Analysis and optimization of Heat Pumps for integration of datacenters in District Heating

Master Thesis

Iñaki Muguruza Fuentes

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DTU Mechanical Engineering
Department of Mechanical Engineering

PROJECT INFORMATION

Title	Analysis and optimization of heat pumps for integration of datacenters in district heating
Student	Iñaki Muguruza Fuentes
Supervisors	Brian Elmegaard
	Pernille Hartmund Jørgensen
	Torben Schmidt Ommen
Company relation	HOFOR
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Abstract

This thesis investigates the integration of large Heat Pumps for the Datacenter cooling, taking profit from the exceeded heat with the District Heating Network, increasing its temperature from 45ºC to 85ºC. Concretely, is planned to help the Danish Government with the emission reduction objectives set by 2050.

The project is particularly focus on a specific datacenter placed in Kassø, which has a cooling necessity of approximately 27MW. For this facility is studied the best option in terms of heat source, Air or Water, and for other components such the compressor type. The result revealed that water as source is preferable, and this option is evaluated for the operation with one-stage and two-stage Heat pumps, looking for some new configurations that can increase the performance of the system.

The pinch method is applied to the resulting best options, and an optimal pinch point will be found to obtain the best efficiency reachable, and an improvement of the 2.84% and 3.76% in the COP is showed as a result of the pinch optimization for the one and two-stage Heat pumps, respectively. For these cases, is necessary to design a Heat Exchanger Network able to fulfil the requirements of the optimization. Therefore, two HEN are designed for the best ammonia configurations, for the case of the one-stage, is not possible to achieve the temperature difference, set as 3K, and for the two-stage 2.93K is reached with 18 exchangers, considering the HEN valid for its implementation.

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List of Abbreviations

CON	Condenser	
COP	Coefficent of performance	
DC	Datacenter	
DH	District Heating	
DSH	Desuperheater	
EB	Energy Balance	
E	Evaporator	
GWP	Global Warming Potential	
HEN	Heat Exchanger Network	
HP	Heat Pump	
H	High	
L	Low	
OC	Oil Cooler	
ODP	Ozone Depletion Potential	
SC	Subcooler	
η_{isen}	Isentropic compressor efficiency	
T	Temperature	ºC or K
P	Pressure	bar
h	Enthalpy	kJ/kg
$\dot{Q}$	Capacity	MW
$\dot{W}$	Power	MW
Cp	Specific heat	kJ/kg K
$\dot{m}$	mass flow	kg/s
S	entropy	kJ/K

Acknowledgements

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First at all, I would like to thank my supervisors Brian Elmegaard, Pernille Hartmund Jørgesen and Torben Schmidt Ommen the opportunity to perform a thesis in such an interesting field, and guide me across the path. Especially to Pernille for the guidance in the initial steps of the modelling.

This thesis is dedicated to my family, who allow me to be this time in Copenhagen economically, and support me emotionally as well. Also I want to dedicate it to the wonderful friends I met during this months, Charly, Mario, Alfonso, Alex…and for sure to my thesis colleague Fabio, who has been a strong support.

Chapter 1: Introduction

1.1 MOTIVATION

Climate change is a global challenge to be taken seriously. Science has said that global emissions of greenhouse gases must peak as soon as possible, and by no later than 2020, if temperatures are not to increase by more than two degrees compared with pre-industrial levels. In 2050, the emissions coming from the developed countries should be cut down by 80%-95% in comparison with the 1990 levels, a target the EU has adopted as part of a joint global climate change mitigation efforts. As part of this, oil for heating purposes and coal are to be phased out by 2030, and electricity and heating supply is to be 100% covered by renewable energy by 2035. [1]

Therefore, in the last years Denmark has compromised to reduce his total amount of fossil emissions, expecting to have a 100% renewable energy supply by 2050. This fact implies that there many things that should improve to reach that objective.

In that scenario it is found one way to contribute to this change, the use of large heat pumps to supply heat through electricity, and particularly, the project will emphasize in a specific type of building, a datacenter. A datacenter is a construction that have inside a lot of informatics servers, which reject heat and need to be cooled, and Denmark it is an excellent location for being constructed.

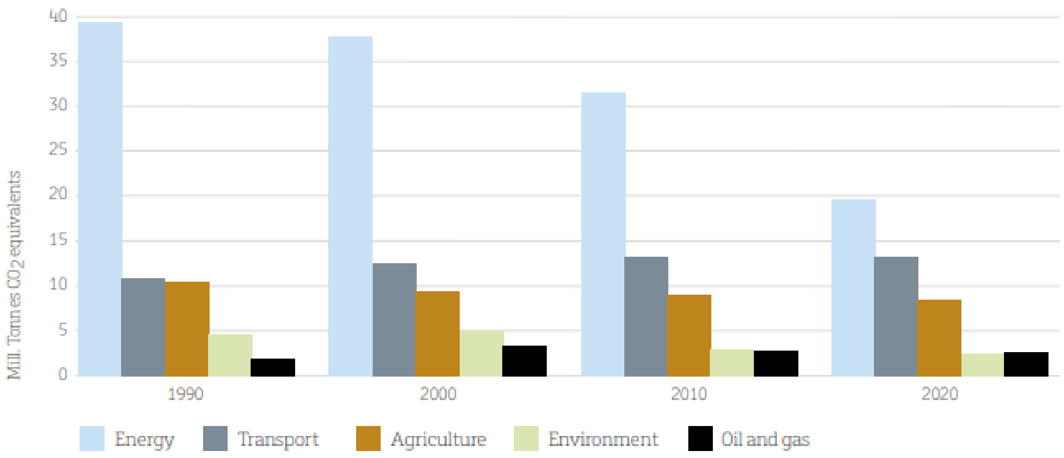
Recently the number of datacenters in Denmark is rising quickly, and such an example, there are a lot of different companies that have already acquired surfaces to build datacenters or directly have installed one in this moment, for instance Apple, Facebook and Google.

Apple is constructing a 166,000 m^2 datacenter near Viborg. A second datacenter is planned near Aabenraa. Facebook selected our third largest city, Odense, for a 55,000 m^2 datacenter. Google has acquired 73 hectares of land in Fredericia and 131 hectares in Aabenraa for potential new datacenters. [2]

The ministry of economic and business affairs has estimated that Denmark's electricity consumption will grow by about 24% by 2025, with datacenters responsible for about half of the increase. This fact taking in account that about 40% of a datacenter's energy consumption is used to cool down the servers [3], makes this project interesting, due to the high projection growth in terms of energy consumption and the huge potential saves that could be profit the cool necessity to full fit the district heating demand.

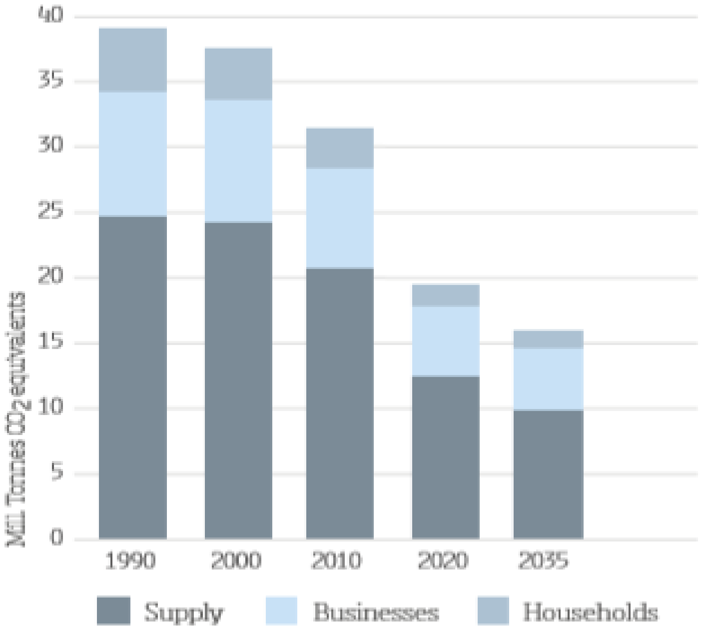
1.2 BACKGROUND

Following with the purpose of energy supply and transport to be full composed by renewable sources in 2030, this requires new initiatives for the period after 2020, for this reason Danish government has introduced other objectives to finish as well, such as oil for heating purposes and coal are to be phased out by 2030, or the reduction of the energy sector, which is the most influent in the greenhouse gases emission. It is estimated that the energy sector will have to be reduce to almost 50% of the emissions comparing with the levels of 1990s [1], and this implies that the source has to move out to renewable, and at the same time the consumption has to be decreased by means of an efficiency improvement.



1.1. Historical and projected Danish greenhouse gas emissions for 1990 to 2020 (Source: Danish Centre for Environment and Energy and the Danish Energy Agency) [1]

In the case that the energy sector is analysed with more detail, it could be found that historically, traditional energy consumption (households, businesses as well as electricity and heat production) is the guilty for most of the Danish greenhouse gas emissions. The supply sector, i.e. electricity and district heating production, is the one with more emissions, followed by individual use of fossil fuels for industrial processing and heating buildings [1]. Consequently, the conclusion is that the factor that should improve more it is the energetic supply, also has to be taken in account that the energetic demand will be higher each year due to the growth of the city and the industry.



1.2. Energy sector emission. [1]

Energy efficiency improvements are crucial to increasing the share of renewable energy in total energy consumption. The agreement includes that the energy saving obligations on energy companies are to be intensified significantly [1].

Regarding the heat production and distribution, it is mandatory to say that Copenhagen has one of most efficient and most successful heating grids in the world. It takes profit of the wasted heat, recovering a great amount of energy coming from incineration plants, and combined heat and power plants, among other things.

In order to achieve the targets that the Danish government has stablished, another public organization, the Danish council on climate change, has analysed 20 different elements of transition, to get clear which are the elements with a higher impact and which ones could be less relevant in the future.

Possible elements of transition within heat supply are HP, solar heating and energy renovation, meanwhile the analysis also includes factors that may contribute to reducing emissions from the industry's energy consumption. In addition, the Danish Council on Climate Change has compiled a package of seven transition elements according the 2030 reduction target. The package consists of energy renovation of the building stack, individual heat pumps, large heat pumps within district heating, solar heating, energy efficiency improvements, natural gas within heavy transport and acidification of cattle and pig manure. The package focusses on energy, but also initiates the necessary renovation in sectors such as the agriculture or transport, which are influenced by the old vehicles with a big contribution to greenhouse gases. Obviously, the exact figure is uncertain, but it shows that Denmark could be able to meet its 2030 targets with low or no costs [4].

1.3. Potentials, costs and 2050 perspective of the elements of transition [4].

In the last graph, it can be observed that there are different extension levels, and also the technologies are separated depending on the socioeconomic cost that probably will have. Regarding the size of the individual circles, it indicates the given element's potential for greenhouse gas mitigation in the period 2021-2030 in tonnes of CO2 equivalents. The position of the circles in the individual spaces is not significant.

Large Heat Pumps are located in the 'easy way' towards the transition of 2050, and it is categorized as cheap, so could be an attractive solution to help with the emission reduction, and this figure does not take in account the possibility of using it combining with the datacenters and the district heating grid, because of that this project has a lot of potential if finally this idea could be developed and implemented in the future.

Following this line of progress, where changes should happen in many fields, and therefore, a great investment it is required, new agreements will take place to reach the 100% renewable supply by 2050.

As an example, in 22 March 2012, an energy agreement earmarked a total of EUR 13.3 million until 2016 to support development and use of new renewable energy technologies for electricity production, and a pool of EUR 4.7 mill. to promote new renewable technologies for heating, e.g. geothermal energy and large heat pumps [1].

That agreement made in the last years it is a huge motivation for new projects to be developed in the future, having the support of the Danish government and an economic background to develop new ideas and concepts.

Moving to another key point of the project, the evolution of the datacenters in the next years, Danish government has the ambition to become a leading Nordic and European location for hyperscale datacenters continues to gain momentum by attracting major industry actors and government support. The country is offering access to a low-cost and expanding renewable energy base to lure global companies such as Facebook, Google and Apple [3].

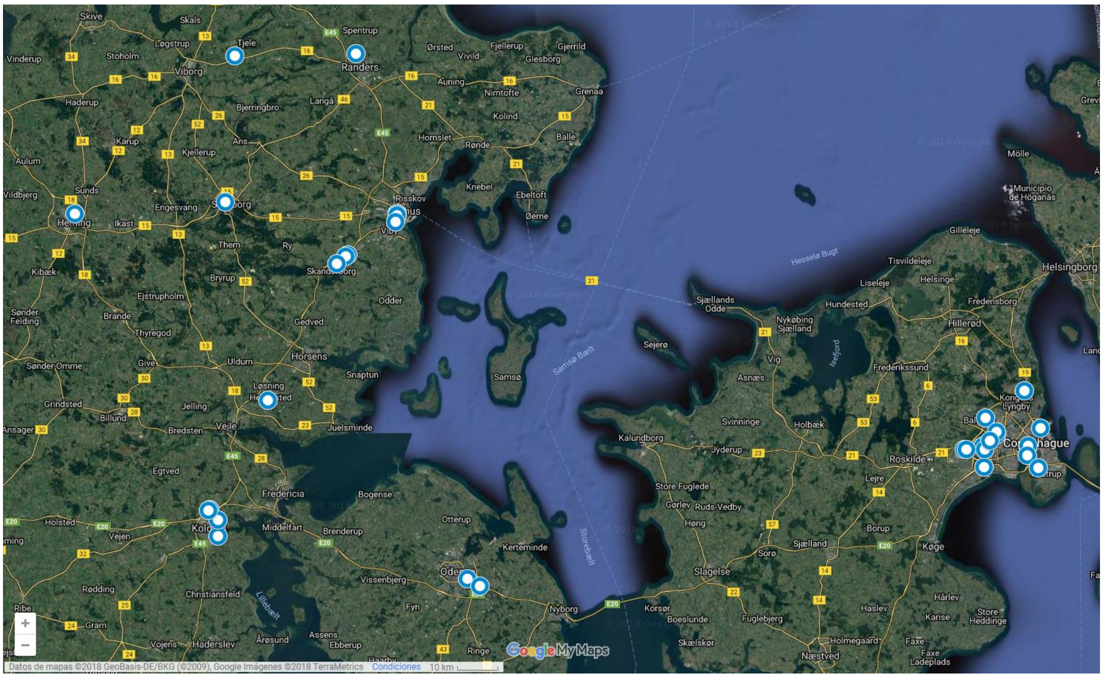
Denmark's Nordic hub initiative is an integral part of the government's digital industry and technology development plan for 2020-2030. The plan, aims to create a strong underlying infrastructure, positioning Denmark in the hub for datacenters in Europe. Provisional estimates, based on data collated by the finance ministries of Denmark and Sweden, indicate that the Nordic datacenter construction market could be worth between €4bn and €5bn by 2024. The projected surge in the number and scale of datacenters is certain to have a dramatic impact on the Danish economy [3].

Some studies and data show that Denmark has an optimal conditions to place datacenters, due to:

- ✓ A reliable power grid with an uptime of 99.99% and 80% of power lines underground.
- ✓ A mild climate that allows low-energy cooling all year round.
- ✓ Good access to large-scale sites due to low population density.
- ✓ 72% of the Danish power supply comes from renewable sources.
- ✓ Reuse of waste heat for district heating, which warms 64% of all Danish homes.
- ✓ Best in Europe when it comes to ease of doing business and dealing with construction permits, according to the World Bank.
- ✓ High-speed global connections.

In addition to all that, the low latency network gives a high-speed connection to Central Europe, Ireland, the UK and the US. High-capacity power connections link Denmark with Norway, Sweden and Germany. Connections to the Netherlands and the United Kingdom are in the pipeline [2].

In the next image, the current location of the datacenter already placed in Denmark can be observed. The majority of those are placed in the Copenhagen area and surroundings, but the biggest ones (also called hyperdatacenters) are located near Odense, which correspond to Facebook, and the other in the surroundings of Viborg, belonging to Apple.



1.4. Map of datacenters in Denmark [5]

In order to have an estimation of the expected growth in the world, the increase of the total worldwide enterprise storage systems factory revenue was up 2.9% year over year and reached $10.8 billion in the second quarter of 2017, according to the International Data Corporation [6].

Furthermore, basing on the worldwide historical increases of datacenters in the period 2005 to 2010, it is know that even with negative factors involving such as the financial crisis of 2008, the electricity used by datacenters in the world increased by 56% and in the period of 2000 to 2005 that the double amount of electricity consumed doubled. Also, another significant point is the electricity share, comparing the used by datacenters and the total electricity use in the world, which in 2010 was between 1.1% and 1.7% [7].

In summary, the rapid rates of growth in data center electricity use that prevailed from 2000 to 2005 slowed significantly from 2005 to 2010, yielding total electricity use by data centers in 2010 of about 1.3% of all electricity use for the world, following with that consumption, it is estimated that in 2014 the share was of 1.62% and nowadays it is around the 3% [8], which implies that in the last years the capacity and amount of datacenters has been huge, and therefore, maybe new problems will arrive.

The problem could be that if the increase of electricity it is not supplied by renewable energies, the pollution in the electricity transformation will make datacenters one of the biggest polluters in just the seven next years, because it is predicted that in 2025 datacenters will consume until 1/5 of the global electricity [8]. That means that the quantity of renewable energy should grow as fast as the electricity consumed, or maybe in other way, just make the current systems much more efficient, for example, taking profit of large heat pumps to cool down the datacenters, which is part of that buildings with the highest electricity consumption, and at the same time, heat up the district heating water. Investigation showed that energy consumption by cooling datacenter Information Technology equipment is between 30% and 55% of the total energy consumption. Cooling and ventilation system consumes on average 40% of the total energy consumption in a datacenter [9]. Therefore, a reduction in the cooling consumption meanwhile the heat is profited at the District Heating circuit looks like an optimal solution in this case.

Despite this, the implementation of large heat pumps it is very complex and it has different factors, such as the limited knowledge in terms or performance or the variation in the operational conditions that could change the feasibility of the facility, comparing with a traditional combined heat power plant.

For this reason this project wants to study and perform different models of heat pumps, working with distinct configurations, operational conditions and heat sources, in order to contribute to the incoming knowledge of this field in the future.

1.3 OBJECTIVES

The purpose of this thesis is to investigate the benefits that a Heat Pump connected to a datacenter could introduce to the District Heating line, producing at the same time a profit for the cooling of the datacenter and for the District Heating. Due to it is an idea with a low background, specific configurations and the pinch method will be applied to reach the highest performance as achievable.

- *"Which heat source could be better, Air or Water, in terms of efficiency, capacity and logistic?"*
- *"How will affect to the COP the implementation of new configurations?"*

- *“How will the pinch method be applied to a Heat Pump?”*
- *“How will be the Heat Exchanger Network in order to make feasible the model?”*

To achieve the answer to this questions, the modelling of different Heat Pumps it is going to be developed modifying some parameters and looking at the results to check that the models could be carried out in the reality, making the COP the most important variable through this project, in favour of the maximum performance.

1.4 THESIS OUTLINE

In this section a short review of each chapter in this report it is going to be advanced, to construct some sort of readers guide inside the thesis.

Chapter 2: 11Heat Pumps explains which are the most important concepts to understand how a heat pump works, that are essential to understand the models and the work that has been done.

Chapter 3: Case Description introduces the current state of the art for heat pump integration with datacenters, and explains the basic case for the modelling, where the inputs come from and other things that will be developed later.

Chapter 4: Thermodynamic model presents the modelling phase with the main equations and the assumptions taken for the simulations carried.

Chapter 5: Pinch Method contents what it is the pinch analysis in general firstly, and then the integration of it to a Heat Pump, attaching the example of its application made in this project.

Chapter 6: Results addresses all the results obtained at the simulations, and the explanations of the elections made. In addition, the cause of every result is developed to have an idea of how the system works.

Chapter 7: Discussion reviews the assumptions and the state of the art of the project, aiming to a possible future work to follow the study carried

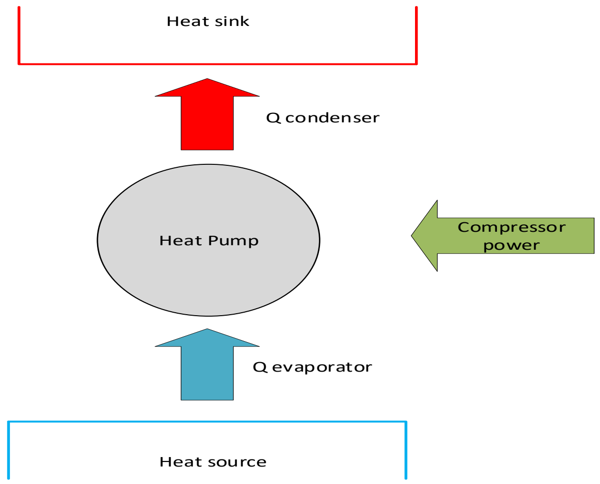
Chapter 8: Conclusions summarizes the main results and considerations of the thesis.

Chapter 2: Heat Pumps

In this chapter there are going to be explained all the principles and knowledge that should be known to understand the modelling and optimization of a Heat Pump.

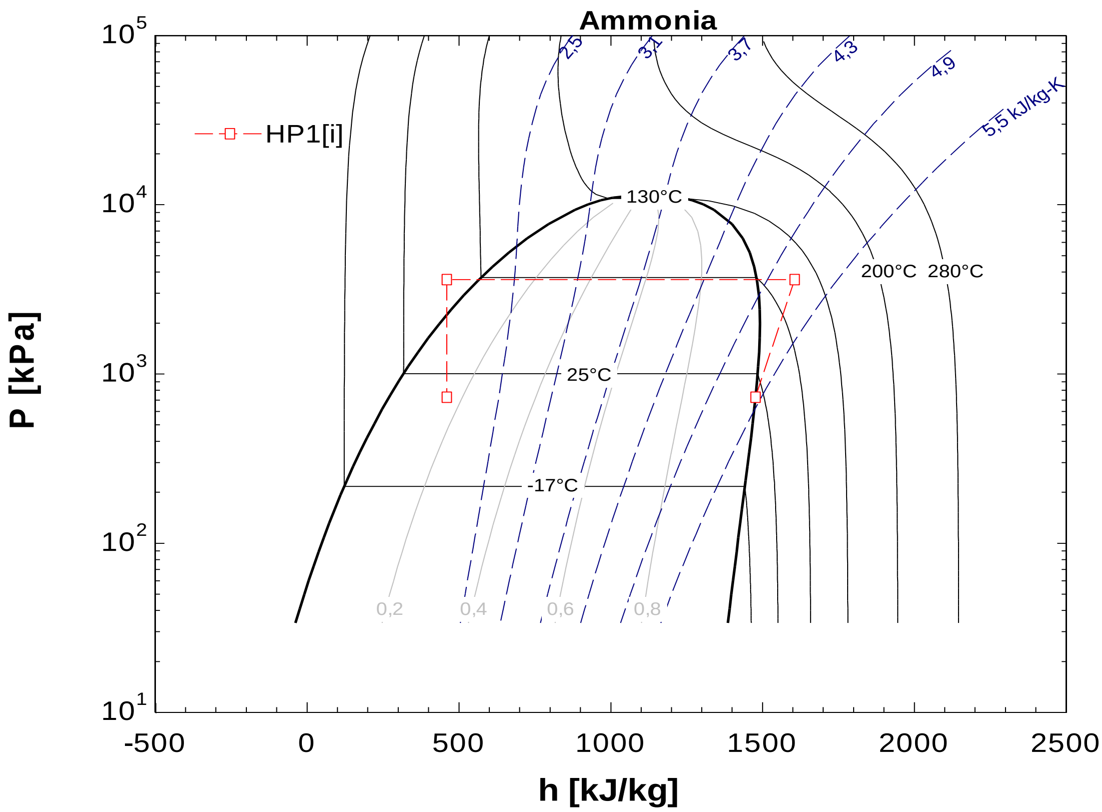
2.1 WORKING PRINCIPLE

A Heat Pump it is a machine that can take profit of a waste heat excess flow, and use it to heat up another flow to a high temperature. A heat pump always have two parts, the source and the sink, the source it is the part which give the required heat to the HP, and the sink it is the opposite section, where the heat is profited to increase the temperature of the flow.



2.1. Heat Pump working principle.

The working principle it is based on the compression and expansion of a working fluid, also called refrigerant. The compression phase make the fluid increase in pressure, temperature and enthalpy, so the refrigerant in that step has a high potential for being used and warm up another flow of directly the ambient. The exchange it is produced at the condenser, where the fluid arrives to saturated liquid, and then, turns back to the initial pressure by means of an expansion valve. The cycle will continue absorbing the heat from an external flow in the source, or evaporator in this case, where the refrigerant arrives to saturated vapor and closes the cycle entering the compressor again after that. This process can be followed in the image below.



2.2. P-h diagram for a simple vapor cycle

One of the most important variables to take in account it is the Coefficient of Performance (COP), which is an a dimensional factor that measures the performance of the Heat Pump, it can be calculated in many ways, but can be defined as the total useful energy divided by the power required: $COP = \frac{\dot{Q}_{useful}}{W_{total}}$

2.2 REFRIGERANTS

The refrigerant it is the working fluid that operates inside the Heat Pump and it is submitted to compression and expansion phases. The refrigerant in the compression cycle strongly determines different parameters, such as the COP, the size of the compressor, the materials that could be used in the cycle, and also the necessary protections in the facility (gas escape detectors…).

For the refrigerants, exist some thermodynamic, technic, security, environmental an economic requirements that has to be accomplished, and none of the refrigerants it is able to accomplish all the desired requirements.

All the refrigerants are classified in families, depending on its components and the mixes that form between each other. The most important are:

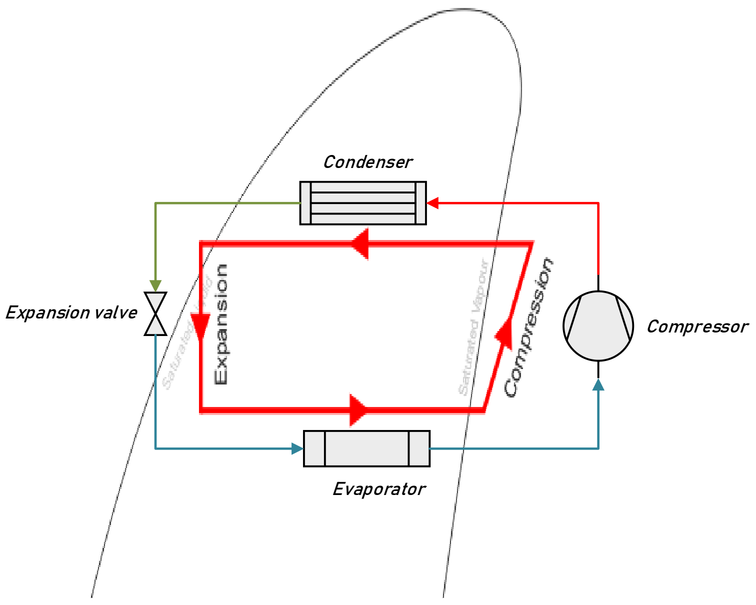
- CFCs: Organic compounds that contain chloride and Carbon atoms (R11), high Global Warming Potential (GWP) and Ozone Depletion Potential (ODP).
- HCFCs: Organic compounds with Hydrogen, Chloride and Fluoride atoms (R22). High GWP and low ODP
- HFCs: Organic compounds without Chloride atoms, but with Fluoride and Hydrogen atoms (R134a). High GWP and void ODP.
- Naturals: Fluids that are present in the environment naturally (Ammonia R717, Carbon Dioxide R744). Low ODP and void GWP.

Nowadays, CFC are forbidden and the HCFC since 1990's and cannot be replaced for its future use, meanwhile the HFC have also been forbidden and before 2030 have to disappear progressively, due to its high potential to harm the Ozone layer [10]. The target to achieve with this prohibitions are purely environmental, and challenges the new generation of machines, that should keep increasing on efficiency with natural refrigerants and olefines.

