 3.65 compressor. The highest COP and lowest discharge temperature were found at an injection ratio at 0.25, while the highest sink temperature was found at 0. Figure 6 show how the vapour quality at the injection stages in the compressor and the outlet of the compressor is affected by the injection ratio, if it is assumed that everything is mixed at the injection stages.

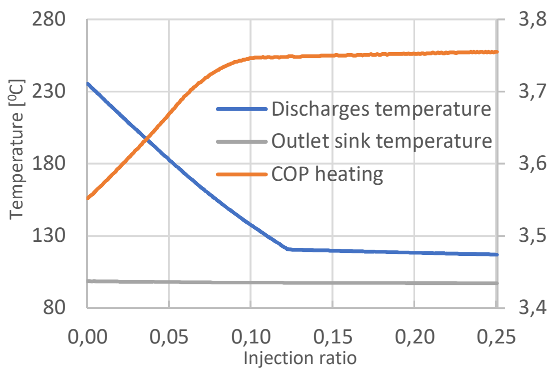


Figure 5:Compressor discharges temperature, Sink temperature and COP as a function of injection ratio for VI 3.65 compressor

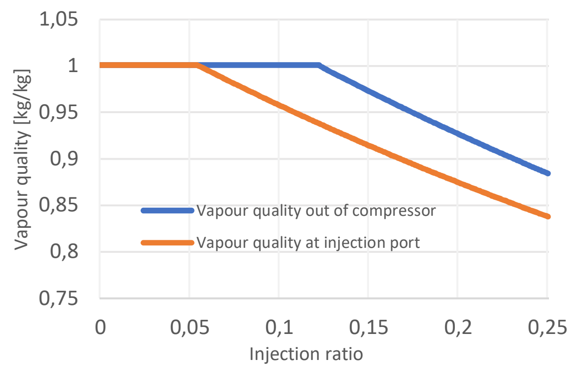


Figure 6: Vapour quality at the injection port and out of the compressor as function of injection ratio for VI 3.65 compressor

Another suggestion from Mayekawa were to use the same compressor with a VI at 5.8 this gave an optimal pressure ratio at 10.96. From Figure 7 the highest COP is obtained at injection ratio at 0.17 and the discharges temperature reduces drastically up to an injection ratio at 0.22, while the outlet sink temperature is not to much affected by the injection ratio. Both the outlet sink temperature and discharges temperature is higher than for VI 3.65 compressor, while the COP is lower. As seen in Figure 8 the vapour quality is higher than for VI 3.65 compressor for high injection ratios.

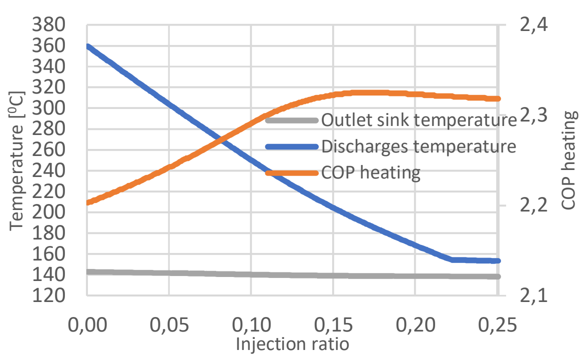


Figure 7:Compressor discharges temperature, Sink temperature and COP as a function of injection ratio for VI 5.8 compressor

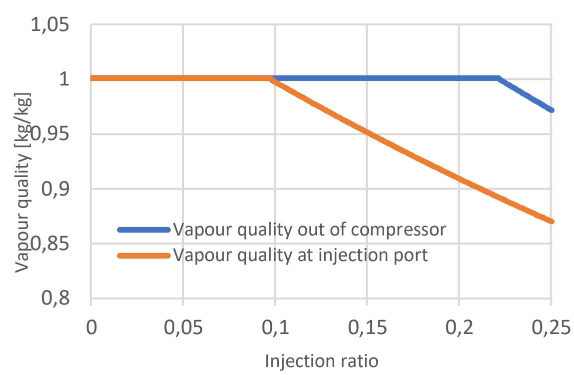


Figure 8:Vapour quality at the injection port and out of the compressor as function of injection ratio for VI 5.8 compressor

To evaluate the circulation ratio for the VI 3.65 compressor the injection ratio is set to 0.1 and the other inputs parameters is unchanged. From Figure 9 the highest COP is obtained at circulation ratio on 0.5 and the highest temperature is achieved at a circulation ratio on 0.55 and 0.95. In Figure 10 it is possible to see how the heat transfer to the sink, compressor work and pump work is affected by the circulation ratio.