For this reason, the choose of the refrigerant has been a key point, and there are some evidences that point Ammonia as one of the best fluids to work with Heat Pumps, due to its critical temperature $T_{crit} = 132.3ºC$, also the cost of the Ammonia it is quite lower than the HFCs, and because of its lower density on liquid phase, a lower refrigerant mass flow it is needed [11]. However, Ammonia it is toxic in very ow concentrations in air but has a built-in warning system by its characteristic smell in much lower concentrations than the hazardous level. By last, Ammonia gives quite high temperatures out of the compressor and therefore two-stage systems or liquid injection are common with Ammonia [12].

2.3 COMPONENTS

As it was explained before, the refrigerant goes through distinct phases, and for each change of its conditions, there is a specific component connected to each other and composing a cycle. Depending on the Heat Pump and the cycle to carry out many other components could appear to increase the performance or as a security measure, but the simplest elements will be explained next.



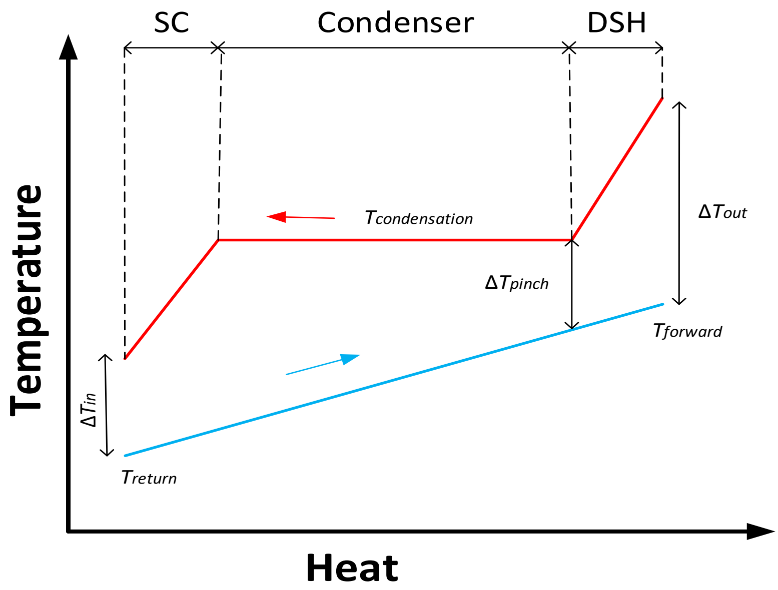
2.3. P-h Diagram for a simple HP.

2.3.1 Condenser

The condenser is basically a heat exchanger that allows the working fluid to decrease its temperature until the saturated liquid phase, meanwhile the heat obtained from the change of phase it is profited to increase the temperature of the sink. Such the condensation happens at a constant temperature, the heat that the condenser takes benefit is mainly latent heat, provided from the transformation of the refrigerant to liquid state.

Moreover, it is possible to decrease the refrigerant temperature below the saturation temperature, that strategy it is called subcooling, and it is usually employed. The addition of 10K as subcooling can give the cycle a non-negligible improve in terms of performance, around the 7% [13], furthermore, it is interesting to introduce subcooling in the condenser, because in that way it will be ensured that the refrigerant arrives to the valve on liquid state, avoiding cavitation problems.

Normally, three different sections in a condenser can be split. The first part is the desuperheater (DSH), where the working fluid at its hottest point is cooled down until saturated vapor state, the following one is the condenser, where as it was developed before, the refrigerant changes its phase to saturated liquid. Finally, after the condenser a subcooler (SC) can be added to increase the efficiency as was explained in the last paragraph.



2.4. T-Q diagram of a generic condenser

The condensing temperature it is usually obtained fixing a ΔT_{pinch} with the sink, and this fact makes the optimization really tricky, because there are a lot of parameters involved for just fixing a temperature difference at that point. Also the subcooling can be set as ΔT with the sink, adding complexity to the optimization.

In the particular case applied to the District Heating network, the sink will be water, which use to go from 40-50°C to 80-90°C. The return and forward temperature it is still a discussion in Denmark, and there is any conclusion of what temperature is the best to keep in the inlet and the outlet of the Heating facility.

2.3.2 Expansion valve

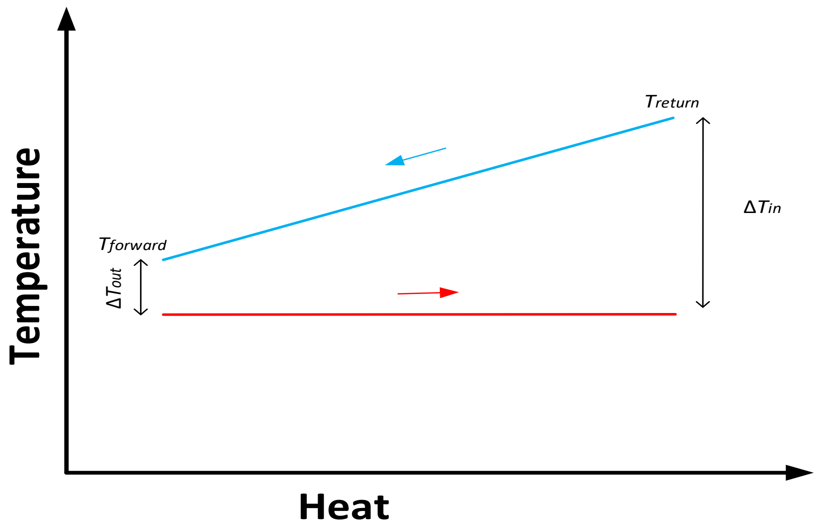
The expansion valve is the device that allows the working fluid to turn back to its initial pressure in the system, before being compressed. The expansion valves are assumed to be isenthalpic, which means that the enthalpy will always be the same in its inlet and outlet, even the pressure will be lower in the outside. In the inlet, the fluid has to be always on liquid state, to avoid cavitation and ensure the integrity of the valve, it is recommendable to have a subcooling in the refrigerant.

Likewise, there are a lot of types of valves, for instance manual, short tube, capillary tube, automatic, thermostatic, electronic and by last high and low pressure float valves, as it is not the purpose of the project, the types of valves are not to be explained, just the most common, that are the thermostatic valves. This valves try to keep a constant superheating at the evaporator, but the pressure losses in the evaporator could affect negatively to its effectiveness [13].

2.3.3 Evaporator

The evaporator is the opposite part of the condenser, in this side of the Heat Pump the refrigerant's temperature is increased until saturated vapor, obtaining the heat to feed this transformation from the Heat Source, cooling down other flow at the same time. Like in the condenser, in the evaporator it is assumed a constant pressure, even in a real case pressure losses can occur.

At the evaporator outlet, it is possible to increase a bit more the temperature, going through the saturated vapor line, and ensuring the vapor state at the inlet of the compressor. This strategy is called superheating, and could be useful or not useful, the useful superheat is the one that is desired to increase the COP and it is only at the evaporator outlet, the not useful is the one caused by the other components that eject heat, and it is in the suction line.



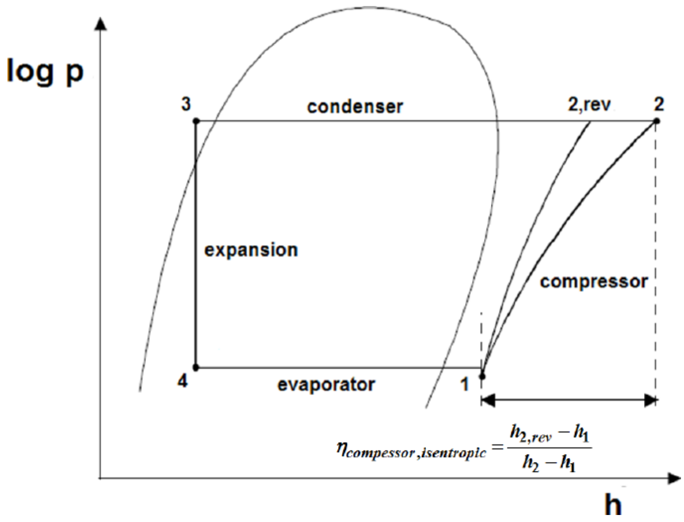
2.5. Evaporator T-Q Diagram.

In the specific case submitted to study, the heat source will be the datacenter. The fluid that provides heat to the refrigerant is a design decision and it will be developed after, but the choices are water and air. Particularly, it is known that the fluid will arrive at 32°C, and the forward temperature will depend on the heat source employed. The possibilities and results obtained with the sources will be developed and one decision regarding the best option in terms of COP will be taken.

2.3.4 Compressor

The compressor transfers the vapour from the evaporator to the condenser at a superheated state, increasing the pressure between both exchangers. The performance of a system is strongly influenced by the compressor, its efficiency, maintenance requirement and life expectancy [12].

When the compression stage is done, the refrigerant pressure is raised, at the same time the temperature and enthalpy of the fluid grow as well. In an ideal case, the compression is adiabatic and reversible, that means not have any heat exchange with the surroundings and any heat loss. But in a real case, the compression will make the outlet point move to the right in the P-h diagram.



2.6. P-h diagram of an isentropic compression [14].

The compressor is composed by an electric motor, that must be cooled somehow, for the hermetic compressor designs this is a special problem. The most common solution is cool the electric motor by means of the incoming refrigerant vapor. Other way to cool down the compressor and the refrigerant is with oil injection. Oil is needed for lubrication of bearings and other moving parts, but also can be used mixing it with the working fluid, and afterwards being separated of it. This process will decrease the temperature in the outlet of the compressor in the cycle. The refrigerant tends to be disolved into the oil, and the oil also is entrained by the refrigerant and carried away throughout the system, if for some reason the oil is trapped anywhere in the system and does not return to the compressor the lubrication will fail [13].

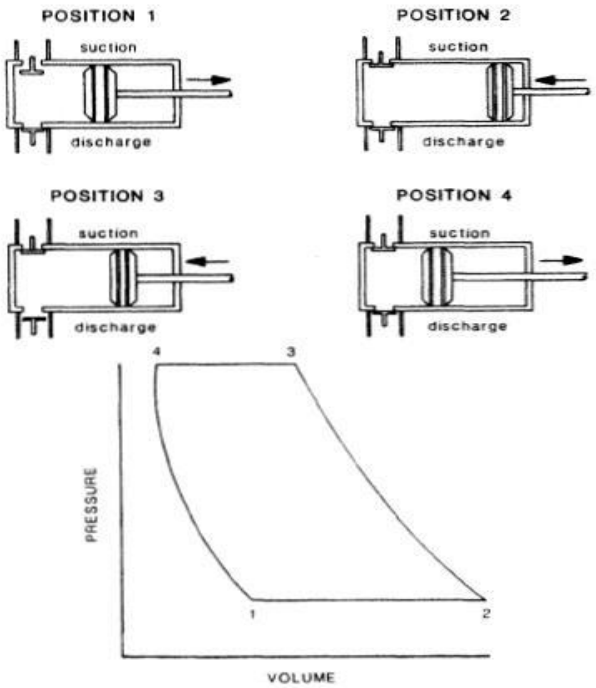
Following with the oil injection, for small systems is estimated than the 0,1-0,3% of the refrigerant mass flow is coming from the oil used as lubricantion, not as oil injection. Anyways, an oil separator is necessary to split the oil and the refrigerant for the systems using oil cooling [12]. Furthermore, the heat absorbed by the oil flow can be profited to feed other heat exchanger, making a normal cycle much more efficient.

By last, many types of compressors have been designed in the history, currently the most used for all the applications are the Reciprocating or piston, Screw, Centrifugal, Rotary and Scroll. Screw and centrifugal are used for large capacities, the piston can be used for everything but have lost applications for the largest sizes to the screw. The other are employed for small capacities [12].

Piston compressors

The first mechanical compressor that were built were of this type. Reciprocating compressors have a simple working principle and geometry, which is an advantage because it is easy to machine the components with good accuracy, but at the same time a piston compressor has a big number of moving parts and it is difficult or impossible have a model free of vibrations.

At piston compressors, it is an unavoidable part of the vapour that does not leave the cylinder when the piston is at the top position, this volume is called dead space [12]. In a normal cycle of this kind of pistons, the cylinder is filled with gas coming from the evaporator at first, with the cylinder at the top position, then the cylinder is moves and reduces the volume inside it, making the pressure increase. At this point the discharge valve is opened and the gas goes to the next component, the DSH.



2.7. Piston working cycle [15].

Piston compressors still are used for many applications, from the smallest sizes until flow rates about 500m^3/h.

Screw compressors

The principle and design it is based on two twin rotors with matching profiles, one male and other female. This two parts are the only moving in the compressor and therefore will not exist any dead volume inside it, moreover, there is not any valve in the design, instead of valves, and inlet and outlet gas flow is controlled by fixed ports. For this reason, screw compressor have a give built-in volume that will define the compression ratio. It is desirable to the outlet port to be as small as possible, in order to maximize a large compression [12].

The isentropic efficiency of a screw compressor it is strongly influenced by the pressure ratio, and it can be estimated with the next equation:

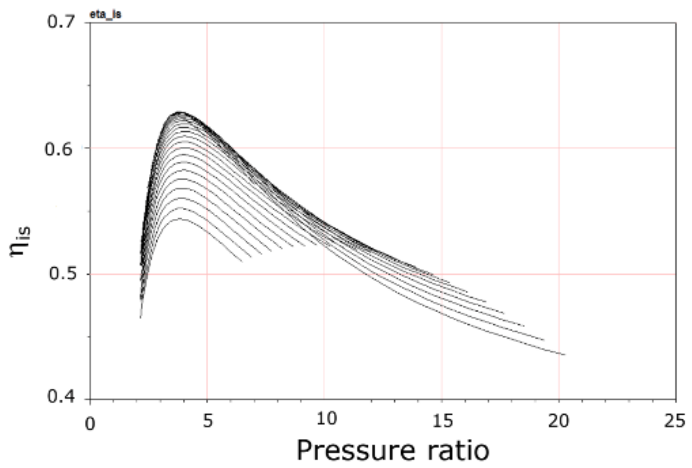
$$\boldsymbol{\eta_{isen}} = \frac{(\frac{p_1}{p_2})^{(k-1)/k} - 1}{\pi_i^{(k-1)/k} - \frac{k-1}{k}\left(\pi_i - \frac{p_1}{p_2}\right) - 1} \tag{2.1}$$

Where P_1/P_2 is the pressure ratio, π is the pressure built in ratio, deduced from the volume ratio raised to the polytrophic exponent k. As the pressure ratio increase, the isentropic efficiency decreases, depending on the refrigerant and the parameters of the compressor.

In the screw compressors oil can be injected in large quantities, in order to cool down the gas and preventing excessively high temperatures during the compression, the pressure ratio can be higher as well, due to the refrigeration of the gas. Oil injected screw compressors are dominating the market, specially for large capacities.

2.4 TWO STAGE HEAT PUMPS

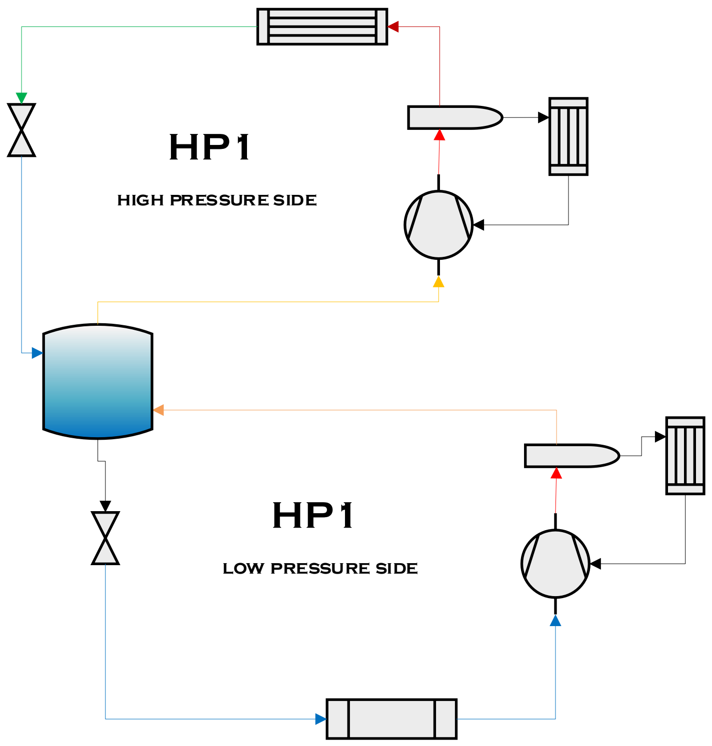
A two stage cycle is in some way an evolution of the simple vapor cycle. In a two stage there are two different compressors, and that fact allows the system to work in better pressure relation conditions, reducing the discharge temperature and making the compression closer to an isothermal case.



2.8. Isentropic efficiency with pressure ratio [16].

With a two stage cycle there are much more possibilities to improve the performance of the system, but the configuration that is going to be carried out it is the one with an oper intercooler. This type of configuration reduces the discharge temperature, increases the capacity and the efficiency of the system. An disadvantatge could be the cost of the control in the system, because is difficult to measure the quantity of volume inside the tank, and if some accuracy is required, the devices with low error are quite expensive.

In the particular scheme presented in 2.9. Scheme of a two stage HP with open intercooler., has a flash tank and two different oil coolers for the compressor, which means that will be screw compressors probably.



2.9. Scheme of a two stage HP with open intercooler.

The two-stage model will be developed for the best option regarding the results of the heat source (Air/Water).

Chapter 3: Case Description

3.1 HEAT PUMPS IN DATACENTERS

Regarding the integration of Heat Pumps in Datacenters, is a field that is shortly developed, but there are still some studies and even facilities already done implementing this idea. Maybe the fact that make this kind of installation not as feasible as it looks is that the District Heating network for the countries with not so cold climate have a low heat demand at many times in the year.

Moreover, for the reuse of the waste heat in datacenters exist two different troubles, the first is the low temperature and the heat transport, which is difficult. In datacenters cooled with air, the waste heat use to be captured between 25ºC and 35ºC. On the other hand, for datacenters cooled with liquid, temperatures can be captured at a higher range due to the heat can be extracted closer to the processors, until 50ºC and 60ºC [17], [18].

As was introduced before, even the waste heat coming from the datacenters is at low temperature, is enough to be used as a heat source for the heat pumps, which also can be used to increase the temperature of the sink. Then, if HP are integrated to datacenters, they can produce cooling for the server room. However, it is still not common use HP as the unique source for cooling. But in that case, HPs should be connected to the DH system, in order to feed the waste heat [17].

The COP of the heat pumps in datacenters is typically ranged between 3.0 and 6.3 and the performance decreases when the forward temperature is increased. An interesting question in integrating a heat pump into a datacenter is who will make the investment. Heat pumps are heavy investments; therefore, if the investment is made by a datacenter operator, there must be perspectives for being able to sell the heat. In the scenario that the heat pump is not integrated into the datacenter, the investment is used to be done by the DH operator. The situation where the DH operators can buy waste heat coming from external sources could have a huge impact on its benefits, because the cost production at the peak is much higher than simply buy the waste heat for those moments, in addition, normally the peak load production is obtained by fossil fuels, consequently buying the waste heat will reduce the amount of emissions [17].

A current example of integration of HP for datacenter cooling is Fortum Värme, which has also developed a specific district cooling service with heat recovery that is used by Interxion's datacenter in Kista [19]. Also Facebook's planned Danish datacenter will supply hot air to the DH system of the nearby city of Odense, once it opens in 2020. When the site is built, Facebook's waste heat will be heated up by a HP, and delivered as hot water into a heating system. Odense is the third largest city in Denmark, with 175,000 citizens - and Facebook believes it can arrive to supply up to 100,000 MWh of energy per year waste heat could warm up until 6,900 homes [20].

Recent studies made by Wahlroos, says that for one specific studied case, waste heat utilization saved total operational costs of the simulated DH system by 0.6%-7.3%, depending on the waste heat utilization level (18.7 MW - 58.5 MW). However, the profitability was tightly linked to the pricing structure and electricity prices of the simulated years. Utilizing waste heat improves the energy efficiency of a datacenter [17].

Waste heat could be an interesting way to earn money for the datacenters. For example, a 1.2MW datacenter that sells all of its waste heat, could translate into more than 300,000€ per year. That may be as much as 14% of the annual gross rental income from a datacenter of that size, with very high profit margins [18]. According to Stenberg [21], investing in a heat pump for a datacenter, estimating that will produce waste heat at 75ºC, and selling that energy to the district heating company, the heat pump will have nearly less than two years of payback.

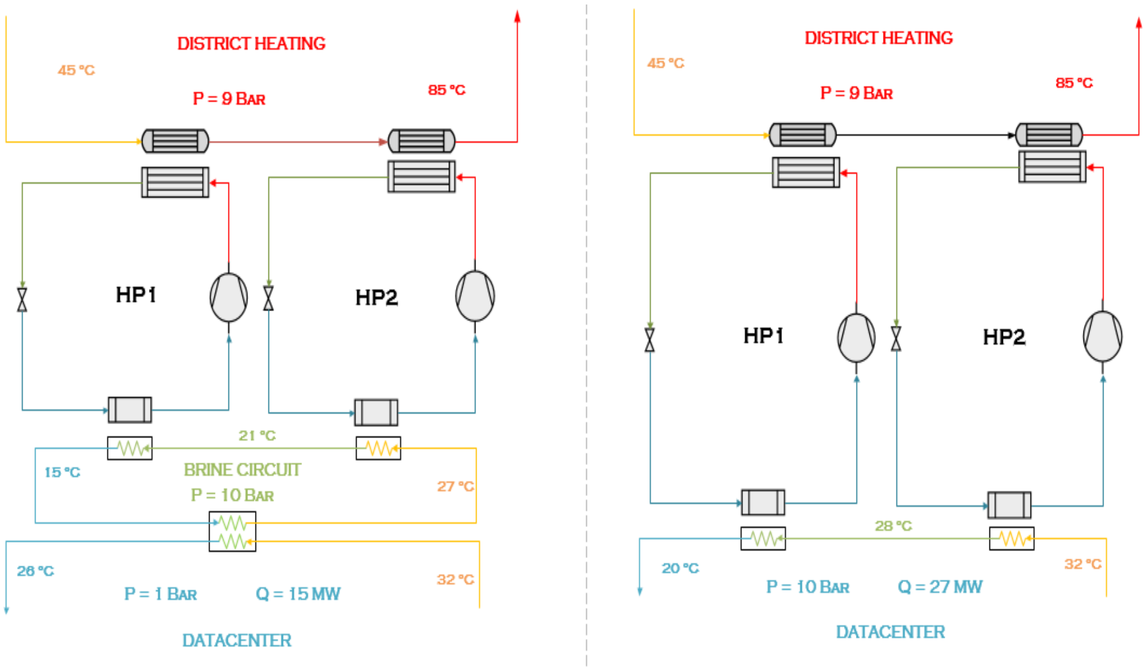
3.2 CASE BOUNDARY

Particularly, the datacenter that is going to be studied is placed in Kassø, near of Aabenraa. This building is being constructed by Apple, and it will help working to all the Apple's systems in north Europe, being the second datacenter from this company made in Denmark. It is estimated that the surface will be 1,786,809m^2, and its operation will start in the second quarter of 2019. Moreover, it is planned that the datacenter will be run on 100% renewable energies, and Apple has contracted with a local renewable energy developer to make a 30MW wind farm capacity to run it [22].

At this point, the project will take as a reference the engineering report made by the company PRIEBE [23], where the possibilities to install some type of heat pumps were evaluated and the main parameters given as a result.

Firstly, two different sources are purposed, air and water, the air heat pump additionally will have a brine circuit to exchange the heat between the air of the datacenter and the evaporator of the heat pump. Meanwhile in the case of the water heat pump a condensing tower will be needed. The inputs of the return and forward of the datacenter and the district heating network have been taken from the report, but afterwards it will be discussed what happen if the datacenter temperatures are modified.

Initially, the refrigerant that will be used is the Ammonia, due to its low cost and its low ODP for being a natural refrigerant. Despite this, other refrigerants will be tried after in the project, such HFOs, to compare the efficiency between both.



3.1. Schemes of the main inputs for the Air/Water heat pumps case 0.

In the last figure both heat pumps can be observed, and the differences are remarkable, specially in the capacity of the system. In the first case the air model has

15MW and in the second the capacity of the evaporator is 27MW, almost the double, this capacities are taken from a percentage of the total cooling necessity, which is indicated as 30MW in the report. Also is easy to notice that the forward temperatures to the datacenter are not equal, and the water heat pump will have to work in worse conditions, cooling the flow until 20ºC, and therefore its COP will be influenced, because the higher the forward temperature at the source the higher COP.

Chapter 4: Thermodynamic model

4.1 ONE STAGE

In order to obtain a model that simulates the reality some assumptions have to be done. For the one stage case, which is the simplest, the taken considerations were:

- Evaporator capacity is fixed.
- Temperatures of source and sink taken from the report as explained before.
- No pressure loss at the heat exchangers.
- Isentropic efficiency and compressor heat losses assumed. $\eta_{isentropic}$=0.75 Heat losses at piston compressor 5%
- ΔT of 5K for the evaporator and condenser at the beginning.
- 5K as subcooling and no superheat considered.
- Flows are assumed to be stationary.
- Pressure of the district heating network and the datacenter circuit is set to 9 and 10 bar, respectively.

The modelling is going to be developed in EES, and two different sources will be studied to see the best points and the weaknesses of each one. Also in this first stage the addition of oil cooling for screw compressors against the piston compressors will be evaluated.

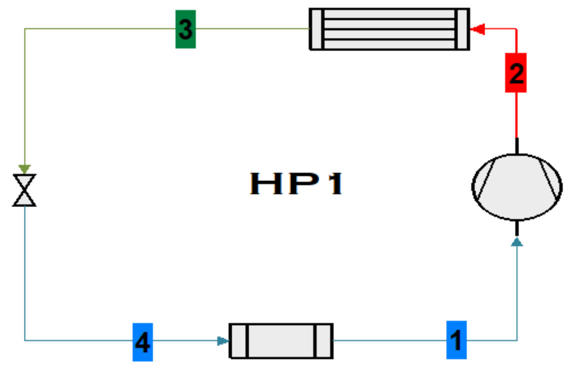
4.1.1 Air Heat Pump

Regarding the Air source, the assumptions were explained before and the basic scheme of the Heat pump can be found in figure 3.1, where are indicated the inputs of the system in this case as well. The interesting part of the scheme is the brine circuit, this circuit is purposed because the specific heat of the air is quite lower than the water, and therefore is easier to exchange heat with an additional brine closed circuit at a higher pressure that will need lower mass flow amount. The brine is supposed to be NaCl with a concentration of 0.5, but could be simply water because at any point of the system is below 0 ºC.

Moreover, the fact that the return temperature of the datacenter is lower in the air case comparing it with the water one, could be because the heat transfer is not direct to the air, and has to be exchanged with the brine firstly. Other consideration to be taken is that the forward temperature for the brine is 15ºC, and if the ΔT difference for the evaporator is applied, the evaporator temperature will be 10ºC, and for this kind of application that temperature difference with the heat sink will decrease the COP considerably.

Regarding the temperature at the brine circuit, at the report only were specified the forward and return temperatures, so the mid temperature has been set in the middle of both, in that way the capacity at the evaporator will be almost the same for the two heat pumps, and for the test made, this strategy performs the best. This model is done with a piston compressor, without oil cooling system and a 5% of heat losses to the surrounding.

The state points are specified in the next figure 4.1:



4.1. State points for a single stage HP.

And consequently the temperatures fixed by the pinch are:

$$\boldsymbol{T_{1HP1} = T_{return,brine} - \Delta T_{evaporator}} \tag{4.1}$$

$$\boldsymbol{T_{1HP2} = T_{mid,brine} - \Delta T_{evaporator}} \tag{4.2}$$

$$\boldsymbol{T_{C,HP1} = T_{pinch,DH,HP1} - \Delta T_{condenser,HP1}} \tag{4.3}$$

$$\boldsymbol{T_{C,HP2} = T_{pinch,DH,HP2} - \Delta T_{condenser,HP2}} \tag{4.4}$$

$$\boldsymbol{T_{3,HP1} = T_{return,DH} - \Delta T_{subcooling,HP1}} \tag{4.5}$$

$$\boldsymbol{T_{3,HP2} = T_{mid,DH} - \Delta T_{subcooling,HP2}} \tag{4.6}$$

Even is not indicated in the scheme, the condenser have subcooling and desuperheater as well, that will be show afterwards at the T-Q diagrams of the condenser.

The mass flow of the datacenter circuit has been calculated with the already knew temperatures, pressure and the capacity at the evaporators, the same as the mass flow of the brine circuit and the DH. On the other hand, the flow rate of the HP has been obtained with the balance (**4.7**), where the rest of the variables can be obtained.

$$\boldsymbol{\dot{Q}_{E_{HP1}} = \dot{m_{ref}}_{HP1}(h_{1_{HP1}} - h_{4_{HP1}})} \tag{4.7}$$

At this point with all the assumptions and the key temperatures set by the pinch, all the calculation can be carried for this air model with a piston compressor, but before the rest of the equations regarding the cycle have to be analysed.

The remaining equations to solve the cycle are quite simple and are going to be developed in this section just for the Heat Pump 1, the same principles should be followed to compose the equation for the Heat pump 2.

- ***Evaporator***

$$\dot{Q}_{E_{HP1}} = \dot{m_{brine}}(h_{mid_{brine}} - h_{return_{brine}}) \tag{4.8}$$

$$\dot{Q}_{E_{HP1}} = \dot{m_{ref}}_{HP1}(h_{1_{HP1}} - h_{4_{HP1}}) \tag{4.9}$$

$$\dot{Q}_{E_{HP1}} + \dot{Q}_{E_{HP2}} = \dot{m_{brine}}(h_{forward_{brine}} - h_{return_{brine}}) \tag{4.10}$$

- ***Condenser***

$$\dot{Q}_{C_{HP1}} = \dot{m_{DH}}(h_{mid_{DH}} - h_{return_{DH}}) \tag{4.11}$$

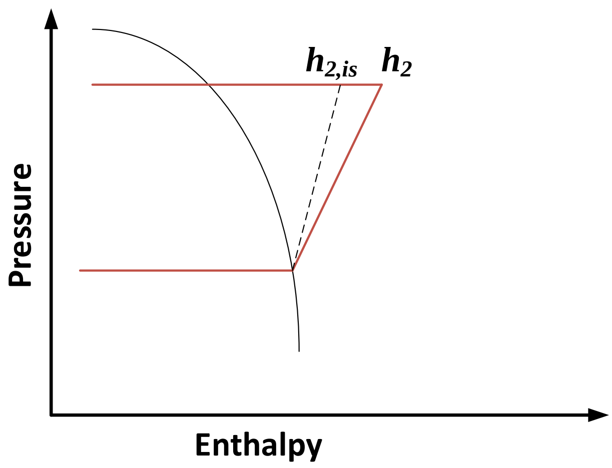
$$\dot{Q}_{C_{HP1}} = \dot{m_{ref}}_{HP1}\ (h_{2_{HP1}} - h_{3_{HP1}}) \tag{4.12}$$

- ***Compressor***

In the compressor case, the isentropic efficiency is a really important parameter, but in this case will be the same as in the water HP in order to compare, and once selected which is the best model to be implemented, a commercial compressor will be choose to have an accurate result.

At the compressor, first is calculated which will be the point 2 for ideal isentropic conditions, and then the equation (**4.13**) is applied to obtain the real point 2, the process can be observed in figure 4.2.

$$\eta_{isentropic} = \frac{(h_{isen} - h_1)}{(h_2 - h_1)} \tag{4.13}$$



4.2. Process at the compressor.

Finally the compressor power is obtained by:

$$\dot{W}_{HP1} = \dot{m}_{ref_{HP1}}\left(h_{2_{HP1}} - h_{1_{HP1}}\right) * (1 + Compressor_{losses}) \tag{4.14}$$

By last, the performance of the cycle is calculated as:

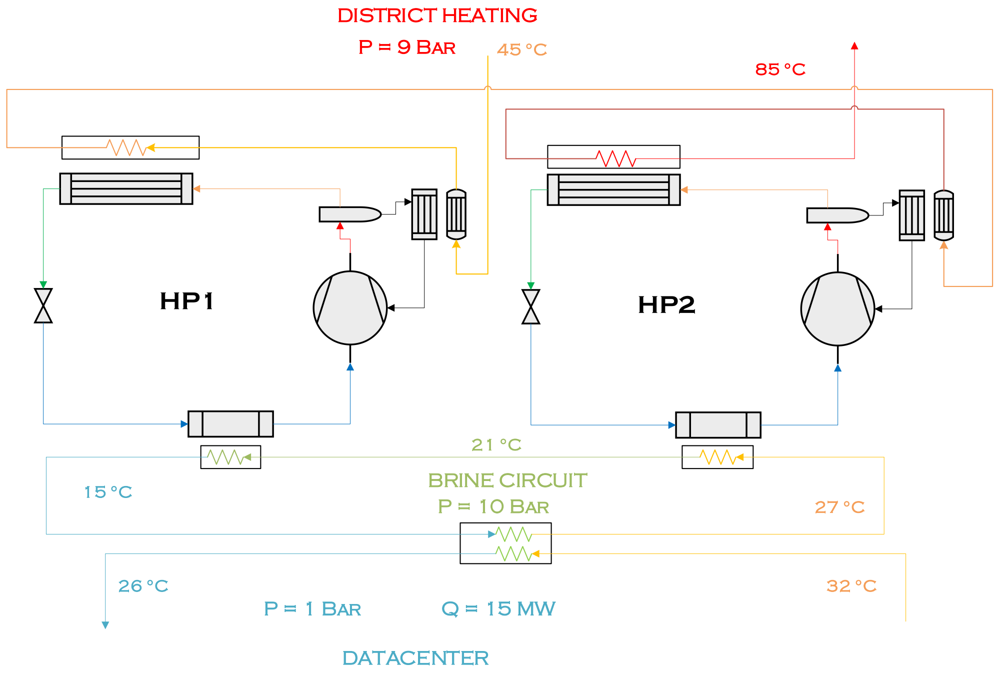
$$COP_{total} = \frac{\dot{Q}_{C_{HP1}} + \dot{Q}_{C_{HP2}}}{\dot{W}_{HP1} + \dot{W}_{HP2}} \tag{4.15}$$

At the same time, an equation is introduced to check the energy balance and ensure the calculations are made properly. The final result should be zero.

$$Energy\ Balance = \dot{Q}_{C_{HP1}} + \dot{Q}_{C_{HP2}} - (\dot{W}_{HP1} + \dot{W}_{HP2} + \dot{Q}_{E_{HP1}} + \dot{Q}_{E_{HP2}}) \tag{4.16}$$

Oil cooling model

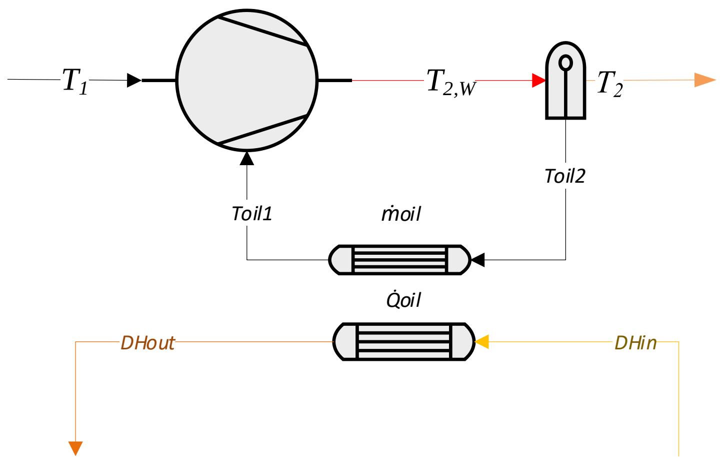
Now the simplest model has been explain, it has to be introduced the model with screw compressor and oil cooling integrated. The differences between both are important, because now the number of heat exchangers has been doubled, and the temperature at the outlet of the compressor will be lower, which means less capacity at the condenser.



4.3. One stage cycle with Oil cooling with Air as a heat source.

At the figure 4.3, a standard configuration is carried out, where the DH circuit goes at first through the OC of the first heat pump, then to the condenser of the first heat pump as well, and then follows the same path for the second heat pump. The OC are stablished in that position because if are located at last the temperature at the DH line will be too high for them to heat it.

Figure 4.4 shows the scheme that is followed to run the OC model, at first, the temperature in the inlet of the compressor, T_{oil1} and the oil mass flow, $\dot{m}_{oil}$, will be inputs taken from a commercial compressor software. Then, is assumed that the temperature of the refrigerant T_2, is the same as the outlet temperature of the oil flow, T_{oil2}, because in reality both flows will be mixed and then the oil being separated afterwards of the mix, assumable at the same temperature. The oil is cooled down by the heat transfer with the DH line, taking profit of the excess heat.



4.4. State points in the Oil Cooling circuit.

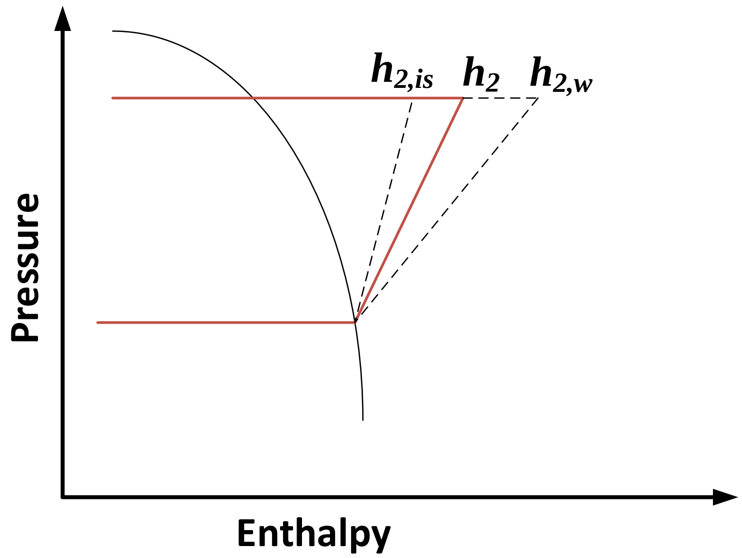
As in the other model, the equation of the isentropic efficiency is needed, but in this case the resulting point will be 2W, a high temperature that will decrease due to the oil flow.

$$\eta_{isentropic} = \frac{(h_{isen} - h_1)}{(h_{2W} - h_1)} \tag{4.17}$$

And consequently, the compressor power will be taken from 2W and not from 2.

$$\dot{W} = \dot{m}_{ref}(h_{2W} - h_1) \tag{4.18}$$

The state points are placed in the P-h diagram for the Ammonia in the figure 4.5, and can be observed that the highest temperature is reached for the point 2W, and the difference with the point 2 will set the heat exchanged by the OC.



4.5. P-h diagram of the Oil Cooling.

On the other side, the capacity of the OC is calculated as:

$$\dot{Q}_{oil} = \dot{m}_{oil} Cp_{oil}(T_{oil2} - T_{oil1}) \tag{4.19}$$

The calorific power of the oil is calculated taking as temperature the average between the inlet and outlet temperature of the oil.

$$\dot{Q}_{oil} = \dot{m}_{ref}(h_{2W} - h_2) \tag{4.20}$$

$$\dot{Q}_{oil} = \dot{m}_{DH}(h_{DH_{out}} - h_{DH_{in}}) \tag{4.21}$$

For this model, the COP will be different, due to in this case there are two additional exchangers:

$$COP_{total} = \frac{\dot{Q}_{C_{HP1}} + \dot{Q}_{C_{HP2}} + \dot{Q}_{oil_{HP1}} + \dot{Q}_{oil_{HP2}}}{\dot{W_{HP1}} + \dot{W_{HP2}}} \tag{4.22}$$

Normally, the possibility to add OC makes the system more efficient, and in some situation also makes feasible the heat pump, because of the too high temperatures at the outlet of the compressor, that without the OC could reach 180ºC, a temperature that should not be allowed for working with Ammonia.

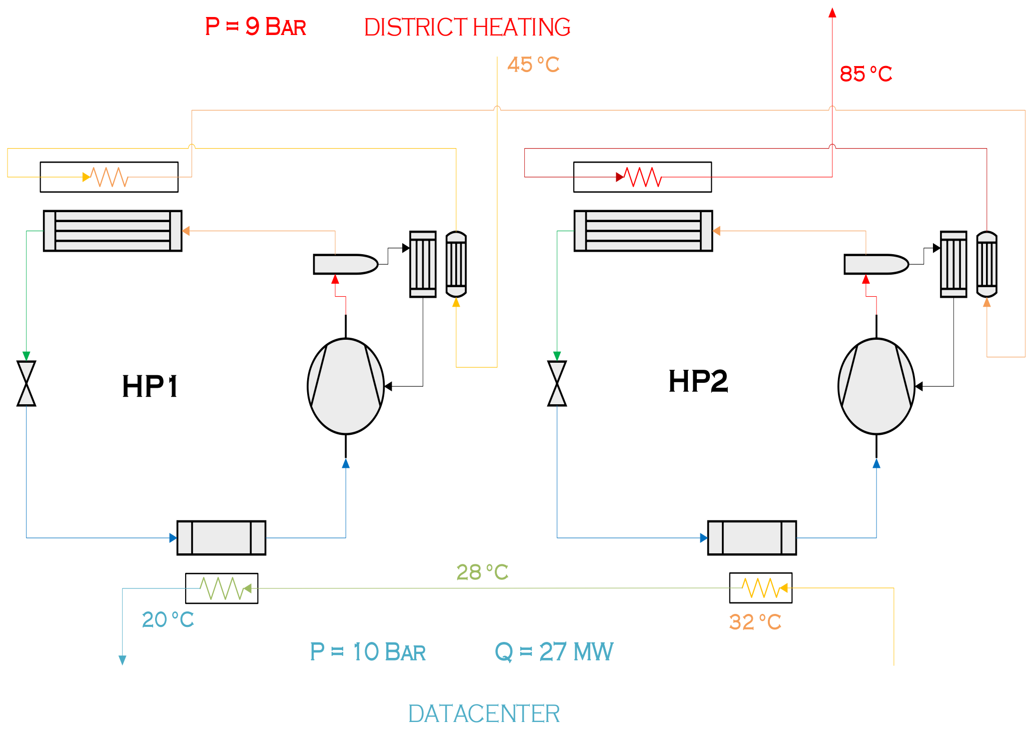
4.1.2 Water Heat Pump

The case with water as source (figure 3.1) do not have any circuit in the middle as the brine for Air source, this means that the exchange will be done directly from the datacenter side, which implies that the transmission has less steps, so should be more efficient.

In this situation, the total capacity of the evaporator is 27MW, almost the double comparing with the other case, and being water presents some advantages comparing with Air in terms of specific heat, which allows the installation to be much smaller. Moreover, the forward temperature to the datacenter is 20ºC, that means that the evaporator of HP1 will be at 15ºC, which is more than the 10ºC of the Air case, and at the same time, the forward temperature will be lower, so consequently, the Water source model presents some evidences that show probably the performance will be better.

The thermodynamic model of this case is quite similar to the air one, for that reason, the majority of parameters are not going to be explained again.

Moreover, for the water heat source two configurations will be tried at first, as in the air heat source case, with a piston compressor and with a screw compressor. The one with piston has been already showed in figure 3.1, and the model with screw compressor could be observed in figure 4.6.



4.6. Water One stage HP with screw compressor and OC.

The model with OC has the same configuration as the case of air with screw compressor, in order to compare properly in the same conditions the cases. In addition, the main variables and state points for this case and the air are identical, and consequently explain them other time do not have much sense, the entire code is placed on the appendix A.

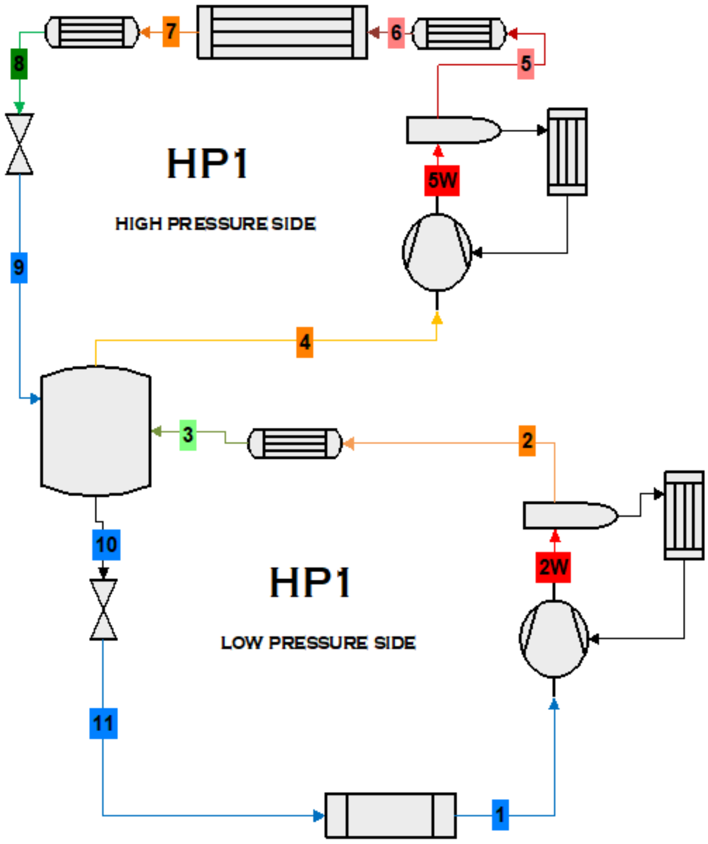
To sum up, four different configurations have been explained with their essential parameters, and the result and discussion will take place on the next chapter

4.2 TWO STAGE MODEL

In this section the thermodynamic model of the 2 stage heat pump will be explained, and the assumptions are the same as before, but particularly this time the isentropic efficiency of the compressor is not going to be estimated, and will be set with the software RTselect.

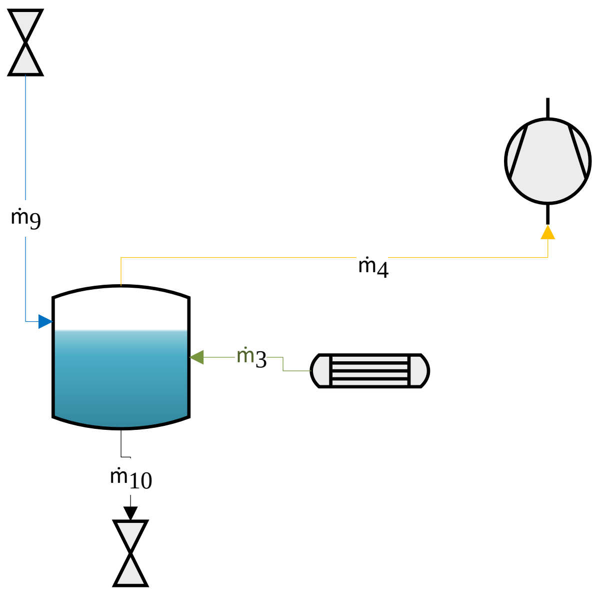
The new cycle will have two particularities, the first is the open intercooler, and the second is the desuperheater of the low stage. The benefit of the two stage system is that the compression is divided in two parts, which reduce the compression relation and therefore the isentropic efficiency of the compressor, which works in proper conditions.

This model is only going to be simulated with OC, due to the compressor selected are screw and the OC can be easily implemented. Also for the model the only heat source considered is the water. The refrigerant used is Ammonia, but another options will be evaluated in other Chapter. At figure 4.7 the state points can be recognized.



4.7. State points of the two stage HP case0.

Regarding the open intercooler, it can be simulated taking in account that for the state point 4 the refrigerant is saturated vapour, and for the point 10 the refrigerant is saturated liquid, for the other related points the working fluid is in a dual phase state, it can be observed in figure 4.8.



4.8. Open intercooler two-stage.

By means of an energy balance can be easily made to obtain the refrigerant mass flow of the high stage, since the mass flow of the low stage is determined by the capacity an temperatures at the evaporator, as in the one stage scenario.

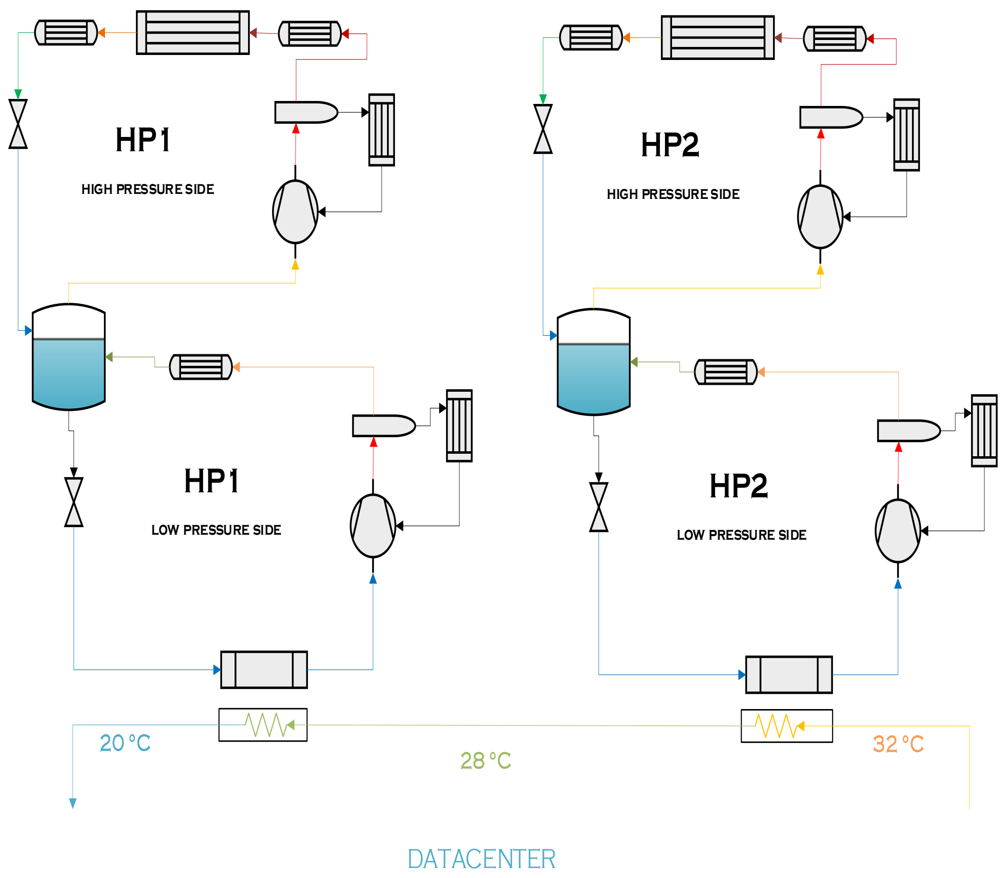
$$\frac{\dot{m_{ref}}_H}{\dot{m_{ref}}_L} = \frac{h_{10} - h_3}{h_9 - h_4} \tag{4.23}$$

$$\dot{Q}_E = \dot{m_{ref}}_L (h_1 - h_4) \tag{4.24}$$

$$\dot{Q}_E = \dot{m_{DH}} (h_{mid} - h_{forward}) \tag{4.25}$$

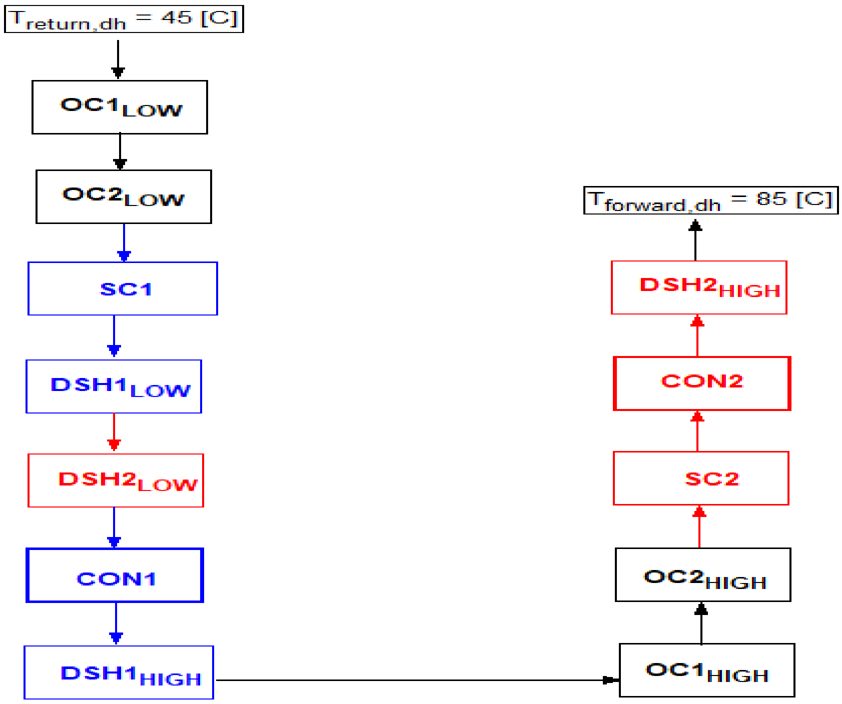
The intermediate pressure of the system, which is the one in the intercooler, has to be set. One solution could be make an optimization looking for the temperature that gives the highest COP, but in reality could be difficult to obtain compressors of the required parameters, and even in the situation where can be found, is possible that its isentropic efficiency is really low, and consequently it will be another optimal temperature that does not fit with the theoretical optimal. In this case the temperature has been stablished looking for real compressors with an acceptable efficiency in a temperature range close to the theoretical optimum.

In principle twelve different exchangers are set in the case0 model for the two-stage cycle, which compared with the one stage scenario is much efficient.



4.9. Two-stage HP with OC and low stage DSH.

By last, the purposed configuration regarding the district heating circuit is found in figure 4.10, is quite simple and does not present any division in strings. The low stage OC have a small impact in the system, and exchange a heat quantity that could be considered even as neglective, for this reason it should be placed at first, when is easier to exchange heat with the district heating line due to its low temperature.



4.10. Configuration for the two-stage HP case0.

Chapter 5: Pinch Method

5.1 INTRODUCTION

The pinch analysis methodology is essentially a graphical method, and one of its advantages is that permit the author to visualize quickly the streams of a process and presuppose the potential heat recovery by internal heat exchange. This method was introduced by Linnhoff, and is based on the process description as streams to be heated or cooled, its analysis, with the utilities formation and the comparison between the hot and cold streams at their temperature level makes the pinch analysis [24].

The first key concept of pinch analysis is setting energy targets, but the targets obtained at the pinch are not that usual as 'energy savings', are absolute thermodynamic targets, which show what the process is able to achieve if the heat recovery, heating and cooling systems are correctly designed [25].

The easiest way to explain how pinch method works is by means of an example:

There are two hot streams and one cold in a specific industry, the parameters can be followed in the next table.

Number	**Type**	**T_{start} (ºC)**	**T_{end} (ºC)**	**$\dot{m}Cp$ (kJ/s* ºC)**	**$\dot{Q}$**
1	Cold	20	90	2	140
2	Warm	70	10	1,5	90
3	Warm	110	50	1	60

5.1. Parameters of the streams.

The first step is divide all the streams into intervals, that in this case can be set as 10ºC, but in other cases could be needed more accuracy, so the calculation step will be much lower. Also is required to construct two different tables, one with the cooling demands containing the hot streams (table 5.2) and other with the heating demand containing the cold streams (table 5.3). The heat exchanged for each interval will be:

$$\dot{Q}(kW) = \dot{m}Cp(\frac{\mathbf{kJ}}{\mathbf{s^{o}C}}) * \Delta T(^{o}\mathbf{C}) \quad (5.1)$$

And then, the accumulative addition should be done, starting from the lower temperature until the higher.