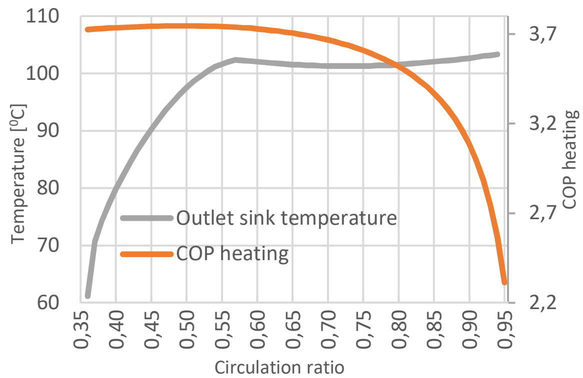


Figure 9: Outlet sink temperature and COP as a function of circulation ratio for VI 3.65 compressor

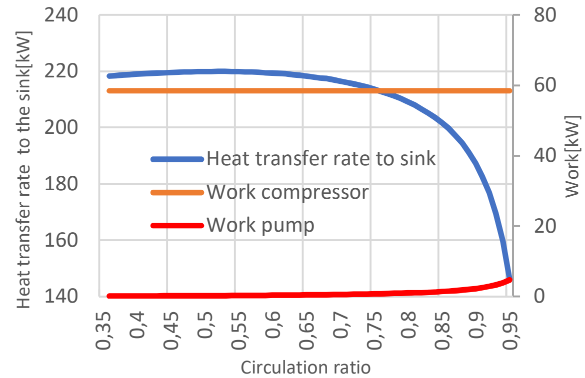


Figure 10: Heat transfer rate to the sink and COP as a function of circulation ratio for VI 3.65 compressor

The evaluation of how the desorber pressure are affecting the system with a VI 3.65 compressor and with an injection ratio on 0.1 could be seen in Figure 11. The inlet mass flow rate in the simulation is changes based on the changes in density and the averages swept volume for the compressor. As seen in Figure 11 both the outlet sink temperature and compressor discharges temperature are increasing with increase desorber pressure, while the COP is decreasing.

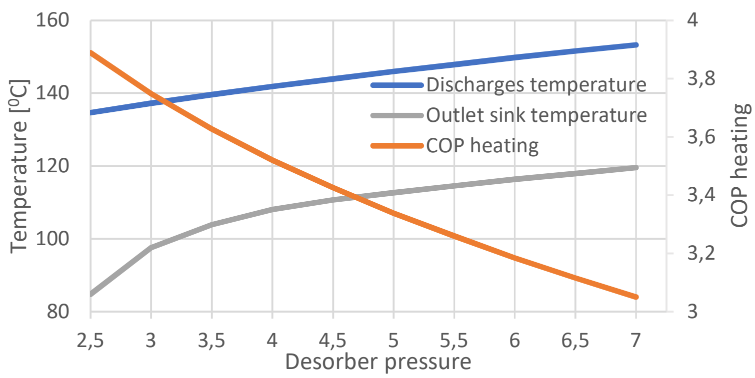


Figure 11: Outlet sink temperature, compressor discharges temperature and COP as a function of desorber pressure for VI 3.65 compressor

4. Discussion

4.1. Compressor arrangement

As seen from Table 2 to Table 5 the injection from rich solution and IHEX option 2 have the highest COP for almost every cases but the differences in COP between those two configuration and injection from lean solution is small. On the other hand, IHEX option 2 had a lot lower COP in almost every cases. This result is consistent with the findings of Jensen (2015), which does not investigate the configuration with injection from lean solution. On the other hand, the IHEX option 2 have the highest temperature out of the sink for all cases or the lowest compressor discharges temperature for case 2. The IHEX option 1 had the second highest sink temperature for all cases except case 4 there the injection from lean solution had. This indicate that the injection ratio has a huge impact on the system. This is not consistent with Jensen (2015), which had the lowest discharges temperature for IHEX option 1. The problem here is that Jensen (2015) inject down to saturation point and between two compressors, so the amount of liquid, which is injected are different. Therefore, it is hard to compare the injection from rich solution with each other. However, Jensen (2015) achieved lower discharges temperature for IHEX option 1 than IHEX option 2, which is the opposite of result from Table 3. The reason that the discharges temperature is lower for IHEX option 2 in Table 3 is that the fluid cooling down

the vapour is colder in IHEX option 2 because the lean solution has not been heated up with the heat exchanger from the rich solution first. Therefore, with about the same pressure ratio IHEX option 2 get a lower discharges temperature, since the vapour between the compressor is colder.
On the other hand, Jensen (2015) also mentioned that the liquid injection compressor may have lower investment cost and the need of spaces will also be reduced.