T_{high} (ºC)	T_{low} (ºC)	$\dot{Q}1_{hot}$	$\dot{Q}2_{hot}$	$\dot{Q}_{total_{hot}}$	$\sum \dot{Q}_{total_{hot}}$
110	100	0	10	10	150
100	90	0	10	10	140
90	80	0	10	10	130
80	70	0	10	10	120
70	60	15	10	25	110
60	50	15	10	25	85
50	40	15	0	15	60
40	30	15	0	15	45
30	20	15	0	15	30
20	10	15	0	15	15
10	0	0	0	0	0

5.2. Cooling demand.

T_{high} (ºC)	T_{low} (ºC)	$\dot{Q}_{cold}$	$\sum \dot{Q}_{cold}$
110	100	0	155
100	90	0	155
90	80	20	155
80	70	20	135
70	60	20	115
60	50	20	95
50	40	20	75
40	30	20	55
30	20	20	35
20	10	0	15
10	0	0	15

5.3. Heating demand.

For this theoretical example the ΔT_{pinch} is going to be 0ºC, in reality this will have no sense, because an infinite surface of exchanger will be needed [26].

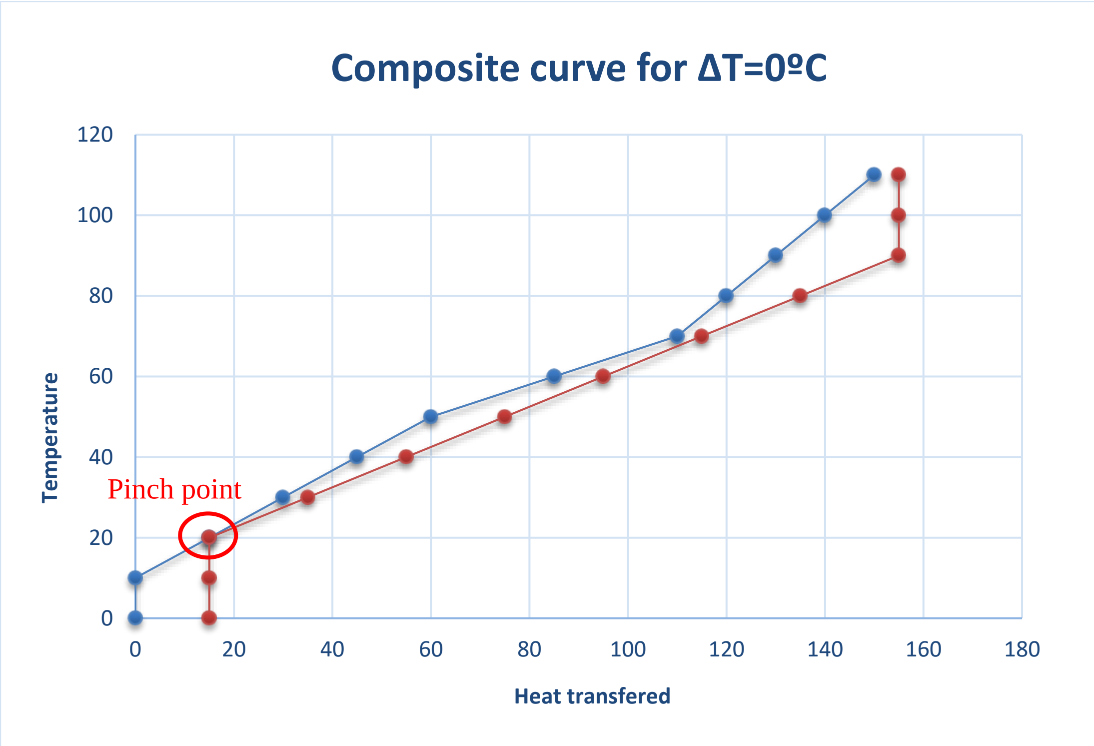
As Elmegaard says, the external utility requirement is calculated in three steps by doing [26]:

1. An energy balance for each interval. This is shown in table 5.4 in column 5, Net Cooling Demand.
2. Accounting for the energy surplus/deficit at higher temperature levels for each interval, as shown in table 5.4 column 6, Sum of Net Cooling Demands. The demand for external heat is the minimum value in this column, in this case -5kJ/s. The pinch temperature is the lowest temperature of this interval, in this case, 20°C.
3. Finally, adding the required external heat to the top interval gives the summation of energy surplus when the required heat is added, as in column 7, Sum of net cooling demands after heating, in Table 5.4. The minimum requirement for cooling is the net cooling demand at the lowest interval, 15 kJ/s [26].

T_{high} (ºC)	T_{low} (ºC)	$\dot{Q}_{total_{hot}}$	$\dot{Q}_{cold}$	Net Cooling Demand $\Delta\dot{Q}[kJ/s]$	Sum of Net Cooling demands $\sum\Delta\dot{Q}$	Sum of Net Cooling demands after utilities $\sum\Delta\dot{Q}+\dot{Q}_{utility}$
					0	5
110	100	10	0	10	10	15
100	90	10	0	10	20	25
90	80	10	20	-10	10	15
80	70	10	20	-10	0	5
70	60	25	20	5	5	10
60	50	25	20	5	10	15
50	40	15	20	-5	5	10
40	30	15	20	-5	0	5
30	20	15	20	-5	-5	0
20	10	15	0	15	10	15
10	0	0	0	0	10	15

5.4. Heat balance of each interval.

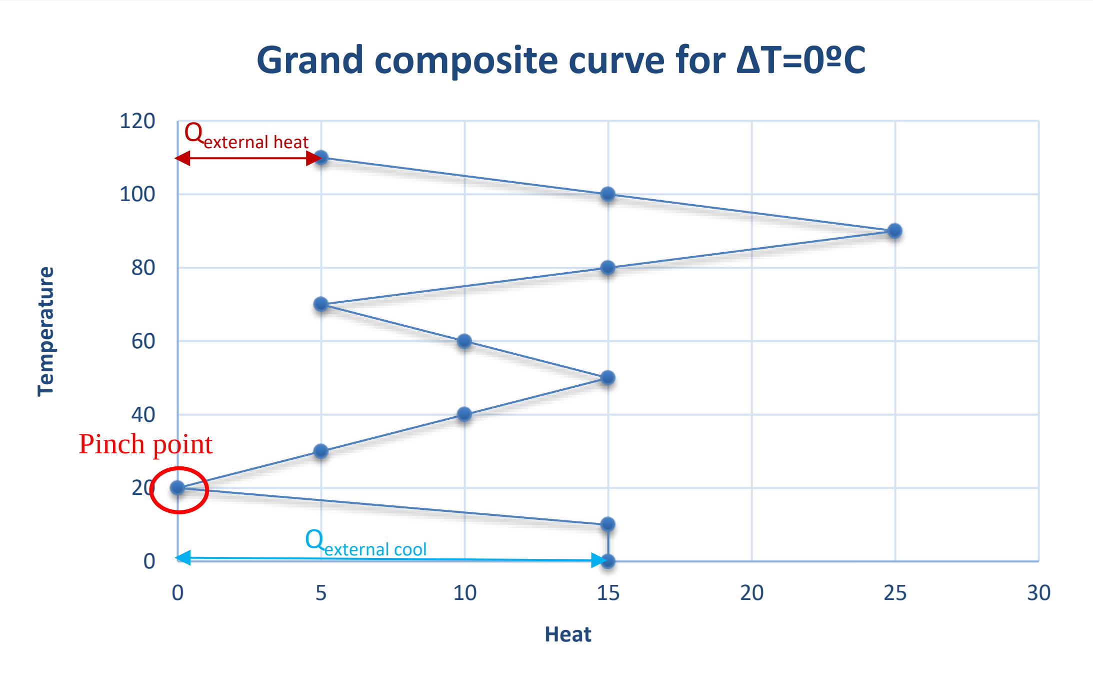
Once the accumulative heating and cooling demand is set, the composite curves could be made. Composite curves are single $T/\dot{Q}$ curves that represent all the streams in the system, and the overlap between the composite curves represents the maximum amount of heat recovery possible at the process.



5.1. Composite curves of the simple case.

The pinch point splits the process into two. Considering what is happening at the pinch temperature, the heat flow of all the hot streams $\dot{Q}_H$, relative to that at the pinch $\dot{Q}_{H_{PINCH}}$, is $\Delta\dot{Q}_H$. In the same way, the heat flow of all cold streams belonging to that at the pinch is $\Delta\dot{Q}_C$. There is an imbalance that must be supplied by external heating and cooling. Above the pinch, $\Delta\dot{Q}_C > \Delta\dot{Q}_H$ and the difference must be supplied by hot utility. Consequently, below the pinch $\Delta\dot{Q}_H < \Delta\dot{Q}_C$ and the excess heat is removed by cold utility [25].

Then, knowing the shifted composite curves, the minimum quantity of cooling or heating that is needed to be supplied at any temperature is found. A graph of net heat flow against shifted temperature can then easily be plotted. This is known as the grand composite curve .It represents the difference between the heat available from the hot streams and the heat required by the cold streams, relative to the pinch, at a given shifted temperature. Thus, the Grand Composite Curve is a plot of the net heat flow against the shifted (interval) temperature, which is simply a graphical plot of the heat cascade [25].



5.2. Grand Composite Curve for the simple case.

But the method does not end at this point, as Linnhoff said, the approach to the problem has been radically altered by two major discoveries: network performance targets and the network temperature pinch. The network temperature pinch represents a bottle-neck to feasible heat recovery in HEN design, and the Network performance targets exist for the minimum utility usage and the minimum number of “units” (i.e. process and utility exchangers). Calculation of these targets is simple and is possible independent of design. Thus, targets can be used, first to stimulate the designer towards better designs, and second to give the designer confidence that his solution is near-optimal [27].

That means that the last step in the pinch method is to perform a Heat Exchanger Network that could fit with the new parameters of the optimization, and in Chapter 6: Results, some HEN will be tried to obtain one that fits with the system and theoretically solves the problem.

5.2 PINCH METHOD APPLIED TO HEAT PUMPS

The application of the pinch method to heat pumps is not developed at all, and there is just a few studies that suggest how is supposed to be done, even that, it is not clear if the method works properly or not, and this thesis will try to clarify it.

The pinch analysis of the HP cycle is different from the conventional pinch method, due to the HP is balanced by itself from the compressor power, such that all the cooling and heating demands from the cold and hot streams were internally recovered [28].

In the particular case of a Heat Pump integrated with the District Heating Network, the only cold stream to cover is the one that belongs to the District Heating line, and the rest of the exchangers will compose the curve of hot streams.

In a general pinch method application, is important to find where can be the pinch point, but in the application for HP, since the beginning is set a ΔT at some points, leaving that variables free to the criteria and experience of the engineer, in order to obtain results in the simulations close to the reality. Therefore, in this application, for a model of a two-stage HP, as the one presented in figure 4.9 and 4.10, the number of variables to optimize increases a lot, having until 6 pinch points for the optimization.

Key temperature	Designation	Pinch temperature difference
$T_{HP1}[8]$	Temperature after SC in HP1	$\Delta T_{SC1} = T_{HP1}[8] - T_{OC2_{LOW_{outDH}}}$
$T_{HP2}[8]$	Temperature after SC in HP2	$\Delta T_{SC2} = T_{HP2}[8] - T_{SC2_{in_{DH}}}$
$T_{HP1}[3]$	Temperature after Low stage DSH in HP1	$\Delta T_{DSH1_{LOW}} = T_{HP1}[3] - T_{SC1_{out_{DH}}}$
$T_{HP2}[3]$	Temperature after Low stage DSH in HP2	$\Delta T_{DSH2_{LOW}} = T_{HP2}[3] - T_{DSH2_{LOW_{in_{DH}}}}$
$T_{C_{HP1}}$	Condensing Temperature in HP1	$\Delta T_{C_{HP_1}} = T_{C_{HP_1}} - T_{CON1_{in_{DH}}}$
$T_{C_{HP2}}$	Condensing Temperature in HP2	$\Delta T_{C_{HP_2}} = T_{C_{HP_2}} - T_{CON2_{in_{DH}}}$

5.5. Key temperatures of the two-stage cycle.

The target of the application of this method at Heat pumps will be the COP of the total system in this case, parameter that by its own evaluate the performance of heat recovery. As was explained in the last section, the first part of the pinch method is discretise the temperatures into ranges, which are defined by:

$$\Delta T_{interval} = \frac{T_{max} - T_{min}}{n - 1} \tag{5.2}$$

Where T_{max} represents the highest temperature at the HP, placed at the inlet of the desuperheater, T_{min} corresponds to the lowest temperature at the HP, which is at the outlet of the subcooler, and *n* is the number of intervals, set in this case to 100. On the other hand, the cold stream of the system is simulated as a line going from 45°C to 85°C, and its equation will correspond to:

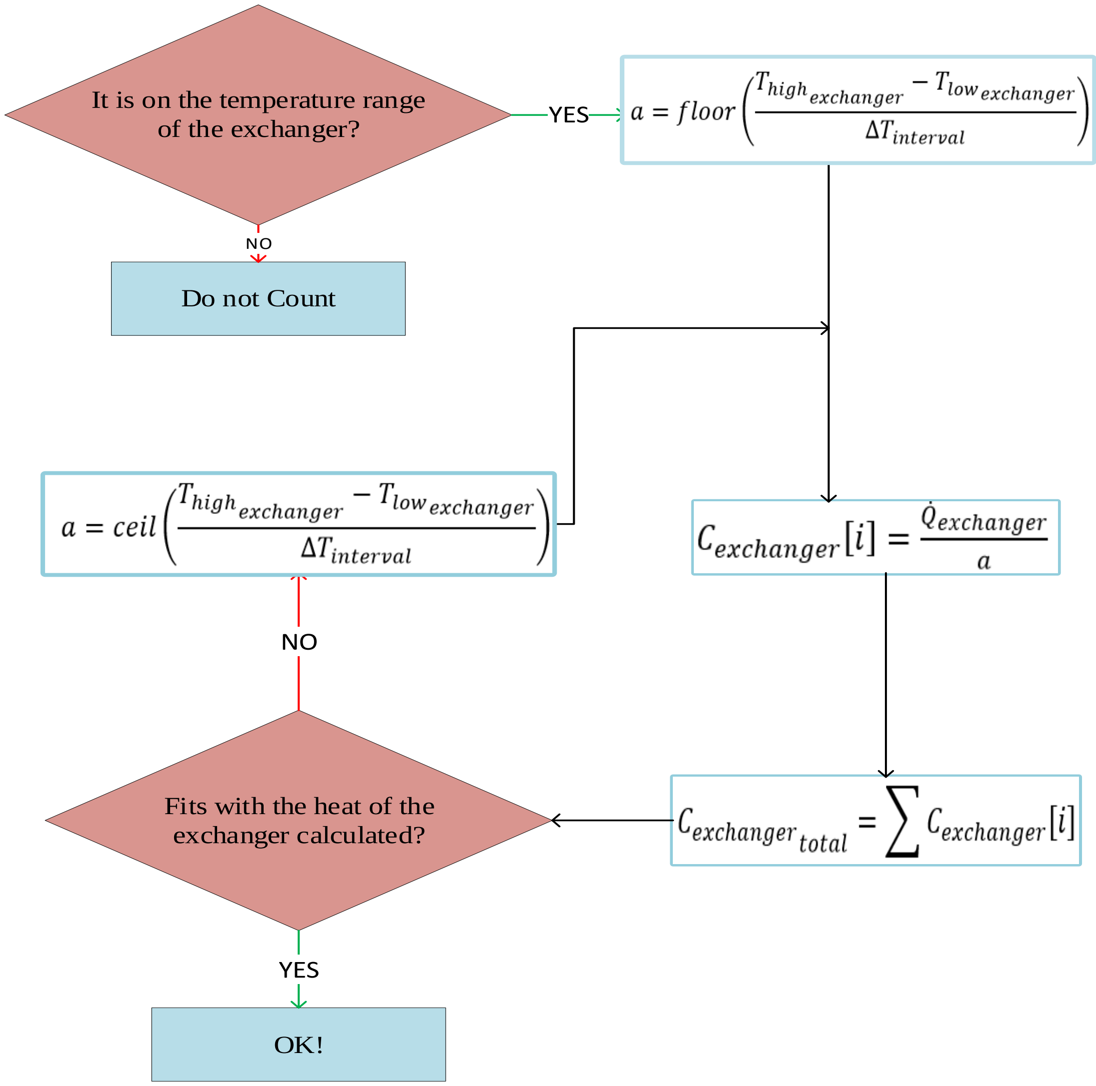
$$T_{cold}[i] = \frac{\left(T_{forward_{DH}} - T_{return_{DH}}\right) * \dot{Q}_{heat}[i]}{\dot{Q}_{condenser}} + T_{return_{DH}} \tag{5.3}$$

Meanwhile the hot stream is divided into the corresponding intervals, conditioned by $\Delta T_{interval}$.

$$T_{hot}[i+1] = T_{hot}[i] - \Delta T_{interval} \tag{5.4}$$

$$T_{low}[i] = T_{hot}[i] - \Delta T_{interval} \tag{5.5}$$

Each exchanger has its loop, where an if condition is set in order to check if the range temperature is at the range of that specific exchanger, and then a conditional with a floor is applied, in case that it does not fit with the calculated heat exchange, a ceil condition is applied, ensuring that in one or the other way at the end it will be correct. Floor is a function that makes a decimal number to be rounded to the lowest entire number, ceil rounds to the closest upper entire number.



5.3. Diagram of the code for the exchangers calculation.

Then, all the heat transmitted at any range by the exchangers has to be included in a variable, which will be called CThot[i].

$$\boldsymbol{CT_{hot}[i] = C_{exchanger_1}[i] + C_{exchanger_2}[i] + C_{exchanger_n}[i]} \quad (5.6)$$

Finally, the heat according to each temperature will correspond to:

$$\dot{Q}_{heat} = \dot{Q}_{condenser} - CT_{hot}[i] \tag{5.7}$$

And in that way, the T-Q diagram can be constructed to see clearly the pinch points and the evolution of the curves. The only thing remaining at this point is set the difference between the hot and cold stream, to check the behaviour at each temperature range, and afterwards constrain the ΔT for every interval with a minimum value of 3K, in order to obtain a global pinch point with a given minimum temperature difference.

$$\Delta T[i] = T_{hot}[i] - T_{cold}[i] \tag{5.8}$$

Then, an optimization using the Variable Metric Method on EES will be performed. The parameter to optimize is the COP, with 8 different variables, the 6 ΔT regarding the pinch and both intermediate temperatures for the two-stage case. The result of the optimization will imply the design of a Heat Exchanger Network, with the corresponding minimum difference of 3K.

Chapter 6: Results

6.1 ONE STAGE

In this section, the results of the one stage model will be shown and a technical decision in terms of the best source option will be set. Then other configurations of the exchangers will be done with their corresponding evaluation, and the pinch method will be applied to the best configuration by last, with its corresponding HEN proposition.

6.1.1 Air Heat Pump

The results for the air source are conditioned by an assumed isentropic efficiency and heat losses for the piston compressors. Only the main parameters will be exposed, since the majority of results do not have a significant impact in the analysis.

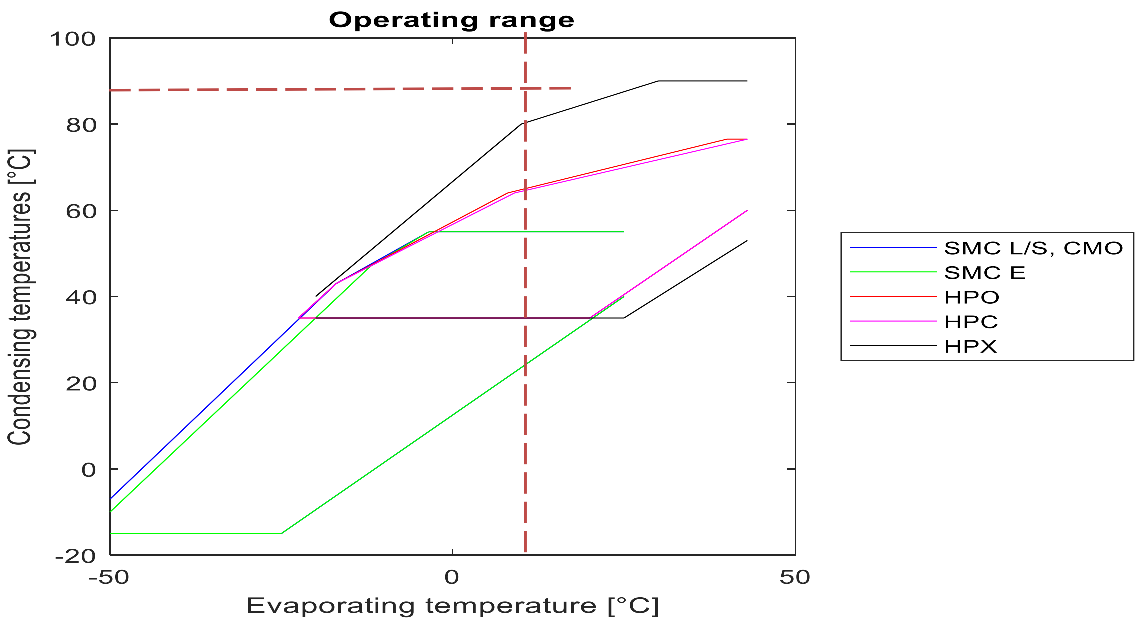
Regarding the configuration with the piston compressor, the simulation determines the next results:

- $COP = 3.671$
- $\dot{Q}_{condenser} = 20.26\ MW$
- $\dot{Q}_{evaporator} = 15\ MW$
- $\dot{W}_{compressor} = 5.518\ MW$
- $T_{C_{HP1}} = 65$°C
- $T_{C_{HP2}} = 84$°C
- $T_{discharge_{HP1}} = 159.2$°C
- $T_{discharge_{HP2}} = 192.1$°C

The first thing to consider is the total COP of the Heat Pump, which clearly is not really good taking in account the operational temperature range of the HP, from 32°C to 26°C. When comparing COPs is essential to know the temperature range to analyse it. The capacities are almost split by its half, which means both HP are exchanging more or less the same heat with the source and sink, but at a different temperature range.

The main problems of this simulation are the high temperatures at the discharge of the compressor, 160°C and 192°C, which are not recommendable for Ammonia.

And on the other hand, the condensing temperature of the HP2 is almost on the limit of the piston compressors with an evaporating temperature of 16°C, arriving to a pressure of $P_{C_{HP2}} = 45{,}2\ bar$. As can be followed in the figure 6.1, the only compressor that could fit with that is the HPX series.



6.1. Operating range of Johnson piston compressors.

At the same time, the piston compressor applications for large facilities is not really extended, and for carrying with the compressor power equal to 5.518 MW, the required number of piston compressor could be massive. Therefore, a screw compressor configuration is tried, due to the OC possibilities and also the bigger size of an individual unit.

Oil Cooling

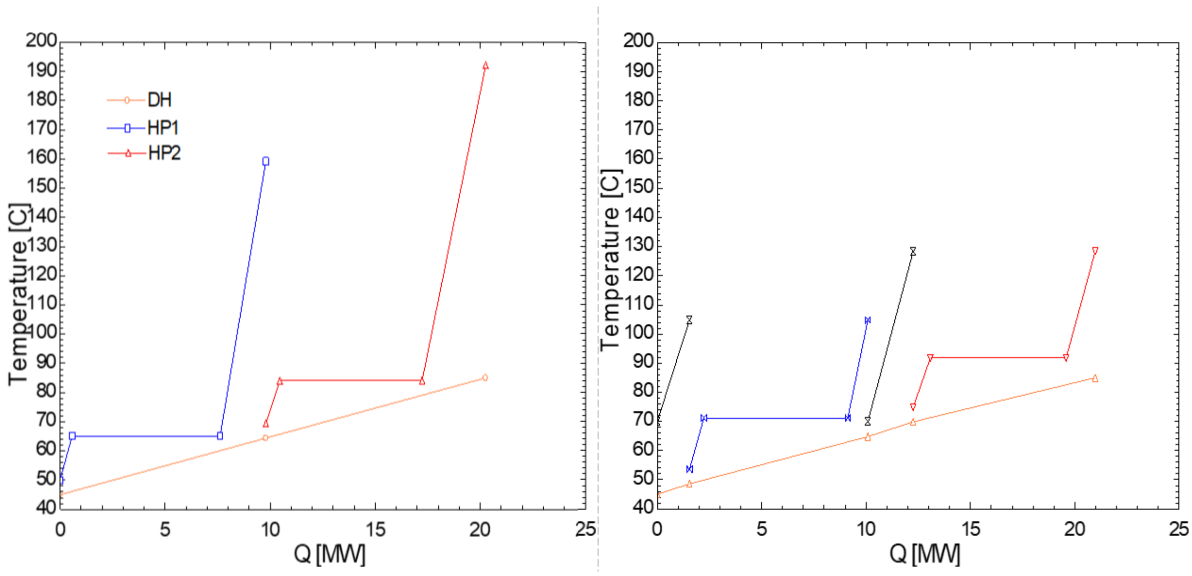
The main results of the Oil cooling model are the following:

- $COP = 3.509$
- $\dot{Q}_{condenser} = 20.89\ MW$
- $\dot{Q}_{evaporator} = 15\ MW$
- $\dot{W}_{compressor} = 5.978\ MW$
- $T_{C_{HP1}} = 71.15°C$
- $T_{C_{HP2}} = 91.78°C$

$$\bullet\ T_{discharge_{HP1}} = 104.8°\text{C} \quad \bullet\ T_{discharge_{HP2}} = 128.3°\text{C}$$

Analysing the results, is curious to notice that the configuration with OC has a lower COP, and that happens due to the compressor power has increased more than the condenser capacity. This fact can be explained because the temperature at the point *2W* at the outlet of the compressor, just before the OC has effect, has increased in this model. In the piston model, heat losses are considered, not as in this case, and that is the explanation of the temperature increase, which consequently makes the power of the compressor increase more than the OC capacity is able to do.

Other factor to consider is the condensing temperature, which has increased in this case remarkably, and taking a look at the figure 6.2 can be explained why. The condensing temperature is fixed with a ΔT_{pinch} with the DH line, in the case of the OC, the temperature is increased before the condensers by means of the heat exchanged at the OC. This implies that at the pinch point, the DH temperature has increased more than in the other case, which makes the condensing temperature higher. The problem that the high condensing temperature is that could be difficult to find compressors of that parameters, and maybe the only option is to implement a two stage cycle.



6.2. T-Q diagram of the air models.

The last thing to look at is the discharge temperature, as can be seen in the figure 6.2, the temperature in the OC model is much lower than in the first case, even the comparison is between two different types of compressor, is not recommendable to work with Ammonia at that high temperatures.

6.1.2 Water Heat Pump

In this case, the water source configuration has distinct inputs and consequently the capacity at the evaporator will be bigger, as the rest of the capacities of the system.

- $COP = 3.949$
- $\dot{Q}_{condenser} = 35.58\ MW$
- $\dot{Q}_{evaporator} = 27\ MW$
- $\dot{W}_{compressor} = 9.01\ MW$
- $T_{C_{HP1}} = 70.41°\text{C}$
- $T_{C_{HP2}} = 86.15°\text{C}$
- $T_{discharge_{HP1}} = 160.7°\text{C}$
- $T_{discharge_{HP2}} = 180.7°\text{C}$

Regarding the results, the first thing that has changed is the COP, being now almost at 4, and 0.5 higher than the air sink case. Since the capacity of the evaporator for the water source raised, the capacity at the condenser and the power at the compressor will increase as well. But the real reason of why did the COP augmented is due to the modification of the temperature at the evaporator. In the air case the temperature at the inlet of the evaporator was fixed by the $\Delta T_{evaporator}$ as 10°C and 13°C, although in the water situation these temperatures are 15°C and 23°C. To sum up, the COP is better because the cooling range is lower, despite at the datacenter the temperature decreases more in the water source configuration.

The condensing temperature for this model has the same problem as the air piston model. At that temperature range and with the evaporating temperatures of 15°C and 23°C is difficult to find a piston compressor (see figure 6.1).

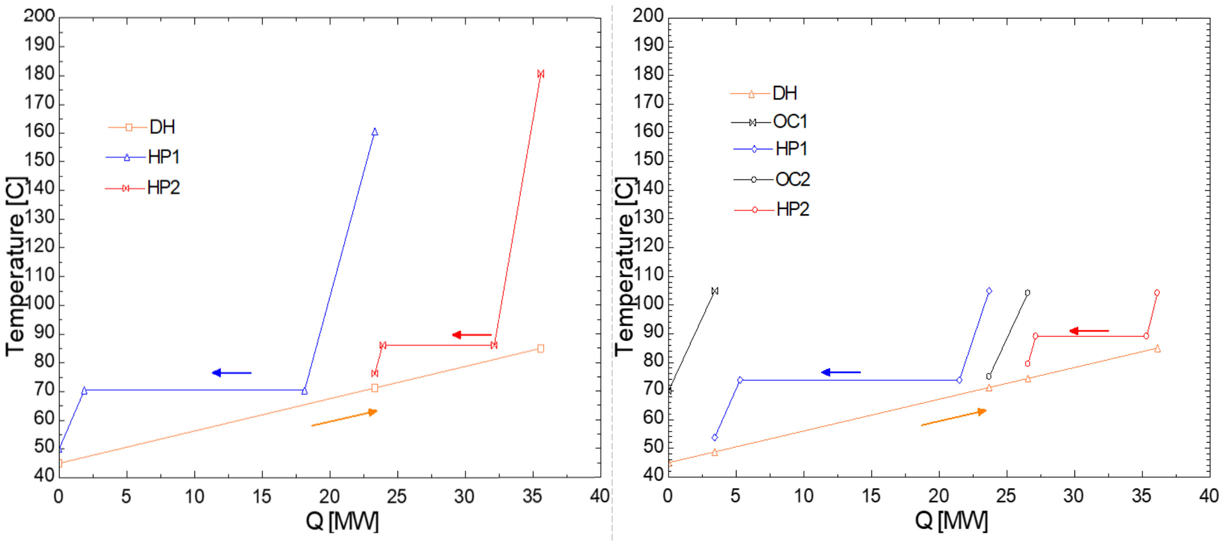
By last, the discharge temperature could be a problem, because of its high temperature and the danger involving the auto-combustion of lubricating cylinder oil in the presence of hot, compressed air.

Oil cooling

Once implemented the OC for the water source, the simulation show the results:

- $COP = 3.929$
- $\dot{Q}_{condenser} = 36.22\ MW$
- $\dot{Q}_{evaporator} = 27\ MW$
- $\dot{W}_{compressor} = 9.219\ MW$
- $T_{C_{HP1}} = 73.79°\text{C}$
- $T_{C_{HP2}} = 89.11°\text{C}$
- $T_{discharge_{HP1}} = 105.2°\text{C}$
- $T_{discharge_{HP2}} = 104.5°\text{C}$

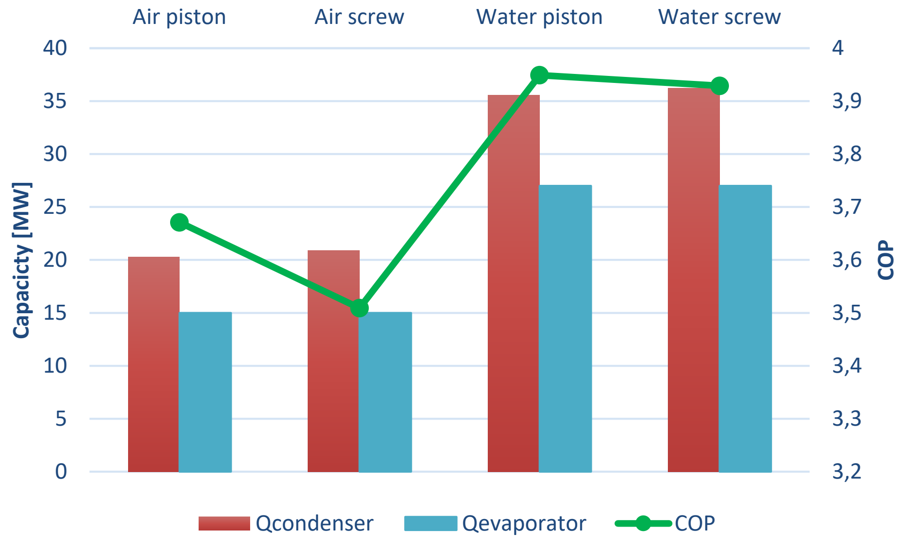
The COP in this case decreases a bit, 0.02. This fact can be explained with the increase of the condensing temperature. When the OC are added to the system, particularly in this case, they heat up the DH line before the condenser, and therefore, the condensing temperature and the discharge (point *2W* in figure 4.4) will be higher, making the compressor power raise more than the condenser capacity. On the other hand, the condenser capacity also increases due to the addition of the OC.



6.3. T-Q diagrams of the water source.

In the last graph it can be observed how the discharge temperature is reduced for the OC configuration, at an acceptable range for screw compressors and ammonia systems.

Now with all the results for the initial one stage HP presented, the comparison could be found in figure 6.4. In terms of COP, the best option to perform is the Water piston configuration, but present some problems due to the high discharge temperature, which is not recommendable, and also for the availability to find a compressor with that high condensing temperature. Even could be possible to find a piston compressor with the required parameters, it will have a size until 150kW-200kW, which will imply the installation and maintenance of a lot of different compressors, and technically is not a good option to implement.



6.4. Comparison of the one-stage HP initial results.

For the reasons explained, the evaluation of the performance and technical parameters makes the Water Screw model the best option to develop. Therefore, in the next sections real compressor data from the software RTselect will be taken, and in addition, new configurations will be constructed, and the pinch method will be applied for the best of them, looking for the highest COP as possible.

6.1.3 Configurations

In this section different ways from the path of the DH line are considered, the main results will be showed, and the configuration with the best COP will be selected to apply the pinch analysis.

For all the configurations the source is water and for the ones with Ammonia (the first four), the discharge temperature is around 110°C. The evaporator capacity is 27MW for every configuration and the ΔT with all the exchangers is 5K.

First configuration

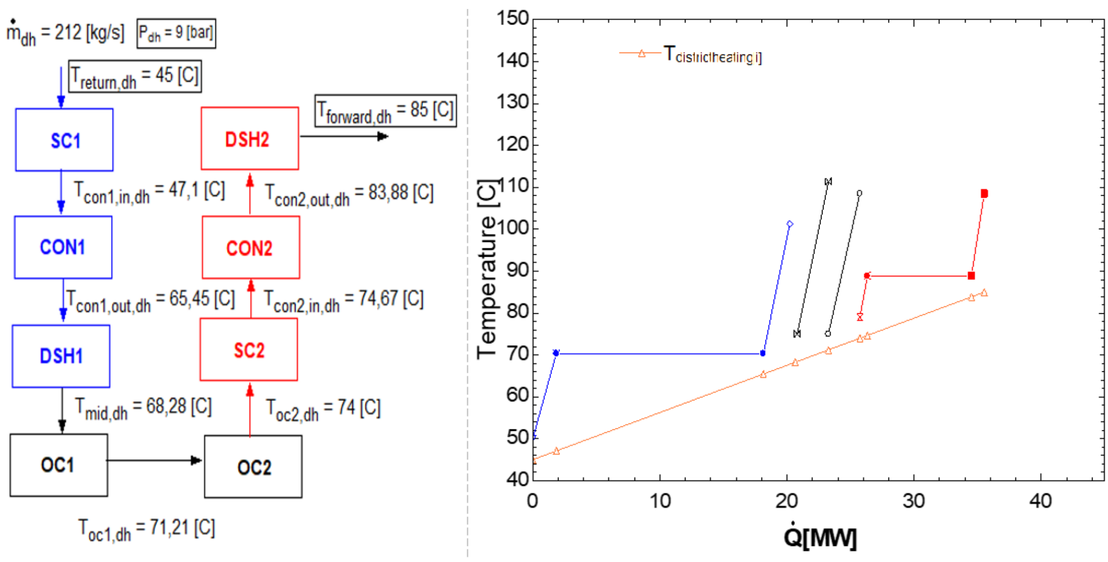
The first change purposed is to locate the OC in the middle of both HP, in this way the condensing temperature of the HP1 is lower, which makes the power in the compressor, because the discharge temperature is reduced. The HP2 has a similar condensing temperature to the other options.

- $COP_{HP1} = 4.43$
- $COP_{HP2} = 3.758$
- $COP_{TOTAL} = 4.173$
- $\eta_{isen_{HP1}} = 0.76$
- $\eta_{isen_{HP2}} = 0.796$
- $\dot{W}_{compressor} = 8.51\ MW$
- $\dot{Q}_{condenser} = 35.51\ MW$
- $T_{C_{HP1}} = 70.45$ °C
- $T_{C_{HP2}} = 88.88$ °C

The fact that the OC are placed after the HP1, increases a lot its COP, and due to its capacity is the double than in HP2, this raise is really influent for the total COP. The path and the T-Q diagram can be observed in figure 6.5. In this configuration the inlet oil temperature has to be increased until 75°C, in order to keep a reasonable difference with the DH.

As can be noticed, the overall COP increased with the strategy of decreasing the condensing temperature of the HP. The problem is that is not possible to locate the OC after the HP2, due to the temperature of the water at that point could be around 82-83°C, and is not possible to simulate the oil entering at that temperature.

By last, the compressors found for this model have an isentropic efficiency higher than in the case 0, this fact also makes the COP better.



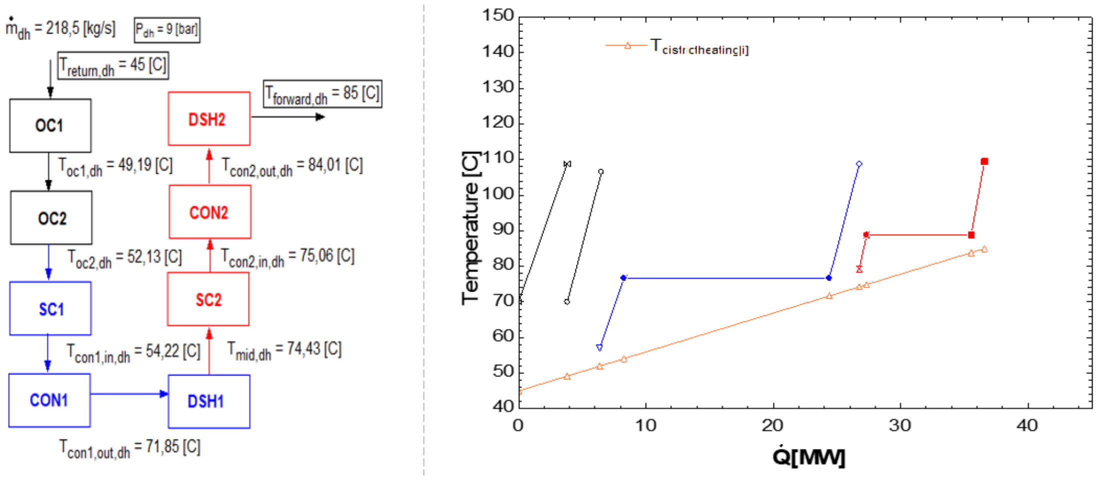
6.5. First configuration one-stage HP.

Second configuration

This configuration places the OCs at first. This strategy is adopted to keep the inlet compressor oil at 70°C, because if are located in the middle maybe is not possible to heat the DH water.

- $COP_{HP1} = 3.895$
- $COP_{HP2} = 3.663$
- $COP_{TOTAL} = 3.813$
- $\eta_{isen_{HP1}} = 0.739$
- $\eta_{isen_{HP1}} = 0.772$
- $\dot{W}_{compressor} = 9.597\ MW$
- $\dot{Q}_{condenser} = 36.6\ MW$
- $T_{C_{HP1}} = 76.85$ °C
- $T_{C_{HP2}} = 89.01$ °C

The simulation shows that the COP is worse in this case than in the base case. This could be explained with a increase of the condensing temperature at HP1, caused by the positioning of the OCs, which are the first exchangers in this case. Also the isentropic efficiency is worse for the HP1, that is why the low COP in that HP.



6.6. Second configuration one-stage HP.

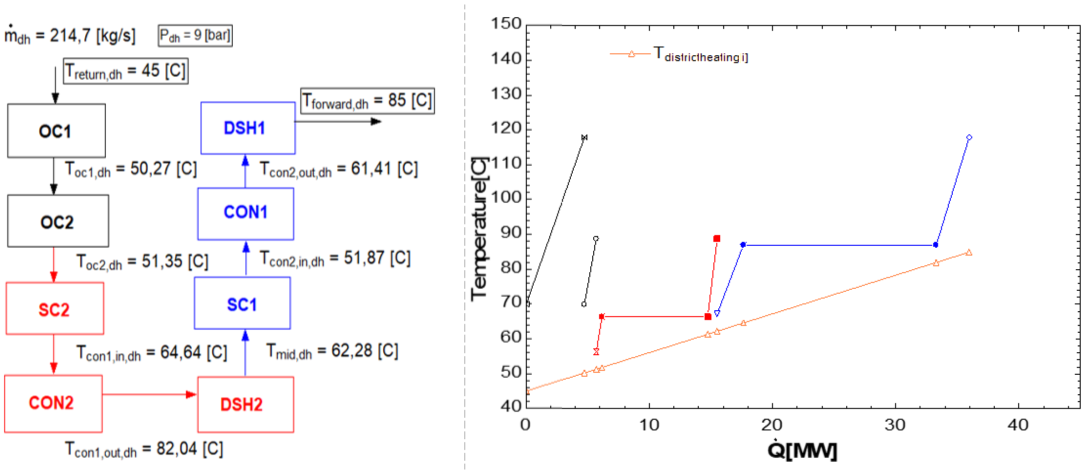
Third configuration

In this case, the OC are placed at the beginning, and the order of the HP is swapped. At first the HP2 and then the HP1. Changing the order is an interesting strategy to follow to check what is happening in the system.

- $COP_{HP1} = 3.511$
- $COP_{HP2} = 6.049$
- $COP_{TOTAL} = 4.016$
- $\eta_{isen_{HP1}} = 0.784$
- $\eta_{isen_{HP1}} = 0.855$
- $\dot{W}_{compressor} = 8.952\ MW$
- $\dot{Q}_{condenser} = 35.95\ MW$
- $T_{C_{HP1}} = 87.05\ °C$
- $T_{C_{HP2}} = 66{,}35\ °C$

Now, the condensing temperature of the HP2 is very low, because the heat of the HP1 is transferred after the HP2. This makes the COP of the second HP increase a lot, until 6, but for the overall COP is not that worth due to the huge decrease of the COP for the first HP.

In addition, the isentropic efficiency for the compressors that have been found is really good, 0,784 and 0,855. The isentropic efficiency is a key point for improving the COP. The conclusion that can be extracted here is that is better to prioritize the efficiency of the HP1 instead of the HP2, due to its capacity and impact on the model.



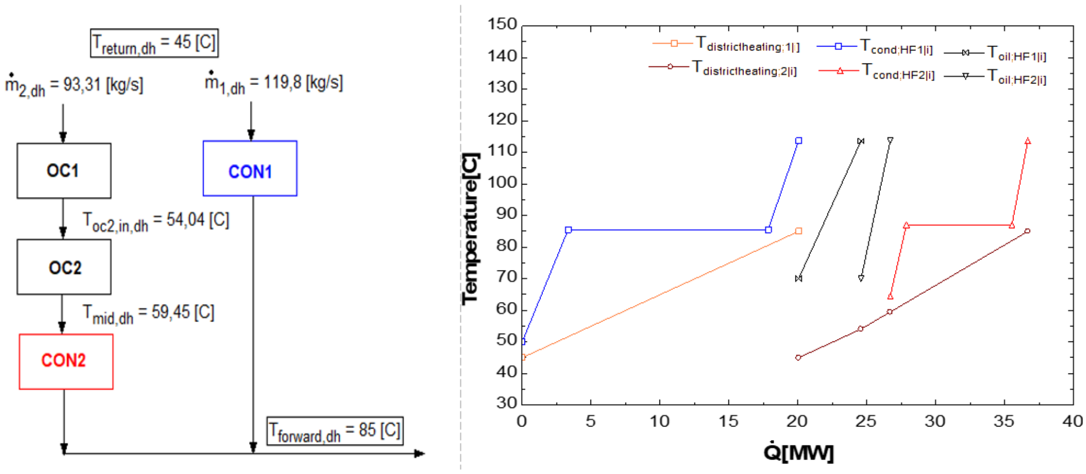
6.7. Third configuration one-stage HP.

Fourth configuration

For the fourth option, the strings are going to be split in two different strings. The OC will be placed with the HP2 and the HP1 will need the capacity to heat up a part of the DH mass flow from 45 to 85°C.

- $COP_{HP1} = 3.736$
- $COP_{HP2} = 3.901$
- $COP_{TOTAL} = 3.789$
- $\eta_{isen_{HP1}} = 0.766$
- $\eta_{isen_{HP1}} = 0.748$
- $\dot{W}_{compressor} = 9.681\ MW$
- $\dot{Q}_{condenser} = 36.68\ MW$
- $T_{C_{HP1}} = 85.58$ °C
- $T_{C_{HP2}} = 87.05$ °C

The results shows that split the flow is not a good plan to keep going, because the condensing temperatures of both HP have increase strongly, and therefore the COP of each HP and the overall decreases. The condensing temperature augment due to both HP have to make the DH mass flow until 85°C, so the temperature at the HP should be in both HPs higher than that value.



6.8. Fourth configuration one-stage HP.

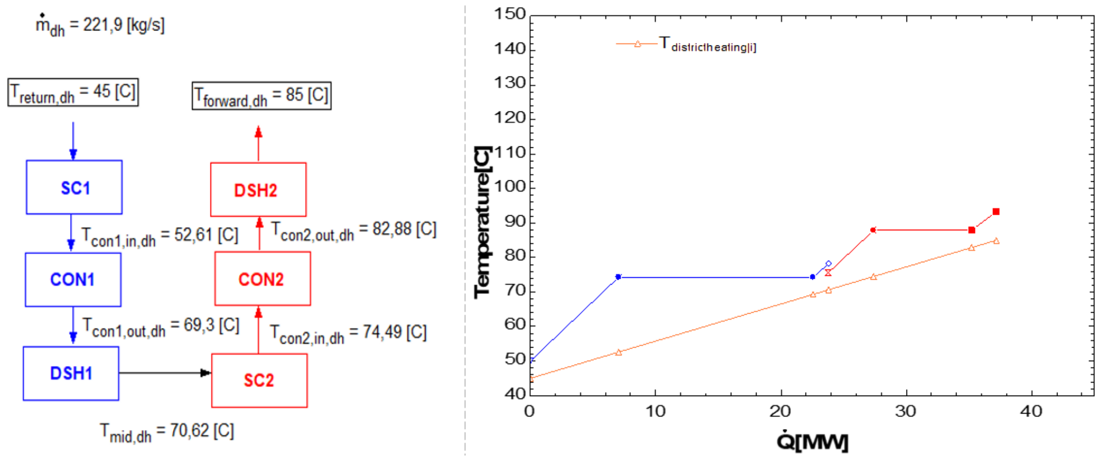
Fifth configuration

The last purposed configuration has a different refrigerant, in this case a HFO is used, particularly the R1234yf. This working fluid is not the best HFO in terms of performance, but it is in security and emissions.

For this kind of refrigerant, a piston compressor is need, and the isentropic efficiency and the heat losses are assumed as in the cases in Chapter 4: Thermodynamic model. For this reason, there is not OCs in this model and the configuration is quite simple.

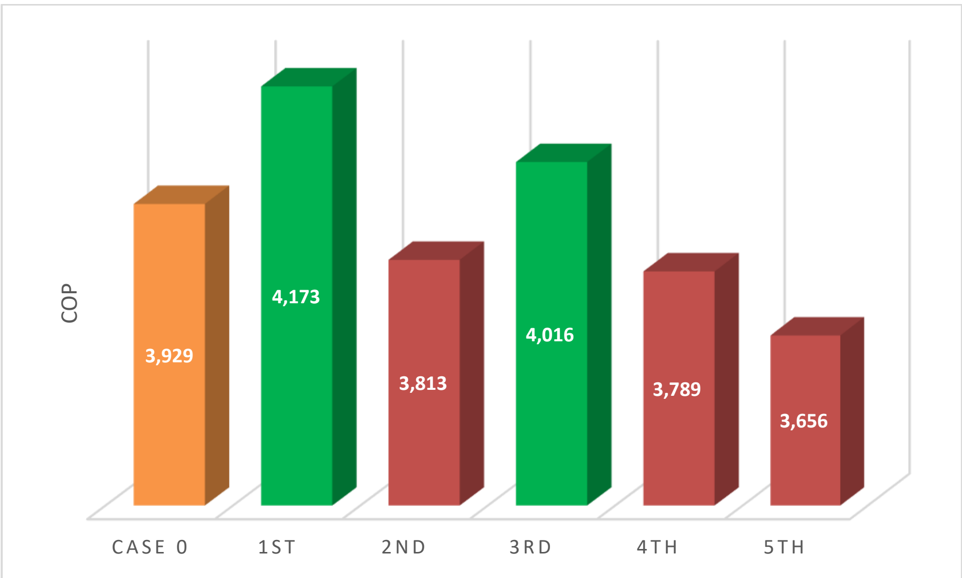
- $COP_{HP1} = 4.114$ • $COP_{HP2} = 3.052$
- $COP_{TOTAL} = 3.656$
- $\eta_{isen_{HP1}} = 0.75$ • $\eta_{isen_{HP1}} = 0.75$
- $\dot{W}_{compressor} = 10.166\ MW$ • $\dot{Q}_{condenser} = 37.17\ MW$
- $T_{C_{HP1}} = 74.3\ °C$ • $T_{C_{HP2}} = 87.88\ °C$

The discharge temperature is much lower in this case comparing with the Ammonia configurations, even the COP is worse. It can be observed that the HP1 works at a condensing temperature of 74°C, and its COP is 4.114. On the other hand, in HP2 at a higher condensing temperature the COP decreases until 3, which is a value really bad comparing with the other cases. From this fact can be deduced that the HFOs do not work properly at high temperatures, and then its use for this kind of application is limited.



6.9. Fifth configuration one-stage HP.

With all the results of the configurations, the graph 6.10 shows which the COP of each configuration is. And can be noticed that the best option is the Configuration 1, the one with the OCs placed in the middle. Consequently, the pinch analysis will be applied in the next section to that configuration.



6.10. COP comparison of the configurations.

In the next table 6.1 are the main parameters of the HP for each case, and is showed as a percentual difference, in comparison with the case 0. The conclussion that can be made from the table is that the variable that has more impact is the condensing temperatue of the HP1. For instance, the 3rd configuration has the best reduction of the condensing temperature for the HP2, and also the best isentropic efficiency, but the condensing temperature increases, so in the end the COP is not as good as it may llok for the other parameters.

Configuration	COP	$Q_{condenser}$	$W_{compressor}$	T_{CHP1}	T_{CHP2}	$\eta_{isen_{HP1}}$	$\eta_{isen_{HP2}}$
1st	6,21%	-1,96%	-7,69%	-4,53%	-0,26%	1,33%	6,13%
2nd	-2,95%	1,05%	4,10%	4,15%	-0,11%	-1,47%	2,93%
3rd	2,21%	-0,75%	-2,90%	17,97%	-25,54%	4,53%	14,00%
4th	-3,56%	1,27%	5,01%	15,98%	-2,31%	2,13%	-0,27%
5th	-6,95%	2,62%	10,27%	0,69%	-1,38%	0,00%	0,00%

6.1. Percentual difference of purposed configurations with case 0.

6.1.4 Pinch Method Optimization

In the application of the pinch method, for the one-stage scenario there are four key temperatures to optimize, showed in the table below.

Key temperature	Designation	Pinch temperature difference
$T_{HP1}[5]$	Temperature after SC in HP1	$\Delta T_{SC1} = T_{HP1}[5] - T_{return_{DH}}$
$T_{HP2}[5]$	Temperature after SC in HP2	$\Delta T_{SC2} = T_{HP2}[5] - T_{OC2_{DH}}$
$T_{C_{HP1}}$	Condensing Temperature in HP1	$\Delta T_{C_{HP_1}} = T_{C_{HP_1}} - T_{CON1in_{DH}}$
$T_{C_{HP2}}$	Condensing Temperature in HP2	$\Delta T_{C_{HP_2}} = T_{C_{HP_2}} - T_{CON2in_{DH}}$

6.2. Key temperatures for the one-stage HP.

The procedure to develop the pinch method has been explained in 5.2 Pinch method applied to heat pumps, in this case is a one-stage HP model, but the strategy to follow will be the same.

The next step is to simulate at first the model in EES with the pinch temperatures set as 5K, update the guess values, and at that point remove the four ΔT at condenser and the subcoolers. Then make an optimization taking the COP as the variable to maximize. The method to apply is the variable metric method.

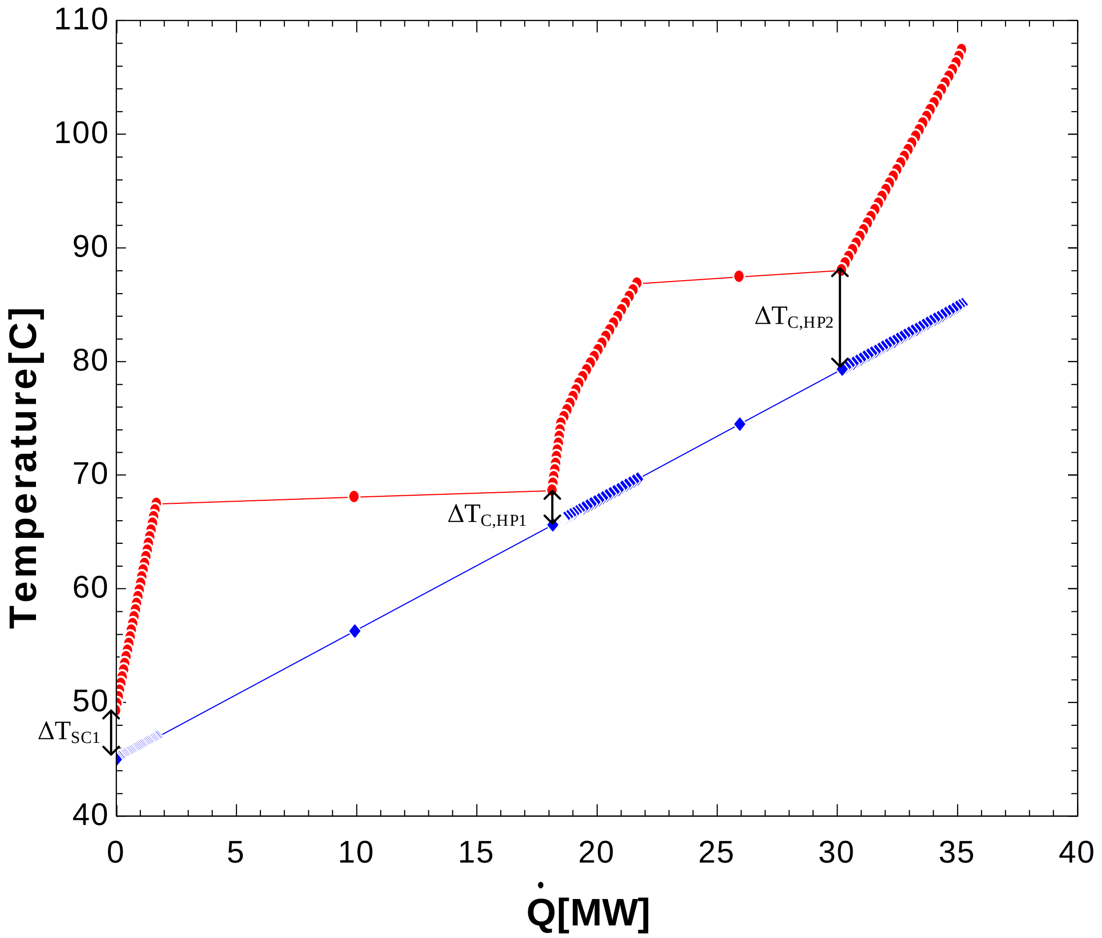
As a result of the simulation, the temperature is discretized into ranges, calculating the specific $\Delta T_{interval}$, and the heat exchanged in each interval is obtained as well. On the other hand, the composite curves and the output ΔT between the hot and cold stream are determined by the simulation. In addition, the resulting COP and pinch temperatures are:

- $COP_{HP1} = 4.593$
- $COP_{HP2} = 3.826$
- $COP_{TOTAL} = 4.295$
- $\dot{W}_{compressor} = 8.194\ MW$
- $\dot{Q}_{condenser} = 35.19\ MW$
- $T_{C_{HP1}} = 68.06$ °C
- $T_{C_{HP2}} = 87.52$ °C
- $\Delta T_{SC1} = 4.268$ °C
- $\Delta T_{SC2} = 4.429$ °C
- $\Delta T_{C_{HP1}} = 2.399$ °C
- $\Delta T_{C_{HP2}} = 3.639$ °C

Comparing with the previous model without optimization, the performance of the cycle has been increased in all its forms, and the compressor power and the capacity have decreased, specially the compressor power, this fact can be explained with the reduction of the condensing temperatures, which makes the compressor work with a better pressure ratio.