4.2. Operational boundaries condition

From Figure 5 and Figure 7 both the highest COP and lowest compressor discharges temperature is obtained with high injection ratios without affecting the outlet sink temperature too much. On the other hand, as seen from Figure 6 and Figure 8 the vapour quality in the compressor and out of the compressor are increasing with rising injection ratio, so the liquid content will be bigger in the compressor. As Bakken et al. (2018) concluded, that more liquid in the compressor lead to a lower efficiency of the compressor and this is not taken into account in the model. This will lead to a lower COP for the system and especially for high injection ratios. Compared to Lee et al. (2015), which achieved the highest COP at an injection ratio on 0.1 the COP for the VI 3.65 compressor in Figure 5 start flatting out around an injection ratio on 0.1 and not decreasing drastically as in Lee et al. (2015) experiment. The reason for this is most likely that the efficiency of the compressor in the model is not affected by the liquid content.
As seen in Figure 9 the highest COP is achieved at a circulation ratio on 0.5 before it reduces and compared with Nordtvedt (2005), which achieved the highest COP at circulation ratio on 0.4 before it reduces the differences is not too big. However, Jensen (2015) concluded that the optimum circulation ratio is dependent on the type of system and operating condition, so to find the optimal circulation ratio for a given system the simulation model should be run with the given parameter for the given system. Therefore, to decide the optimum circulation ratio both the COP and outlet sink temperature should be considered.
From Figure 5 and Figure 7 it is possible to see that the discharges temperature is much higher for the VI 5.8 compressor than the VI 3.65 compressor. The main reason for this is that the pressure ratio is higher for the VI 5.8 compressor and to achieve a discharges temperature under 180 ^{0}C, which Nekså et al. (1998) recommended the VI 5.8 compressor need an injection ratio higher than 0.18. As seen from Figure 8 this will lead to a high liquid content in the compressor. The injection ratio will most likely have to be higher than that as well since some of the liquid is going to be used as a lubricant and to sealing. A problem with the high injection ratio is the time of evaporation, since the amount of time the vapour is in the compressor is short because of the compressor speed and length. This may lead to that the temperature of the vapour leaving the compressor will never be lower than 180 ^{0}C for the optimal pressure ratio, in Addison to the efficiency of the compressor will be reduced with the high liquid content. As seen from Figure 11 the desorber pressure should try to be as high as possible to achieve the highest possible sink temperatures, but the limitation of how high pressure the components could handle should be evaluated. Because with increased desorber pressure the absorber pressure will increase more.

5. Conclusion

The injection from lean solution seems to be the best compressor arrangement for this test rig with relatively high COP and sink temperature. In Addison, to be a cheaper solution than the IHEX configuration and the need of spaces is less. The VI 3.65 compressor suit the test rig best and the optimal circulation ratio is between 0.55-0.6 for this simulation with an outlet sink temperature at 102.25 ^{0}C and COP at 3.738. For the same compressor the optimal injection ratio looks to be between 0.07-0.12. Changes in desorber pressure show a huge impact on the COP and temperature out of the sink.

6. Further work

The evaporation process in the compressor is an uncertainty in the model, so to do experiment measurement to find a correlation that suit the evaporation process and then implementing into the model should be prioritized. Compared the result from the model with the measured value of the test rig. An interesting optimization of the test rig could be to use an ejector to mix the vapour and lean solution at the inlet of the absorber and this may reduce the compressor work.

7. Nomenclature

Latin letters

$\dot{C}$	Capacitances rate	[kW/K]
C_P	Specific heat capacity	[kJ/kg*K]
CR	Circulation ratio	[-]
K	Correction factor	[-]
$\dot{m}$	Mass flow rate	[kg/s]
P	Pressure	[bar]
PR	Pressure ratio	[-]
$\dot{Q}$	Rate of heat transfer	[kW]
T	Temperature	[K or ^{0}C]
W	Injection ratio	[-]

Greek letters

ε	Thermal efficiency	[-]
Δ	Difference	[-]
η	Efficiency	[-]

Abbreviations

CFC	Chlorofluorocarbon
COP	Coefficient of performances
EES	Engineering equation solver
HCFC	Hydrochlorofluorocarbon
IHEX	Internal heat exchanger
VI	Volume ratio

Subscript

1	Inlet
2	Outlet
3	Outlet sink
abs	Absorber
cold	Cold
com	Compressor
des	Desorber
hot	hot
HP	High pressure
ihex	Internal heat exchanger
in	Inlet
innj	Injection
isen	Isentropic
LP	Low pressure
min	Minimum
mot	Motor
MP	Intermediate pressure
rec	Reciprocating compressor
Rich	Rich solution
Screw	Screw compressor
Sou	Sources
volu	Volumetric
Vp1	Inlet compressor
wa	Sink

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