The new ΔT for the exchangers are the variables that change all the results, and is quite obvious that 5K as pinch it was too much and in any case would be a good option.

Other important thing to consider is that for every interval of the ΔT between the cold and hot streams, ΔT[i] was simulated taken 3K as a minimum, and can be noticed in the figure 6.11, for the pinch point at the condenser of HP1.



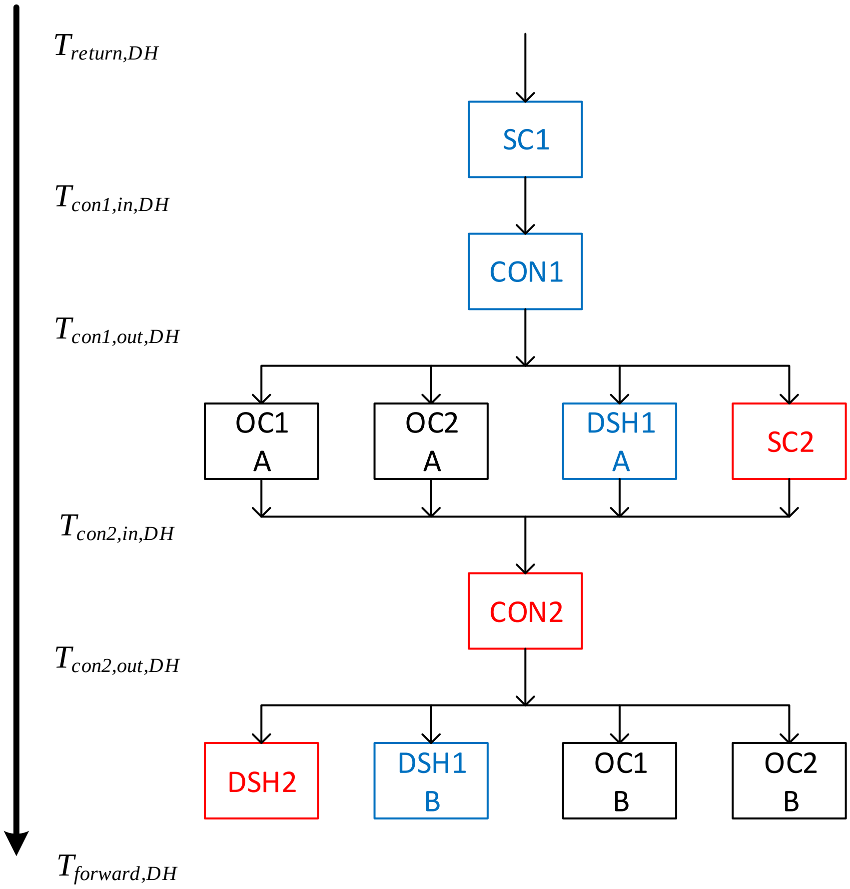
6.11. Composite curves for the one-stage HP.

The next step to ensure that the optimization could work is to find a Heat Exchanger Network that its temperature difference is above the 3K in any point.

6.1.5 Heat Exchanger Network

The purposed Heat Exchanger Network splits the exchanger on the condensing temperature. That means that the condensers will have all the flow and the rest of the exchangers will distribute the flow depending on its temperature range. In the case that one exchanger goes through the condensing temperature, it will be divided at that temperature. The minimum pinch temperature should be respected in every heat exchanger.

The HEN made for this configuration can be observed in figure 6.12, and has 11 different exchangers, six of them are the same exchangers as before but split in two sections.



6.12. HEN for the one-stage HP.

In order to simulate the HEN, some parameters of the previous simulations have to be taken, such as the temperature at the outlet of the DSHs, or at the inlet of the SCs, but the most important is the condensing temperatures.

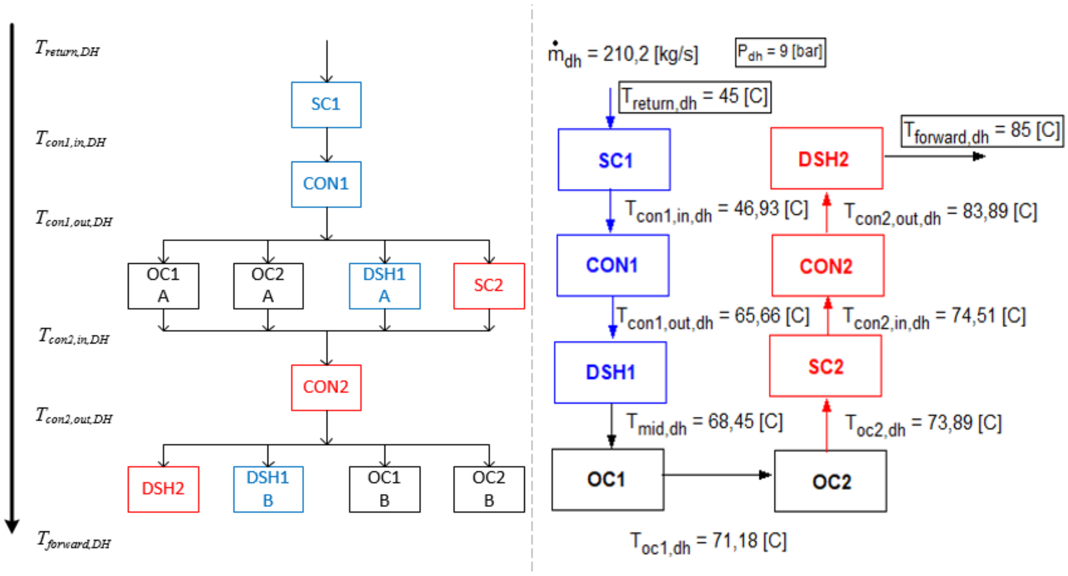
Once the simulation is done, the temperature differences are showed in table

HEX	**ΔT inlet**	**ΔT outlet**	**$\dot{Q}$ (MW)**	**HEX**	**ΔT inlet**	**ΔT outlet**	**$\dot{Q}$ (MW)**
SC1	4.27	21.13	1,69	CON2	17.56	8.18	8.26
CON1	21.13	2.4	16.46	DSH2	8.18	22.38	0.98
OC1 A	9.34	17.56	0.96	DSH1 B	8.18	21.32	1.09
OC2 A	9.34	17.56	0.92	OC1 B	8.18	21.32	1.44
DSH1 A	2.4	17.56	1.35	OC2 B	8.18	22.38	1.46
SC2	12.66	17.56	0.55				

6.13. Temperature differences in HEN one-stage.

The results in the table shows that the temperature at the outlet of the first condenser is below 3K, then the minimum is violated and this configuration will not be feasible to implement in reality.

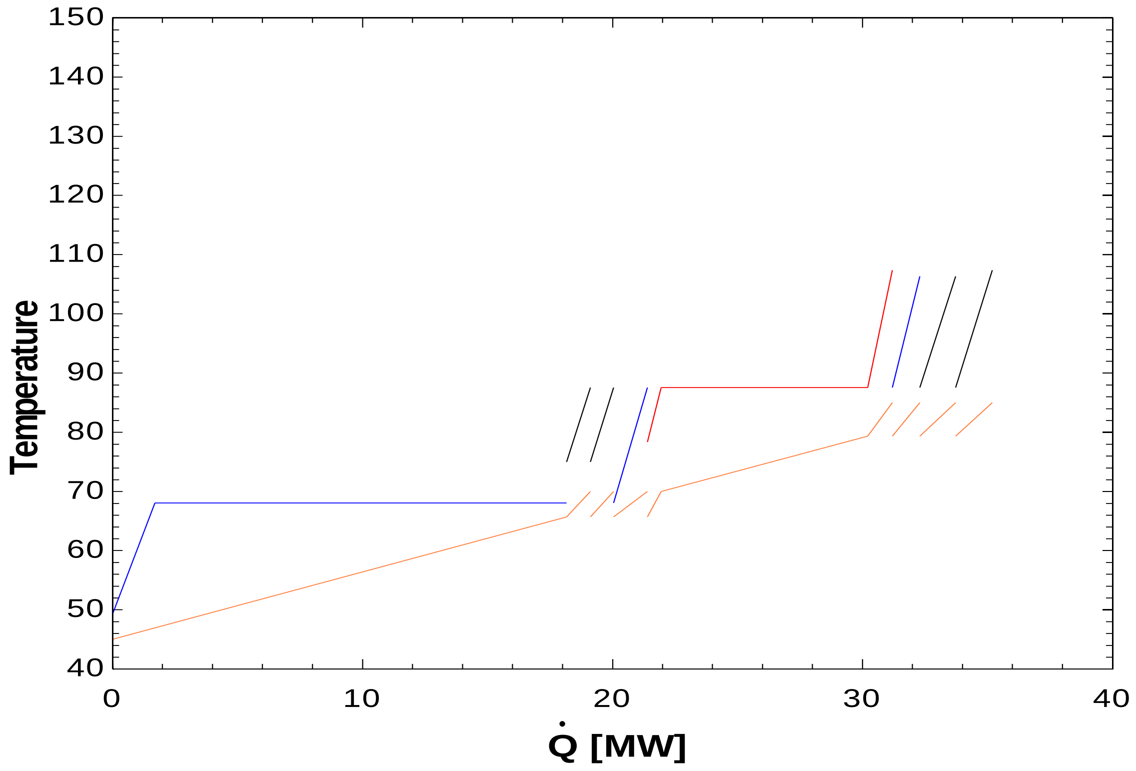
The reason why this HEN has failed is because with the current configuration it was not possible to find a HEN. This can be explained looking at the exchangers:



6.14. Comparison of exchangers CON1/HEN one-stage HP.

The temperature difference that does not fit with the minimum is the one of the outlet at the condenser of HP1. The first two exchangers of both configurations are the SC and the CON1, and this order can not be changed and additionally is not possible to include any other exchanger due to the temperatue range. The problem can be observed as well in figure 6.15, where the condensing temperature is that low that any other exchanger can be placed in this position.

Therefore, since the condensing temperature of the HP1 is obtained by means of the optimization and is selected as an input for the HEN, meanwhile $T_{con1,out,DH}$ is calculated with the only two exchangers that actually can transfer heat at that range, the ΔT for the outlet of the condenser will always be equal to the pinch difference obtained in the pinch method optimization, at 6.1.4, Pinch Method Optimization.



6.15. T-Q diagram of HEN one-stage HP.

6.2 TWO STAGE

The initial model corresponding to the two-stage HP was introduced in 4.2 Two stage model, and the assumptions were explained as well, with a fixed ΔT of 5K for all the exchangers.

The main results of the simulations are:

Config	$COP_{HP1}/\ COP_{HP2}/\ COP_{TOTAL}$	$\dot{W}_{compressor}/\dot{Q}_{condenser}$	$T_{C_{H_{HP1}}}/T_{C_{L_{HP1}}}$
Case0	4.671 / 4.178 / 4.49	7.735 / 34.74	73.28 / 34
Config	$T_{C_{H_{HP2}}}/T_{C_{L_{HP2}}}$	$\eta_{isen_{H_{HP1}}}/\ \eta_{isen_{L_{HP1}}}$	$\eta_{isen_{H_{HP2}}}/\ \eta_{isen_{L_{HP2}}}$
Case0	88.95 / 45	0.821 / 0.75	0.806 / 0.798

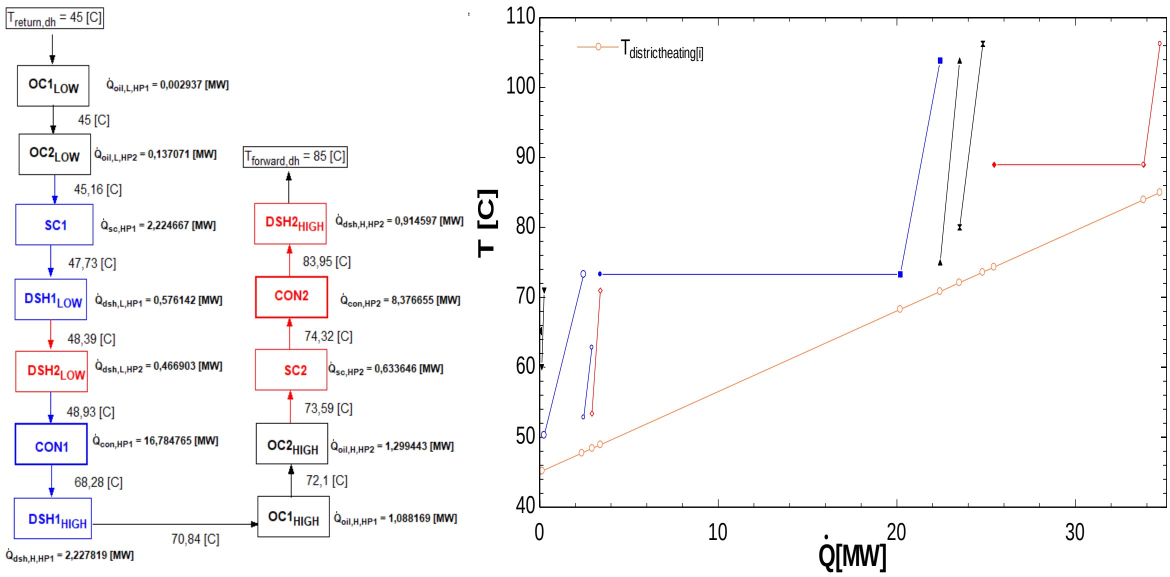
6.3. Results of the two-stage HP case 0.

Looking at the results is clear that the two-stage cycle is much efficient that the other cases studied before. Even without the pinch optimization the COP is better than the best one-stage configuration optimized.

The huge increase of the performance could be explained due to the reduction of the pressure ratio at the compressors, which at the end makes the total power be around 1-2 MW lower.

Regarding the intermediate temperatures, are a key parameter in the two-stage model. In this case at first it was tried to find the optimal value for the performance, which were 34ºC and 50ºC respectively for HP1 and HP2, but in reality the optimal point is not just depending on the temperature, the efficiencies of the compressors affect as well. So the optimal point was found looking at the catalogue of RTselect for the compressors at more or less the same range as the theoretical optimum, taking care about the isentropic efficiency of each compressor. Finally the compressors found reaches a quite good efficiency, but the second intermediate temperature has been changed to 45ºC.

The oil inlet temperature is a bit lower for the first two OCs, 65ºC for the HP1 and 60ºC for the HP2. Reducing the temperature of the oil the ammonia is cooled easily, this fact makes the COP slightly increase.



6.16. T-Q diagram of the two-stage HP case0.

In the figure 6.16 is observed how the exchangers of the low stage have really few impact on the model, specially the OCs, but with screw compressors and ammonia is easy to implement the OCs and in the end they aport something to the system cooling down the ammonia flow at the compressor.

The condensing temperatures of the high stage are similar to the condensing temperatures of the one-stage model.

Alternatively, other configurations are purposed, to set which are the most influding parameters and also how affects to the system the use of another refrigerant, such a HFO.

First configuration

This configuration in particular has changed the order of the exchangers, and the model of the compressor aswell. Now, the OCs are placed between both HP, therefore, the temperature of the oil has to be changed to heat the DH line, which has a temperature around 70-75ºC in that part. The oil temperature at the inlet of the OC has been increased to 80ºC for all the exchangers, otherwise it was not possible to keep a pinch difference with the water.

The values of the main variables are stated in table 6.4.

Config	$\boldsymbol{COP_{HP1}/\ COP_{HP2}/\ COP_{TOTAL}}$	$\boldsymbol{\dot{W}_{compressor}/\dot{Q}_{condenser}}$	$\boldsymbol{T_{C_{H_{HP1}}}/T_{C_{L_{HP1}}}}$
Case 1	4.595 / 4.397 / 4.53	7.654 / 34.66	75.96 / 42
Config	$\boldsymbol{T_{C_{H_{HP2}}}/T_{C_{L_{HP2}}}}$	$\boldsymbol{\eta_{isen_{H_{HP1}}}/\ \eta_{isen_{L_{HP1}}}}$	$\boldsymbol{\eta_{isen_{H_{HP2}}}/\ \eta_{isen_{L_{HP2}}}}$
Case 1	88.93 / 46	0.78 / 0.83	0.8 / 0.76

6.4. Results configuration 1 two-stage HP.

In this case the decision to set the intermediate temperature was the same as before, but in this scenario the best solution was 49ºC for the HP1 and 50ºC for the HP2. In a real practical case it is not possible to find a compressor that can work with that temperatures at the high stage. For this reason, the temperatures have been reduced to find a compressor with the best isentropic efficiency reachable.

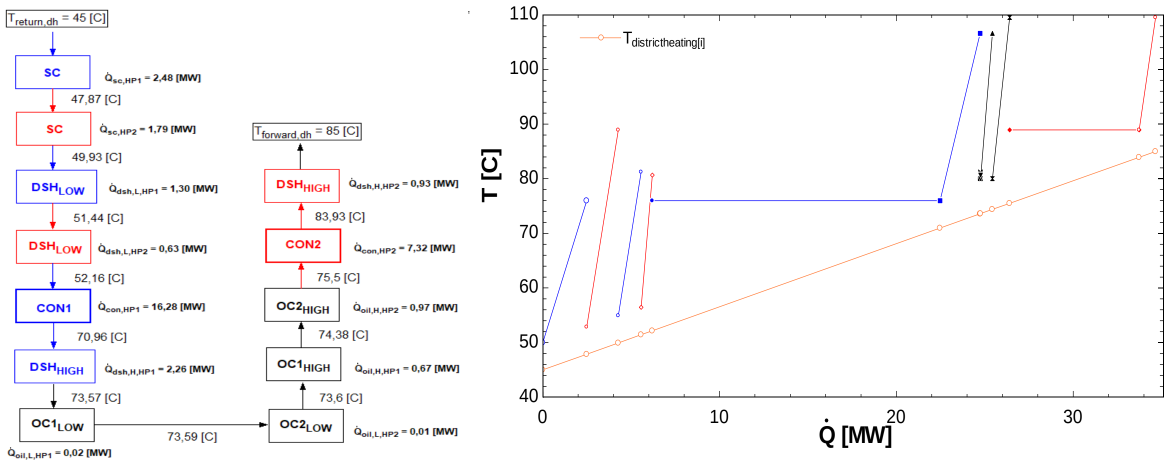
The COP for this configuration has increased comparing with the base situation. The raise of the performance can be explained because of the reduction of the compressor power in the HP2. In the table 6.4 can be noticed that the compressor power and the condenser capacity have both decreased, but the compressor power a -1,05% and the capacity a -0,23%, that makes the COP bigger for this configuration.

Splitting the results in deep, the power of each compressor is (in MW):

Configuration	$W_{H_{HP1}}$	$W_{L_{HP1}}$	$W_{H_{HP2}}$	$W_{L_{HP2}}$
Case 0	3,196	1,705	1,905	0,925
Case 1	2,726	2,282	1,629	1,02

6.5. Compressor power comparison two-stage HP.

With the table 6.5 is easy to say that the difference of performace is due to the reduction of the power for the high stage compressor of the second HP, the evidence is in table 6.4, where the COP of the HP2 is better than in table 6.3. This decrease is caused by the sligh increase of the mid temperature. So as a conclussion, it can be said that one of the parameters that infludes the most to the COP is the intermediate temperature.



6.17. T-Q diagram of the configuration 1 for two-stage HP.

HFO configuration

By last, other refrigerant is tried in order to compare which are the possibilities of the system, even Ammonia should be the best option is interesting to check how much performance would be lose if other working fluid is selected.

The refrigerant to try is a HFO, concretely, R1234yf. As an initial consideration, OC is not an option for this scenario. This is because the HFO do not reach high temperatures at all, being the maximum at 80-90ºC approximately, and therefore is not needed an oil flow to cool down the refrigerant. This is an advantage in terms of equipment and complexity of the cycle, but since that heat is not profited in an external exchanger, the performance will decrease.

In addition, for the configurations with HFO as a working fluid in a two-stage HP, is not conceivable to have a low stage DSH. This kind of exchanger takes profit of the assumable high temperature at the outlet of the compressor in the low-stage part of the cycle. The problem for HFOs is that do not arrives until high temperatures on this stages, being its temperature in the outlet of the low stage compressor around 30-40ºC, and at this temperature the refrigerant is unable to heat the water flow.

With these considerations, the current configuration has only the two condensers as exchanger, so it is really simple. The compressors used will be piston, with an assumed isentropic efficiency of 0.75 and heat losses of 5%. The results are showed in table 6.6.

Config	$\boldsymbol{COP_{HP1}/ COP_{HP2}/ COP_{TOTAL}}$	$\boldsymbol{\dot{W}_{compressor}/\dot{Q}_{condenser}}$	$\boldsymbol{T_{C_{H_{HP1}}}/T_{C_{L_{HP1}}}}$
Case 1	3.867 / 3.815 / 3.85	9.308 / 35.88	80.647 / 30

Config	$\boldsymbol{T_{C_{H_{HP2}}}/T_{C_{L_{HP2}}}}$
Case 1	88.4 / 36

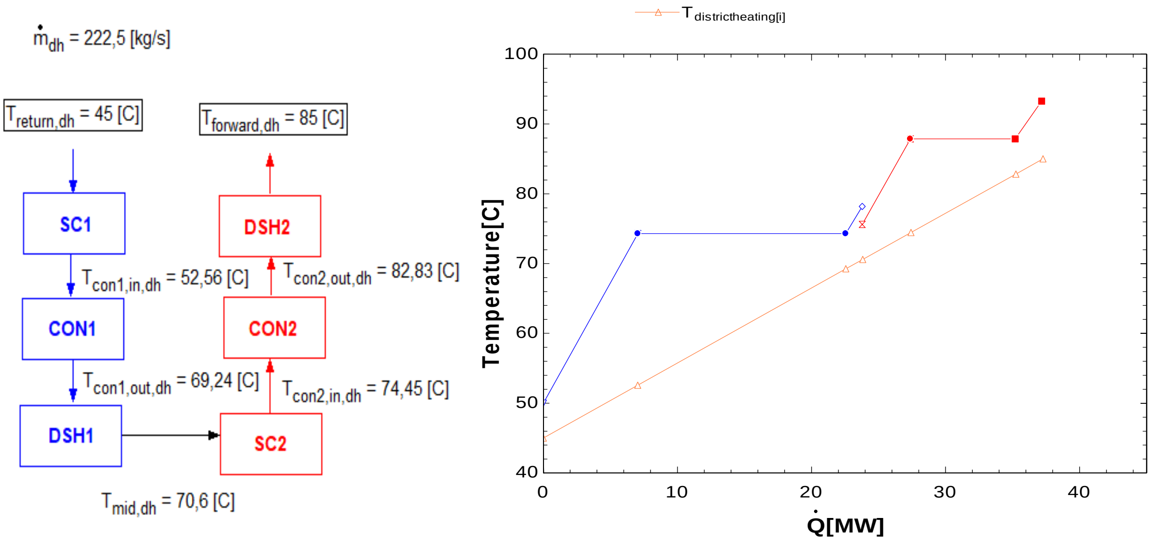
6.6. Results for the HFO configuration two-stage HP.

The first thing to emphasize is the low averall COP, caused by an important increase of the compressor power. This new refrigerant has other characteristics to ammonia, such a lower specific heat that makes the enthalpy small comparing with other working fluids for the same cycle. This lack implies that the flow to compensate the heat transferred should be huge.

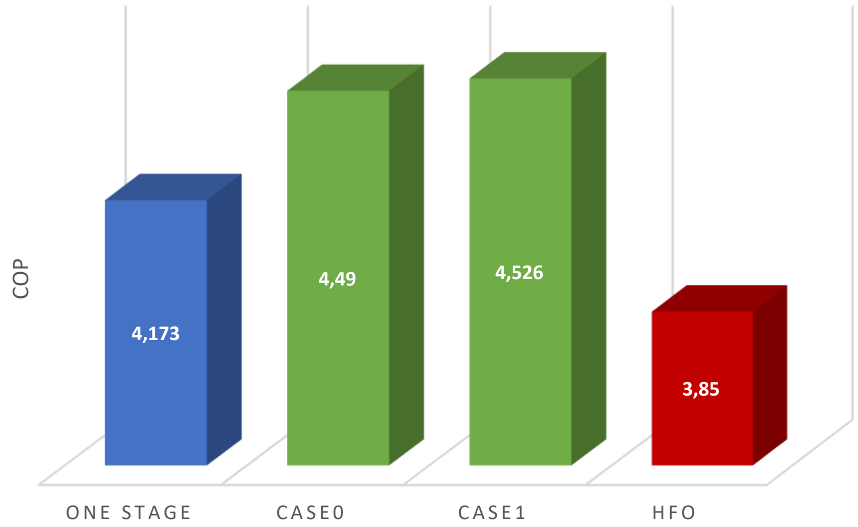
Comparing with the other cases, R1234yf requires a mass flow of 170kg/s at the high stage of the HP1, and Ammonia around 18kg/s, so the difference between the requirements of one cycle and the other is massive. This fact can explain the augment of the compressor power, additionally with the heat losses of the piston compressor.

The intermediate temperatures are determined by means of an optimization, without taking in care the isentropic efficiency of the compressors. Comparing the condensing temperatures with the other configurations do not have a lot of sense because are completely distinct.

In the figure 6.18 the path of the DH line and the T-Q diagram can be observed. As it was introduced before, the temperature at his highest point is slightly above 90ºC.



6.18. T-Q diagram of the HFO configuration of the two-stage HP.



6.19. Comparison between two-stage configurations.

In the figure 6.19 can be observed that the best solution for the two stage cycle is the case 1, and consequently, as in the one-stage case, the pinch analysys will be applied to optimize the model. Anyways, even the HFO configuration does not present a good performance at all, the pinch method is going to be applied as well to that configuration, in order to see the limits and until which point can arrive in terms of performance.

6.2.1 Pinch Method Optimization

For the two-stage configurations, the key temperatures were developed at table 5.5, there are 6 different pinch points that have to be optimized, and also including the intermediate temperatures. Is necessary to set the minimum temperature difference at 3K for the ΔT array containing the difference between the cold and hot streams. Once it is done and with a previous simulation to update the guess values, the variable metric method is applied in the optimization, and the result is:

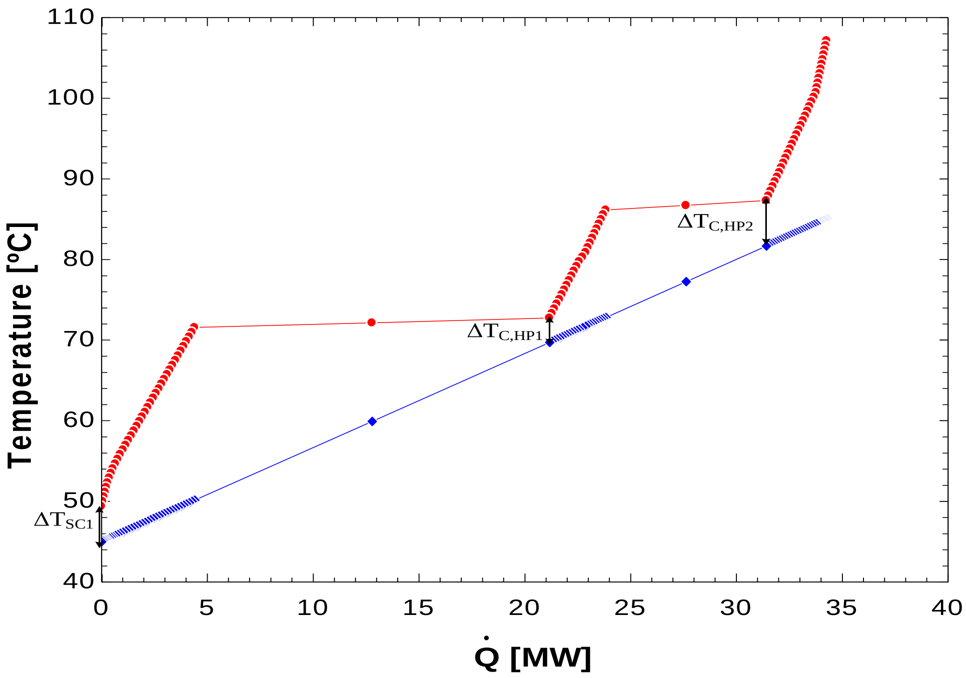
COP_{HP1}/ COP_{HP2}/ COP_{TOTAL}	$\dot{W}_{compressor}/\dot{Q}_{condenser}$	$T_{C_{H_{HP1}}}/T_{C_{L_{HP1}}}$
4.813 / 4.501 / 4.703	7.29 / 34.29	72.54/ 42.66
$T_{C_{H_{HP2}}}/T_{C_{L_{HP2}}}$	$W_{H_{HP1}}/W_{L_{HP1}}$	$W_{H_{HP2}}/W_{L_{HP2}}$
86.98 / 46.38	2.38 / 2.34	1.53 / 1.04
ΔT_{SC1} / ΔT_{SC2}	$\Delta T_{C_{HP1}}$ / $\Delta T_{C_{HP2}}$	$\Delta T_{DSH_{L_{HP1}}}$ / $\Delta T_{DSH_{L_{HP2}}}$
4.455 / 4.821	1.255 / 3.006	4.64 / 4.87

6.7. Results of the optimization for the case 1 two-stage HP.

The optimization shows a result with a COP of 4.7 that has improved the 3.8% comparing to the previous result. Reducing the pinch difference allows the condensing temperature to be closer to the DH circuit, and therefore, the condensing temperature is lower and will have a better performance due to the reduction of the pressure ratio.

In the table 6.5 is showed the power of each compressor, and making a comparison with the optimized results, the compressor that changes more with the original result is the one of the high stage in the HP1. $W_{H_{HP1}}$ is reduced from 2.726 to 2.38, which is a decrease of the 13%, making the HP1 much more efficient. The reduction can be explained by the cut down of the pressure at the highest stage.

The resulting composite curves have a minimum difference of 3K, which can be observed at the outlet of the first condenser, as the pinch point $\Delta T_{C_{HP1}}$ with exactly 3K. On the other hand, $\Delta T_{C_{HP2}}$ is equal to 5.49 and ΔT_{SC1} is 4.41.



6.20. Composite curves pinch optimization two-stage.

Regarding the HFO configuration, the optimization is made with 4 key temperatures, as in the one-stage scenario (table 6.2), corresponding to the condensing temperatures and the subcooling pinch difference.

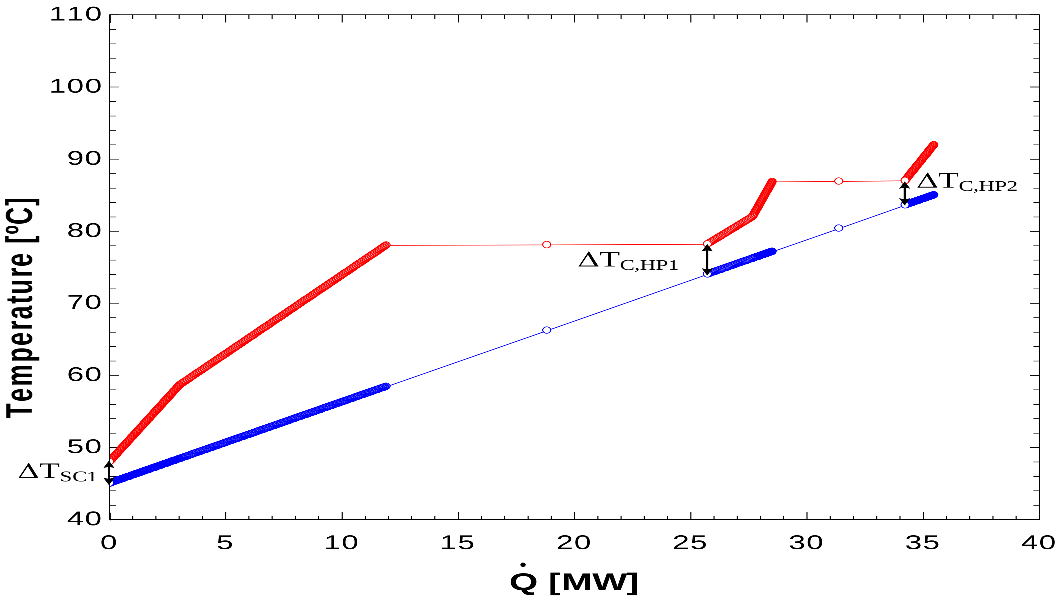
With this configuration it came other problem, the minimum temperature difference it was not of 3K with 100 steps, it was 3.29K, for this reason, it was tried with 300 steps obtaining 3.1K and 500 steps obtaining 3K, increasing a lot the calculation time. Normally is not recommended augment the number of steps like that for all the equations that generates, but in this case the analysis is going to be performed with 500 steps.

COP_{HP1}/ COP_{HP2}/ COP_{TOTAL}	$\dot{W}_{compressor}$/$\dot{Q}_{condenser}$	$T_{C_{H_{HP1}}}$/$T_{C_{L_{HP1}}}$
4.011 / 3.926 / 3.982	8.91 / 35.49	78.15/ 31,53
$T_{C_{H_{HP2}}}$/$T_{C_{L_{HP2}}}$	$W_{H_{HP1}}$/$W_{L_{HP1}}$	$W_{H_{HP2}}$/$W_{L_{HP2}}$
86.98 / 40.84	4.25 / 2.14	1.63 / 0.88
ΔT_{SC1} / ΔT_{SC2}	$\Delta T_{C_{HP1}}$ / $\Delta T_{C_{HP2}}$	
3.033 / 4.107	2.434 / 3.363	

6.8. Results optimization HFO configuration two-stage HP.

The results show that the averall COP has increased a 3.39%. The parameters that have change the most are the intermediate temperatures, both increasing and more in the case of HP2. At the same time, all the pinch temperatures are reduced and this makes the system more efficient, because the condensing temperature decreases in both HP, and then the pressure ratio. Quantitatively, the compressor power is cut down a 4.49%, this is the cause of the raise in the COP.

With these results, the composite curves in figure 6.21 presents that in this case the pinch point of 3K is set at the beginning of the system, in the SC1. Meanwhile in the other pinch points, the difference between the composite curves in the outlet of the frist condenser is 4.196K, and at the outlet of the second condenser is 3.419K.

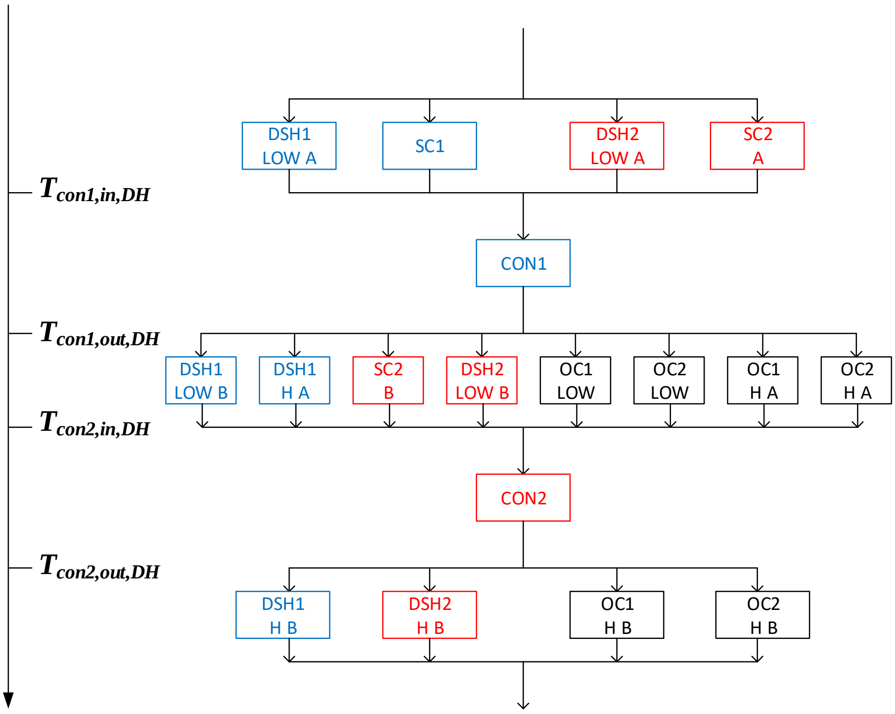


6.21. Composite curves HFO two-stage HP.

6.2.2 Heat Exchanger Network

In this section two different HENs will be purposed, one corresponding to the ammonia configuration, and other for the HFO. In both of them the exchangers have been split at the condensing temperatures, leaving all the flow going through the condensers.

The scheme of the Ammonia configuration can be looked at figure 6.22, it has 18 exchangers. In order to simulate it, the values of the pinch optimization have to be taken as inputs, and then all the flows through the exchangers are modelled and the simulation is done.



6.22. HEN for the Ammonia two-stage HP.

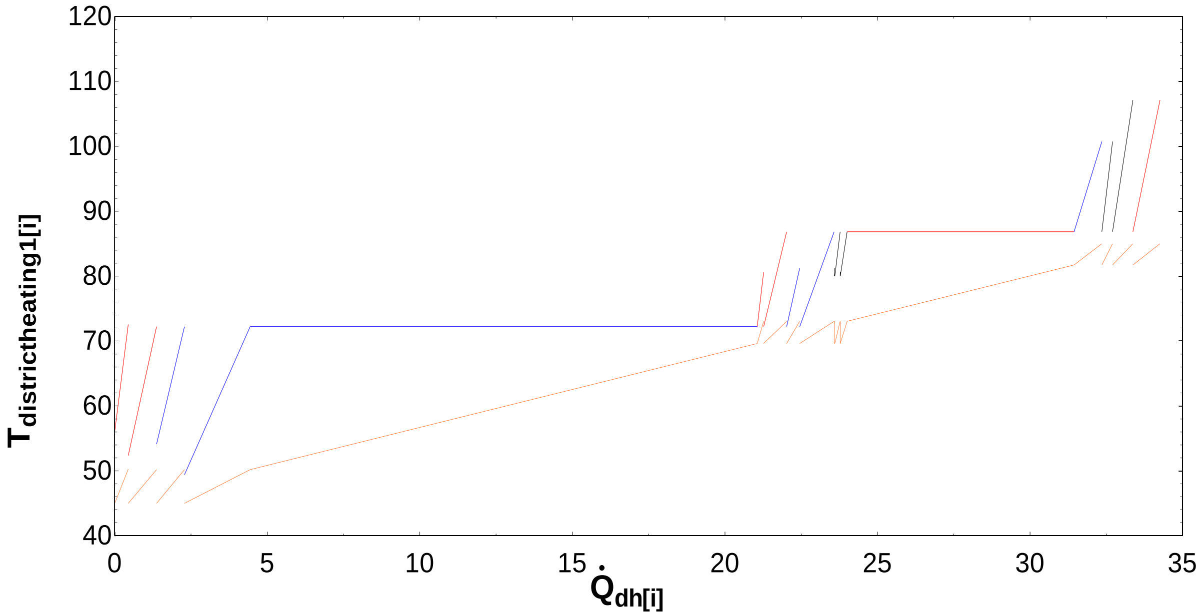
Once implemented the model and simulated, the results have to show the ΔT difference in the inlet and the outlet of each exchanger, and it has to be above the minimum pinch difference set before at the pinch optimization.

HEX	**ΔT inlet**	**ΔT outlet**	**$\dot{Q}$ (MW)**
DSH1 L A	9.17	22.28	0.94
SC1	4.45	22.28	2.176
DSH2 L A	11.05	22,28	0.44
SC2 A	7.37	22.28	0.95
CON1	22.28	2.94	16.57
DSH1 L B	2.94	8.29	0.47
DSH1 H A	2.94	13.86	1.12
SC2 B	2.94	13.86	0.75
DSH2 L B	2.94	8,22	0.22
OC1 L	10.4	8.29	0.04
OC2 L	10.4	8.22	0.01
OC1 H A	10.4	13.86	0.18
OC2 H A	10.4	13.86	0.23
CON2	13.86	5.22	7.42
DSH1 H B	5.22	15.52	0.89
DSH2 H	5.22	22.05	0.88
OC1 H B	5.22	15.52	0.34
OC2 H B	5.22	22.05	0.66

6.9. Temperature difference in HEN Ammonia two-stage.

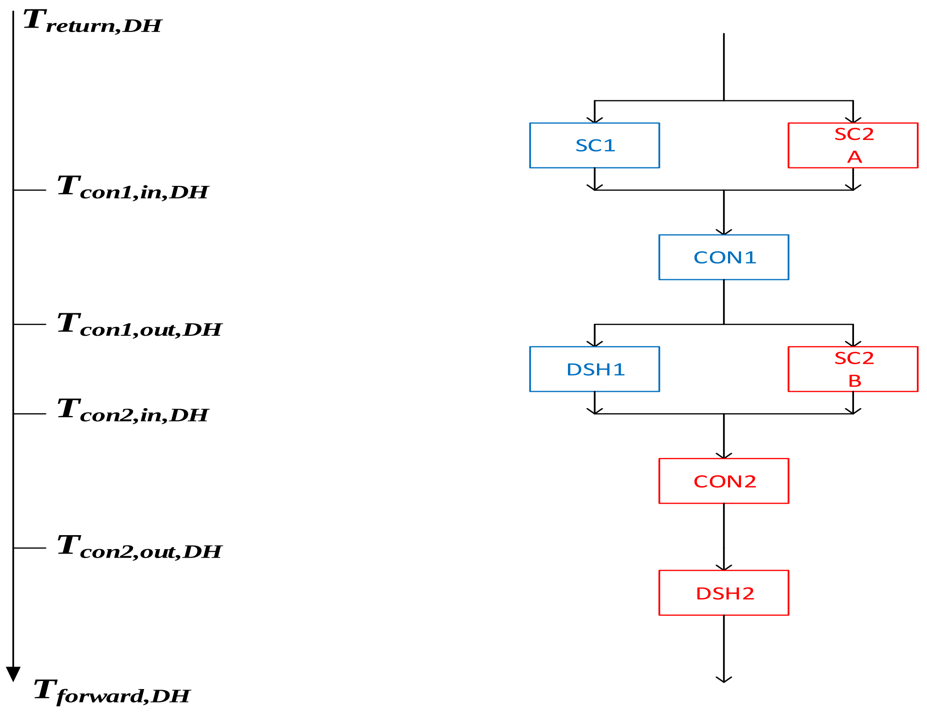
The table 6.9 shows the results of the HEN purposed for the Ammonia configuration, and is clear that not all the results are above the 3K, but 2.94K is a really close value to 3K. The consideration made is that this difference in the temperature could be due to the mismatch of the streams in the pinch, also is considered that the result took at the pinch optimization is “ideal” and therefore the application of the HEN will show results that can be close to them, but not exactly equal. The optimal result is just one, and this means that every result above or below has to be worse in the end.

Consequently, is considered that the Hen fits with the minimum requirement and the difference of 0.06K will have a neglective effect in a hypothetical real system. The T-Q diagram of the HEN is presented in figure 6.23, to see clearly the heat transfer in all the stages of the system.



6.23. T-Q diagram of the ammonia HEN.

The HFO HEN is much simple than the one for the ammonia, and it follows the same strategy of splitting the exchangers at the condensing temperature, because is the easiest way to find a HEN that fits with the requirements. The only exchanger that is divided is the SC corresponding to the second HP, in total there are 7 exchangers for this HEN.

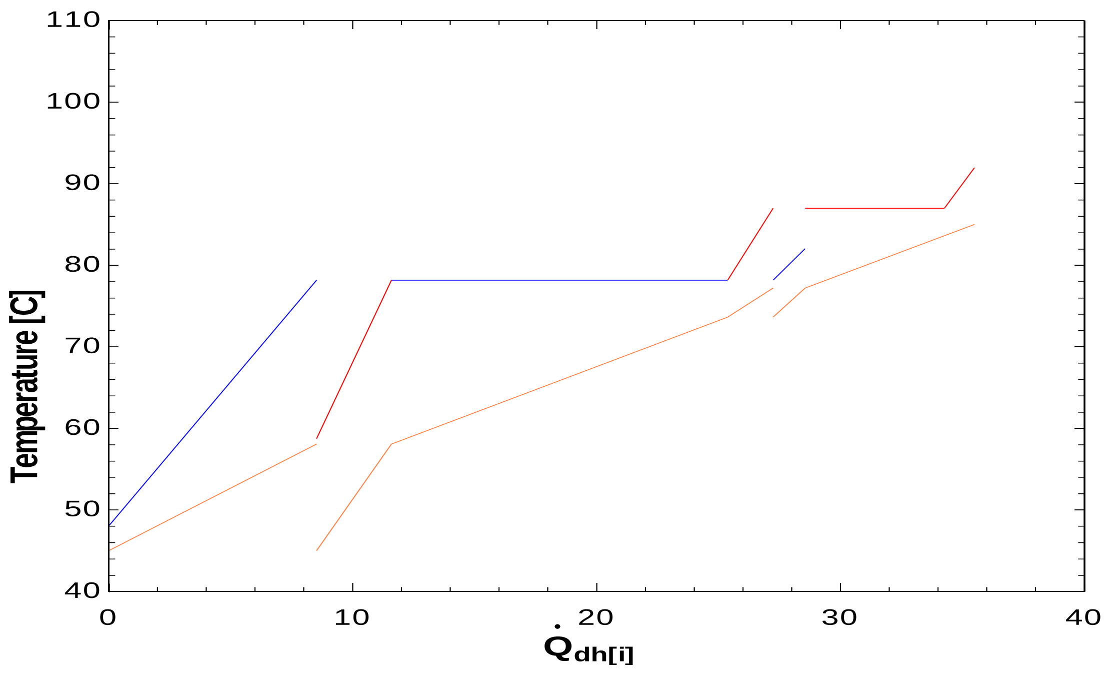


6.24. HEN HFO configuration two-stage HP.

HEX	ΔT inlet	ΔT outlet	Q̇ (MW)	HEX	ΔT inlet	ΔT outlet	Q̇ (MW)
SC1	3.033	20.07	8.51	SC2 B	4.527	9.788	1.86
SC2 B	13.7	20.07	3.07	CON2	9.788	3.363	5.71
CON1	20.07	4.527	13.79	DSH2	3.363	6.96	1.23
DSH1	4.527	4.845	1.30				

6.10. Temperature difference HFO HEN.

In this case is stated that the temperature difference that should be close to 3K is the inlet of the subcooling, because of the form that the composite curves were placed. The temperature difference at that point is almost 3K, and this comes directly from the optimization (table 6.8), where the ΔT_{SC1} was set as 3.033K. For this reason, is considered that this HEN could be feasible to implement this configuration, having a results close to the ones showed in the optimization.



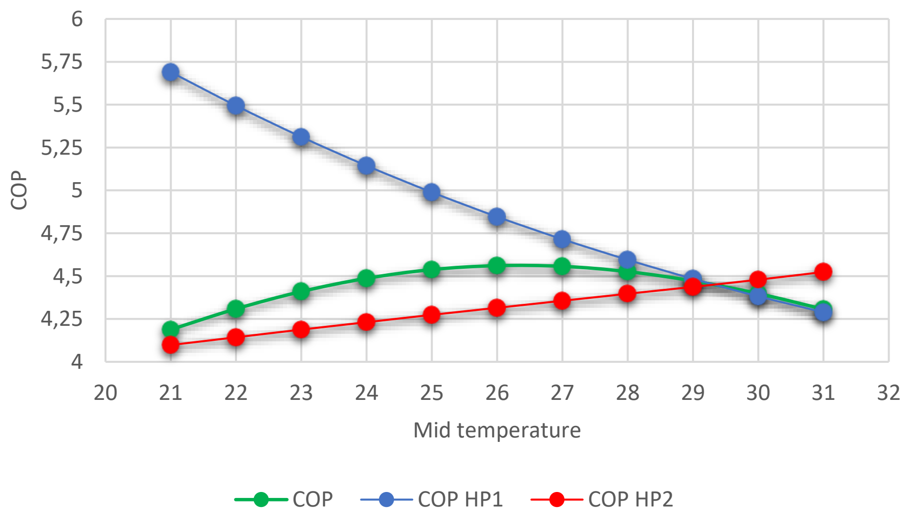
6.25. T-Q diagram of the HEN HFO.

6.3 SOURCE STUDY

In this section the inputs taken from the datacenter are going to be evaluated, since is one of the inputs that have more impact on the model, looking for temperatures that give a better performance to the system.

There are three different temperatures fixed in the source, the return temperature, at 32ºC, the mid temperature at 28ºC, and the forward temperature at 20ºC. The forward and return temperature are set by the conditions desired for the owner of the facility and the requirements of the cooling system which can be changed. But the mid temperature affects directly to the required capacity of the evaporators, which means that the capacity could be balanced to obtain a better performance in the HPs. Therefore, taking the model of the two-stage HP with ammonia that showed the best COP, and trying other mid temperatures, is possible to find a better solution.

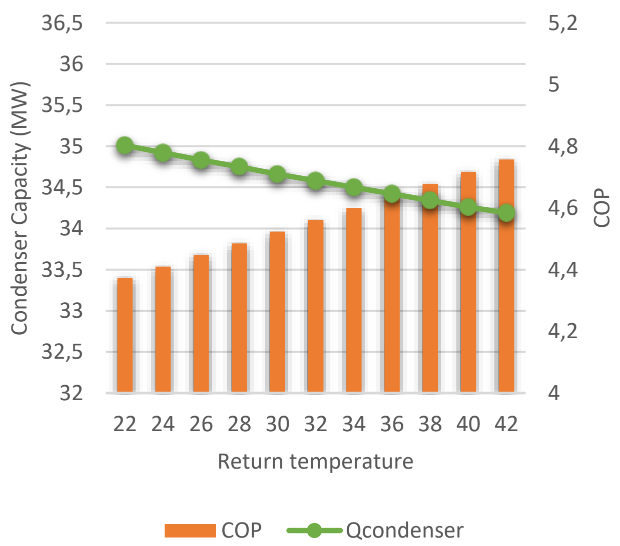
Then, in figure 6.26, is presented the mid temperature against the COP. It can be observed that the COP of the first HP decreases with the augment of the mid temperature, because the range of temperature is bigger. The opposite situation happens with the second HP. On the other hand, the overall COP has an optimum value of 4.561 at the mid temperature of 26ºC, this can be explained because the capacity of the evaporators is exactly balance by its half, having 13.5MW in each HP.

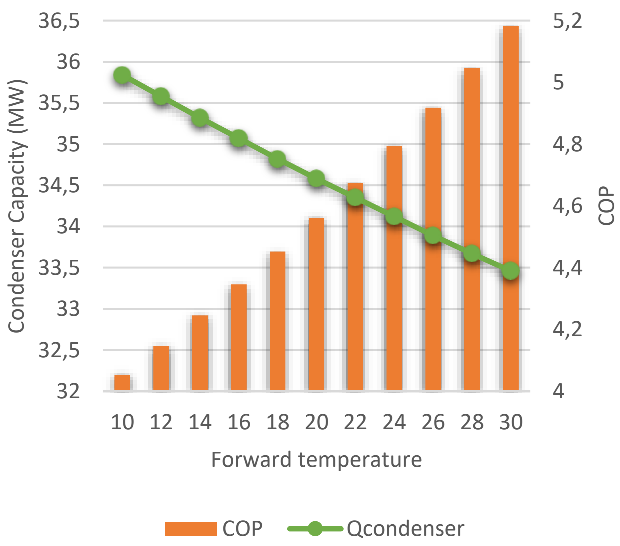


6.26. Changes in the mid source temperature.

So the best option for modelling the stream is splitting the total evaporator capacity in two identical capacities for each HP, and not the input taken in this project, where the first HP has the double capacity than the HP2.

The other thing that should be studied is how infludes the modification of the forward and return temperatures to the COP, in the figure 6.27 is done an evaluation of the COP and the condenser capacity with the changes at the temperature, assuming that both HP have the same capacity at the evaporator. Obviously, the COP is directly proportional to the raise of the temperature, in the case of the forward temperature because is easier to cool down a stream until a higher temperature, and for the return because is easier to extract heat if the stream is hotter.





6.27. Comparison of the source temperature changes.

The conclusion of this section is that is better to have the same evaporator capacity in the HPs, and the forward temperature has more influence in the final result than the return temperature, so it should be fixed with caution.

Chapter 7: Discussion

Even the study carried in this project is very interesting and has relevance to reach the emission reduction objectives of the Danish Government, the current state for the integration of large Heat Pumps is that there are not set officially objectives to achieve, and consequently is expected some targets to be released. The knowledge about large Heat Pumps has to be expanded in the next years, including its availability, limitations, Etc.

Regarding the inputs taken in the modelling, the range of temperatures and the capacity of the system has been obtained directly from an engineering report of a specific datacenter. But the reasons or assumptions that the company made are unknown, which means that maybe these temperatures are not realistic or are estimated by someone who does not have the proper information. Despite this, this temperature range has been submitted to an investigation to get the best performance, in case of this project could help to build a future facility. The final results of this project do not have to be compared with the initial PRIEBE report due to is not stated how the calculations are done in the other case.

Other consideration to think about is the $\Delta T_{evaporator}$. Is a parameter that has been fixed since the beginning of the project, at 5K, because it was desired to have a conservative value of it, like in the isentropic efficiency of the piston compressor, set in 0.75, but in a real facility, is feasible to have a lower value. The $\Delta T_{evaporator}$ is a variable really influent, for example, just with changing it to 2K, which is feasible in reality, the overall COP of the two-stage HP with ammonia increases from 4.526 to 4.751, incrementing a 4.7%. Although, if is applied the pinch it goes until 4.92, a huge improve taking in account the difficulty to boost the COP at this point.

For the modelling for the configurations, it was tried to keep at the inlet of the OC a pinch with the DH line, and being above 70ºC, but sometimes the election could look arbitrary. The mass flow that should be in the compressor catalogue is estimated to reach the discharge temperature set at RTselect software (normally 110ºC). The reason is that is not known how EES model the properties of the oil, and is preferable trust in RTselect for this kind of issues at the modelling.

Moving to the pinch analysis, there were a lot of difficulties founding an optimal COP for the majority of the configurations. The problem was that EES does not work perfectly when the optimization is carried with the pinch method, because sometimes the same operation throws different results and the guess values have to be taken in account at any moment.

Moreover, it was noticed that the maximizing function of the COP shows an theoretically optimal value, but afterwards a better value of the COP could be found in some models just trying new values for the intermediate temperatures, for this reason is considered that the Ammonia model for the two-stage does not perform completely.

Other trouble was to start an optimization for the two-stage model with the six key temperatures and the intermediate temperatures at the same time, 8 variables in total, being the result obtained far from the optimal COP. In addition, some problems have been found when the guess value of the key temperatures is set in 3K, making the optimization not work properly. The pinch methodology applied to Heat Pumps is a field poorly developed and needs further investigation in future.

Furthermore, the temperature difference at the Heat Exchanger Network is took as valid even is not above 3K, considering that the mismatch explained before and the implementation to the cycle to a real solution could show an error below 0.1K.

On the other hand, the evaluation performed in this project is completely focus on the efficiency of the system, which means that for future development will be interesting to run the model with real data of the District Heating demand, looking at the behaviour of the facility working at different loads and control of its operation, or making an techno-economical evaluation comparing with other possibilities to cool down the datacenter.

In addition, the part of the Heat Exchanger Network could be improved, decreasing the number of exchangers and optimizing the size and the cost of them, working on a techno-economic analysis as suggested before.

Chapter 8: Conclusions

In this thesis, the integration of large HP to the datacenter cooling system has been studied. The study starts with the comparison of two different heat sources, with distinct capacities and configurations. Meanwhile is suggested the possibility to have screw compressors with Oil Cooling instead of the piston option. The results of the modelling for all the cases shows that the best option in terms of performance and logistic is the water source in addition to screw compressors with OC, arriving to a COP of 3.93 against the 3.67 of the Air source with piston compressor.

In order to increase the efficiency of the system, other configurations have been purposed for the one stage scenario, and the overall COP of the HPs in increased until 4.17. The pinch method is applied to reach a better performance, until 4.29, but it was not possible to find a Heat Exchanger Network that could fit with the minimum ΔT of 3K set. A two-stage cycle is contemplate to have the best performance as possible, and the result shows an improvement of 7.06% respect to the one-stage cycle. The COP is finally raised until 4.53, with the proposal of new configurations.

The pinch analysis is applied to the two-stage Heat Pump, with the same minimum temperature difference of 3K, with a step of 100. The result for the new COP is 4.7, making it the best option in terms of performance of the report. By last a heat exchanger network with 18 exchangers is developed with a minimum temperature difference of 2.93K, stated as negligible and took as valid solution for the pinch method.

Others configurations with HFO are completed for the one and two stage models, to see which were the possibilities and differences against the Ammonia, but in the end are not considered as proper in front of the Ammonia and screw compressors for large heat pumps.

Bibliography

[1] The Danish Government, “Danish Climate Policy Plan,” 2013. [Online]. Available: https://ens.dk/sites/ens.dk/files/Analyser/danishclimatepolicyplan_uk.pdf.

[2] Ministry of Foreign Affairs, “Invest in DK,” 2018. [Online]. Available: https://investindk.com/set-up-a-business/cleantech/data-centres.

[3] G. O'Dwyer, “Computerweekly,” 2018. [Online]. Available: https://www.computerweekly.com/news/252447406/Danish-government-in-ambitious-datacentre-push.

[4] Danish Council on Climate Change, “Klimarådet,” 2017. [Online]. Available: https://www.klimaraadet.dk/en/nyheder/third-report-transition-towards-2030.

[5] Datacenter Industrien, 2018. [Online]. Available: https://datacenterindustrien.dk/data-center-map/.

[6] IDC, “IDC, analyse the future,” 2017. [Online]. Available: https://www.idc.com/getdoc.jsp?containerId=prUS42913717.

[7] J. G. Koomey, “Growth in Datacenter Electricity Use 2005 to 2010,” 2011.

[8] J. M. Lima, “Data Economy,” 2017. [Online]. Available: https://data-economy.com/mobitv-colab-with-digital-realty-to-open-new-data-centre-in-atlanta/.

[9] X. Zhang, T. Lindberg, N. Xiong, V. Vyatkin and A. Mousavi, “Cooling Energy Consumption Investigation of Data Center IT Room with Vertical Placed Server,” 2016.

[10] Instituto de Ingeniería Energética. UPV, “Refrigerantes,” Universitat Politècnica de València., Valencia, 2017.

[11] Instituto de Ingenieria Energética. UPV , “Ammonia systems,” Universitat Politècnica de València., Valencia..

[12] E. Granryd, Refrigerating Engineering, Stockholm: KTH, 2005.

[13] Instituto de Ingeniería Energética. UPV, “Components of the Vapor Cycle,” Universitat Politècnica de València., Valencia, 2017.

[14] R. Dott, “ResearchGate,” 2014. [Online]. Available: https://www.researchgate.net/figure/Heat-pump-process-in-log-p-h-diagram-with-isentropic-compressor-efficiency_fig4_269930149. [Accessed 10 01 2019].

[15] Jack, “Oil & Gas Process Engineering,” 2009. [Online]. Available: www.oilngasprocess.com/gas-production-facility/compressors/types-of-compressors/reciprocating-compressors-1.html. [Accessed 10 1 2019].

[16] W. B. Markussen, “Two stage cycles and refrigeration,” DTU, 2015.

[17] M. Wahlroos, M. Pärssinen, J. Manner and S. Syri, “Utilizing datacenter waste heat in district heating,” ScienceDirect, Aalto, 2017.

[18] M. Monroe, “DataCenter Knowledge,” 10 May 2016. [Online]. Available: https://www.datacenterknowledge.com/archives/2016/05/10/how-to-reuse-waste-heat-from-data-centers-intelligently. [Accessed 11 January 2019].

[19] Fortum, “Data Center cooling with Heat Recovery,” Stockholm, 2017.

[20] P. Judge, “Datacenterdynamics,” 8 September 2017. [Online]. Available: https://www.datacenterdynamics.com/news/facebooks-denmark-data-center-will-supply-heat-to-city/. [Accessed 13 1 2019].

[21] S. S-Å, “Technical and economical optimization of datacenter waste heat utilization,” Aalto University, Aalto, 2015.

[22] Baxtel, “Baxtel,” [Online]. Available: https://baxtel.com/data-center/apple-denmark-aabenraa-kasso. [Accessed 13 January 2019].

[23] PRIEBE, “Excess heat from the data center in Kassø. Prerequisites for company financial calculations.,” 2018.

[24] A. Zoughaib, From Pinch Methodology to Pinch Energy Integration of Flexible Systems, ISTE Press - Elsevier, 2017.

[25] I. C. Kemp, Pinch Analysis and Process Integration, Butterworth-Heinemann, 2006.

[26] B. Elmegaard, “An Algorithm for Pinch Analysis,” 2006.

[27] B. Linnhoff, “The Pinch Design Method for Heat Exchanger Networks,” *Chemical Engineerging Science,* vol. 38, pp. 745-763, 1983.

[28] P. H. Jørgensen, B. Elmegaard, W. Brix, D. Erasmus, W. B. Markussen and K. Hoffmann, "Design and optimization of the heat exchanger network for district heating ammonia heat pumps connected in series," DTU, 2018.

Appendices

Appendix A

Model of the one-stage ammonia HP

```
"Two 1-stage heat pump connected in parallel with water as source"
"Assumptions:
-Evaporators capacity fixed
-temperatures are taken from the report for heat source and district heating
-assumptions are made regarding the temperature differences at condenser and evaporator
-evaporation and condensation are supposed happening at constant pressure
-no electrical or mechanical losses are considered for the compressor
-no superheating is considered"


"!Pinch method"

Procedure thigh(T_max;T_min;DELTA_T;N:deltaT[1..N])
$Arrays On
$common
T_max;T_min;DELTA_T_int;T_forward_dh;T_return_dh;N;Q_dot_C_HP1;Q_dot_oil_HP1;T_oil1_HP1;T_oil2_HP1;T_HP1[5];T_HP1[2];T_cond2;Q_dot_sc_HP1;T_C_HP1;Q_dot_con_HP1;Q_dot_dsh_HP1;T_pinch_HP1;Q_dot_C_HP2;Q_dot_oil_HP2;T_oil1_HP2;T_oil2_HP2;T_HP2[5];T_cond3;Q_dot_sc_HP2;T_C_HP2;Q_dot_con_HP2;Q_dot_dsh_HP2;T_pinch_HP2;Q_dot_condenser


"Hot streams temperature intervals definition"
Thot[1]:=T_max
i:=1
Repeat
Tlow[i]:=Thot[i]-DELTA_T_int
Thot[i+1]:=Thot[i]-DELTA_T_INT
i:=i+1
Until Thot[i]=T_HP1[5]
thig=Thot[i]

"!Heat transfered HP1"

"Oil cooling hot stream"
i:=1
Repeat
If (Thot[i]<T_oil1_HP1) or (Thot[i]>T_HP1[2]) Then
Chot[i;1]=0
Else
m=floor((T_OIL2_HP1-T_OIL1_HP1)/(DELTA_T_INT))
Chot[i;1]=Q_dot_oil_HP1/m
C1tot=sum(Chot[1..i;1])
Endif
i=i+1
Until i=N

 If C1tot<>Q_DOT_OIL_HP1 Then   "to check that the hot stream is divided between the intervals in the right way"
i:=1
```

```
 Repeat
If (Thot[i]<T_oil1_HP1) or (Thot[i]>T_HP1[2]) Then
Chot[i;1]=0
Else
l=ceil((T_OIL2_HP1-T_OIL1_HP1)/(DELTA_T_INT))
Chot[i;1]=Q_dot_oil_HP1/l
C1tot=sum(Chot[1..i;1])
Endif
i=i+1
Until i=N
Endif

"Subcooling hot stream"
i:=1
Repeat
If (Thot[i]>T_cond2) or (Thot[i]<T_HP1[5]) Then
Chot[i;2]=0
Else
p=ceil(((T_COND2-T_HP1[5]))/(DELTA_T_INT))
Chot[i;2]=Q_dot_sc_HP1/p
C2tot=sum(Chot[1..i;2])
Endif
i=i+1
Until i=N

If C2tot<>Q_DOT_SC_HP1 Then
i:=1
 Repeat
If (Thot[i]>T_cond2) or (Thot[i]<T_HP1[5]) Then
Chot[i;2]=0
Else
k=floor(((T_COND2-T_HP1[5]))/(DELTA_T_INT))
Chot[i;2]=Q_dot_sc_HP1/k
C2tot=sum(Chot[1..i;2])
Endif
i=i+1
Until i=N
Endif


"Condenser hot stream"
i:=1
Repeat
If (Thot[i]>T_C_HP1+DELTA_T_int) or (Thot[i]<T_cond2) Then
Chot[i;3]=0
Else
Chot[i;3]=Q_dot_con_HP1
C3tot=sum(Chot[1..i;3])
Endif
i=i+1
Until i=N

If C3tot<>Q_DOT_CON_HP1 Then   "to check that the hot stream is divided between the
intervals in the right way"
i:=1
Repeat
If (Thot[i]>T_C_HP1+DELTA_T_int) or (Thot[i]<T_cond2) Then
Chot[i;3]=0
Else
Chot[i;3]=Q_dot_con_HP1/2
```

```
C3tot=sum(Chot[1..i;3])
Endif
i=i+1
Until i=N
Endif


"Desuperheating hot stream"
i:=1
Repeat
If (Thot[i]<T_C_HP1+DELTA_T_int)   Then
Chot[i;4]=0
Else
d=ceil((T_max-T_C_HP1)/(DELTA_T_INT))
Chot[i;4]=Q_dot_dsh_HP1/d
C4tot=sum(Chot[1..i;4])
Endif
i=i+1
Until i=N

 If C4tot<>Q_DOT_DSH_HP1 Then
i:=1
Repeat
If (Thot[i]<T_C_HP1+DELTA_T_int)   Then
Chot[i;4]=0
Else
q=floor((T_max-T_C_HP1)/(DELTA_T_INT))
Chot[i;4]=Q_dot_dsh_HP1/q
C4tot=sum(Chot[1..i;4])
Endif
i=i+1
Until i=N
Endif

"!Heat transfered HP2"

"Oil cooling hot stream"
i:=1
Repeat
If Thot[i]<T_oil1_HP2 Then
Chot[i;5]=0
Else
h=floor((T_OIL2_HP2-T_OIL1_HP2)/(DELTA_T_INT))
Chot[i;5]=Q_dot_oil_HP2/h
C5tot=sum(Chot[1..i;5])
Endif
i=i+1
Until i=N

 If C5tot<>Q_DOT_OIL_HP2 Then   "to check that the hot stream is divided between the
intervals in the right way"
i:=1
 Repeat
If Thot[i]<T_oil1_HP2 Then
Chot[i;5]=0
Else
w=ceil((T_OIL2_HP2-T_OIL1_HP2)/(DELTA_T_INT))
Chot[i;5]=Q_dot_oil_HP2/w
C5tot=sum(Chot[1..i;5])
Endif
```

```
i=i+1
Until i=N
Endif

"Subcooling hot stream"
i:=1
Repeat
If (Thot[i]>T_cond3) or (Thot[i]<T_HP2[5]) Then
Chot[i;6]=0
Else
y=ceil(((T_COND3-T_HP2[5]))/(DELTA_T_INT))
Chot[i;6]=Q_dot_sc_HP2/y
C6tot=sum(Chot[1..i;6])
Endif
i=i+1
Until i=N

If C6tot<>Q_DOT_SC_HP2 Then
i:=1
 Repeat
If (Thot[i]>T_cond3) or (Thot[i]<T_HP2[5]) Then
Chot[i;6]=0
Else
t=floor(((T_COND3-T_HP2[5]))/(DELTA_T_INT))
Chot[i;6]=Q_dot_sc_HP2/t
C6tot=sum(Chot[1..i;6])
Endif
i=i+1
Until i=N
Endif


"Condenser hot stream"
i:=1
Repeat
If (Thot[i]>T_C_HP2+DELTA_T_int) or (Thot[i]<T_cond3) Then
Chot[i;7]=0
Else
Chot[i;7]=Q_dot_con_HP2
C7tot=sum(Chot[1..i;7])
Endif
i=i+1
Until i=N

If C7tot<>Q_DOT_CON_HP2 Then   "to check that the hot stream is divided between the
intervals in the right way"
i:=1
Repeat
If (Thot[i]>T_C_HP2+DELTA_T_int) or (Thot[i]<T_cond3) Then
Chot[i;7]=0
Else
Chot[i;7]=Q_dot_con_HP2/2
C7tot=sum(Chot[1..i;7])
Endif
i=i+1
Until i=N
Endif


"Desuperheating hot stream"
```

```
i:=1
Repeat
If (Thot[i]<T_C_HP2+DELTA_T_int)   Then
Chot[i;8]=0
Else
c=ceil((T_max-T_C_HP2)/(DELTA_T_INT))
Chot[i;8]=Q_dot_dsh_HP2/c
C8tot=sum(chot[1..i;8])
Endif
i=i+1
Until i=N

 If C8tot<>Q_DOT_DSH_HP2 Then
i:=1
Repeat
If (Thot[i]<T_C_HP2+DELTA_T_int)   Then
Chot[i;8]=0
Else
f=floor((T_max-T_C_HP2)/(DELTA_T_INT))
Chot[i;8]=Q_dot_dsh_HP2/f
C8tot=sum(Chot[1..i;8])
Endif
i=i+1
Until i=N
Endif


"Total hot stream for every temperature interval"
i:=1
Repeat
CThot[i]=Chot[i;1]+Chot[i;2]+Chot[i;3]+Chot[i;4]+Chot[i;5]+Chot[i;6]+Chot[i;7]+Chot[i;8]
i=i+1
Until i=N


i:=1
Repeat
Qsum=sum(CThot[1..i])
i=i+1
Until i=N


"Sum of hot streams for plotting"
 i:=1
j:=1
Repeat
If  i=1 Then
Qheat[i]=Q_dot_condenser
Else
Qheat[i]=Q_dot_condenser-sum(CThot[1..j])
Endif
i=i+1
j=i-1
Until i=N+1


"Cold stream temperature intervals definition"
i:=1
Repeat
Tcold[i]=40*Qheat[i]/Q_dot_condenser +T_return_dh
```

```
i:=i+1
Until i=N+1


"Temperature difference between hot streams and cold one"
 Duplicate j=1;N
deltaT[j]=Thot[j]-Tcold[j]
End


"minimum temperature differeNce that i can't insert properly
DELTA_T_min=min(deltaT[1..N])
DELTA_T_min=3"

End


T_cond2=T_C_HP1-DELTA_T_int
T_cond3=T_C_HP2-DELTA_T_int
T_max=T_HP2[2]
T_min=T_return_dh

DELTA_T_int= (T_max-T_HP1[5])/(N-1)
N=100


Call thigh(T_max;T_min;DELTA_T_int;N:deltaT[1..N])
"Call qdh1(N;Thigh[1..N];Tlow[1..N]; Q_dot_C_HP1;T_forward_dh;T_return_dh:Qcold1)"


DELTAT_min= 3 [K]

DELTAT_pinch=min(deltaT[1..N-1])



"!Initial considerations and inputs"

DELTAT_evaporator=5
{DELTAT_condenser_HP1=5
DELTAT_condenser_HP2=5
DELTAT_subcooling_HP1=5
DELTAT_subcooling_HP2=5}
R$='ammonia'
Q_dot_datacenter=30 [MW]
Q_dot_evaporator=0,9*Q_dot_datacenter

P_dh= 9 [bar]                                  "Assumption of the pressure in the
district heating circuit"
P_datacenter=10 [bar]                          "Assumption of the pressure in the
datacenter circuit"

"Assumption of the temperature in the inlet of the oil cooling, founded in the catalogue"
T_oil1_HP1=75 [C]
T_oil1_HP2=75 [C]

"Assumption oil cycle and compressor"
eta_is_HP1=0,76
eta_is_HP2=0,796
```

```
"Temperature of the water in the datacenter circuit"
T_return_data=32 [C]
T_mid_data=28 [C]
T_forward_data=20 [C]

"!Source calculations"

"district heating"
T_return_dh=45 [C]
"T_mid_dh=70 [C]"
T_forward_dh=85 [C]
h_return_dh=enthalpy(Water;P=P_dh;T=T_return_dh)
h_mid_dh=enthalpy(Water;P=P_dh;T=T_mid_dh)
h_forward_dh=enthalpy(Water;P=P_dh;T=T_forward_dh)
Q_dot_C_HP1=Q_dot_sc_HP1+Q_dot_con_HP1+Q_dot_dsh_HP1
Q_dot_C_HP2=Q_dot_sc_HP2+Q_dot_con_HP2+Q_dot_dsh_HP2


"datacenter cooling"
h_forward_data=enthalpy(Water;P=P_datacenter;T=T_forward_data)
h_mid_data=enthalpy(Water;P=P_datacenter;T=T_mid_data)
h_return_data=enthalpy(Water;P=P_datacenter;T=T_return_data)
Q_dot_E_HP1=m_dot_data*(-h_forward_data+h_mid_data)/1000
Q_dot_E_HP2=m_dot_data*(-h_mid_data+h_return_data)/1000
Q_dot_condenser=Q_dot_C_HP1+Q_dot_C_HP2+Q_dot_oil_HP1+Q_dot_oil_HP2
Q_dot_evaporator=Q_dot_E_HP1+Q_dot_E_HP2

"Temperatures fixed by deltaT pinch"
{T_C_HP1=70
T_C_HP2=80}
T_E_HP1=T_forward_data-DELTAT_evaporator
T_E_HP2=T_mid_data-DELTAT_evaporator

"Condensing and evaporating pressures"
P_C_HP1=pressure(R$;T=T_C_HP1;x=0)
P_E_HP1=pressure(R$;T=T_E_HP1;x=1)
P_C_HP2=pressure(R$;T=T_C_HP2;x=0)
P_E_HP2=pressure(R$;T=T_E_HP2;x=1)


"!HP1 cycle calculations"

"Evaporator exit"
T_HP1[1]=T_E_HP1
P_HP1[1]=P_E_HP1
h_HP1[1]=enthalpy(R$;P=P_E_HP1;x=1)
s_HP1[1]=entropy(R$;T=T_HP1[1];h=h_HP1[1])
Q_dot_E_HP1=m_dot_ref_HP1*(h_HP1[1]-h_HP1[6])/1000

"After compressor"
P_HP1[2]=P_C_HP1
s_is_HP1=s_HP1[1]
h_is_HP1=enthalpy(R$;P=P_C_HP1;s=s_is_HP1)
eta_is_HP1=(h_is_HP1-h_HP1[1])/(h_HP1_2W-h_HP1[1])
T_HP1_2W=temperature(R$;P=P_C_HP1;h=h_HP1_2W)
W_dot_HP1*1000=m_dot_ref_HP1*(h_HP1_2W-
h_HP1[1])

"Oil cooling"
T_HP1[2]=T_oil2_HP1
```

```
v_dot_oil_HP1=2500*convert(l/min;m3/s)
rho_oil_HP1=density(Engine_Oil_10W; T=(T_oil1_HP1+T_oil2_HP1)/2)
m_dot_oil_HP1=v_dot_oil_HP1*rho_oil_HP1
h_oc1_dh=enthalpy(Water;P=P_dh;T=T_oc1_dh)
Q_dot_oil_HP1=m_dot_dh*(h_oc1_dh-h_mid_dh)/1000
-Q_dot_oil_HP1=m_dot_ref_HP1*(h_HP1[2]-h_HP1_2W)/1000
T_HP1[2]=temperature(R$;P=P_C_HP1;h=h_HP1[2])
cp_oil_HP1=cp(Engine_Oil_10W; T=(T_oil1_HP1+T_oil2_HP1)/2)
Q_dot_oil_HP1=m_dot_oil_HP1*cp_oil_HP1*(T_oil2_HP1-T_oil1_HP1)/1000


"After desuperheater"
P_HP1[3]=P_C_HP1
h_HP1[3]=enthalpy(R$;P=P_C_HP1;x=1)
T_HP1[3]=t_sat(R$;P=P_C_HP1)
Q_dot_dsh_HP1=m_dot_dh*(h_mid_dh-h_con1_out_dh)/1000
T_con1_out_dh=temperature(Water;P=P_dh;h=h_con1_out_dh)
Q_dot_dsh_HP1=m_dot_ref_HP1*(h_HP1[2]-h_HP1[3])/1000

"After condenser"
P_HP1[4]=P_C_HP1
T_HP1[4]=T_C_HP1
h_HP1[4]=enthalpy(R$;P=P_C_HP1;x=0)
Q_dot_con_HP1=m_dot_ref_HP1*(h_HP1[3]-h_HP1[4])/1000
Q_dot_con_HP1=m_dot_dh*(h_con1_out_dh-h_con1_in_dh)/1000
T_con1_in_dh=temperature(Water;P=P_dh;h=h_con1_in_dh)


"After subcooler"
T_HP1[5]=T_return_dh+DELTAT_subcooling_HP1
P_HP1[5]=P_C_HP1
h_HP1[5]=enthalpy(R$;P=P_C_HP1;T=T_HP1[5])
Q_dot_sc_HP1=m_dot_ref_HP1*(h_HP1[4]-h_HP1[5])/1000
Q_dot_sc_HP1=m_dot_dh*(h_con1_in_dh-h_return_dh)/1000


"Pinch and fix condensing temperature"
Q_dot_pinch_HP1=m_dot_ref_HP1*(enthalpy(R$;P=P_C_HP1;x=1)-
enthalpy(R$;x=0;P=P_C_HP1))/1000
Q_dot_pinch_HP1=m_dot_dh*(h_pinch_HP1-h_con1_in_dh)/1000
T_pinch_HP1=temperature(Water;P=P_dh;h=h_pinch_HP1)
T_C_HP1=T_pinch_HP1+DELTAT_condenser_HP1

"After expansion valve"
P_HP1[6]=P_E_HP1
h_HP1[6]=h_HP1[5]
T_HP1[6]=temperature(R$;P=P_E_HP1;h=h_HP1[6])
x_HP1[6]=quality(R$;h=h_HP1[6];P=P_E_HP1)


"!HP2 cycle calculations"

"After evaporator"
T_HP2[1]=T_E_HP2
P_HP2[1]=P_E_HP2
h_HP2[1]=enthalpy(R$;T=T_HP2[1];x=1)
s_HP2[1]=entropy(R$;T=T_HP2[1];h=h_HP2[1])
Q_dot_E_HP2=m_dot_ref_HP2*(h_HP2[1]-h_HP2[6])/1000

"After compressor"
```

```
P_HP2[2]=P_C_HP2
s_is_HP2=s_HP2[1]
h_is_HP2=enthalpy(R$;P=P_C_HP2;s=s_is_HP2)
eta_is_HP2=(h_is_HP2-h_HP2[1])/(h_HP2_2W-h_HP2[1])
T_HP2_2W=temperature(R$;P=P_C_HP2;h=h_HP2_2W)
W_dot_HP2*1000=m_dot_ref_HP2*(h_HP2_2W-
h_HP2[1])


"Oil cooling"
T_HP2[2]=T_oil2_HP2
v_dot_oil_HP2=2400*convert(l/min;m3/s)
rho_oil_HP2=density(Engine_Oil_10W; T=(T_oil1_HP2+T_oil2_HP2)/2)
m_dot_oil_HP2=v_dot_oil_HP2*rho_oil_HP2
h_oc2_dh=enthalpy(Water;P=P_dh;T=T_oc2_dh)
Q_dot_oil_HP2=m_dot_dh*(h_oc2_dh-h_oc1_dh)/1000
-Q_dot_oil_HP2=m_dot_ref_HP2*(h_HP2[2]-h_HP2_2W)/1000
T_HP2[2]=temperature(R$;P=P_C_HP2;h=h_HP2[2])
cp_oil_HP2=cp(Engine_Oil_10W; T=(T_oil1_HP2+T_oil2_HP2)/2)
Q_dot_oil_HP2=m_dot_oil_HP2*cp_oil_HP2*(T_oil2_HP2-T_oil1_HP2)/1000


"After desuperheater"
P_HP2[3]=P_C_HP2
h_HP2[3]=enthalpy(R$;P=P_C_HP2;x=1)
T_HP2[3]=t_sat(R$;P=P_C_HP2)
Q_dot_dsh_HP2=m_dot_dh*(h_forward_dh-h_con2_out_dh)/1000
T_con2_out_dh=temperature(Water;P=P_dh;h=h_con2_out_dh)
Q_dot_dsh_HP2=m_dot_ref_HP2*(h_HP2[2]-h_HP2[3])/1000

"After condenser"
P_HP2[4]=P_C_HP2
T_HP2[4]=T_C_HP2
h_HP2[4]=enthalpy(R$;P=P_C_HP2;x=0)
Q_dot_con_HP2=m_dot_ref_HP2*(h_HP2[3]-h_HP2[4])/1000
Q_dot_con_HP2=m_dot_dh*(h_con2_out_dh-h_con2_in_dh)/1000
T_con2_in_dh=temperature(Water;P=P_dh;h=h_con2_in_dh)


"After subcooler"
T_HP2[5]=T_oc2_dh+DELTAT_subcooling_HP2
P_HP2[5]=P_C_HP2
h_HP2[5]=enthalpy(R$;P=P_C_HP2;T=T_HP2[5])
Q_dot_sc_HP2=m_dot_ref_HP2*(h_HP2[4]-h_HP2[5])/1000
Q_dot_sc_HP2=m_dot_dh*(h_con2_in_dh-h_oc2_dh)/1000


"Pinch and fix condensing temperature"
Q_dot_pinch_HP2=m_dot_ref_HP2*(enthalpy(R$;P=P_C_HP2;x=1)-
enthalpy(R$;x=0;P=P_C_HP2))/1000
Q_dot_pinch_HP2=m_dot_dh*(h_pinch_HP2-h_con2_in_dh)/1000
T_pinch_HP2=temperature(Water;P=P_dh;h=h_pinch_HP2)
T_C_HP2=T_pinch_HP2+DELTAT_condenser_HP2

"After expansion valve"
P_HP2[6]=P_E_HP2
h_HP2[6]=h_HP2[5]
T_HP2[6]=temperature(R$;P=P_E_HP2;h=h_HP2[6])
x_HP2[6]=quality(R$;h=h_HP2[6];P=P_E_HP2)
```

```
"performance"
COP_HP1=(Q_dot_C_HP1+Q_dot_oil_HP1)/W_dot_HP1
COP_HP2=(Q_dot_oil_HP2+Q_dot_C_HP2)/W_dot_HP2

CARNOT_COP_HP1=(T_C_HP1+273,15)/(T_C_HP1-T_E_HP1)
CARNOT_COP_HP2=(T_C_HP2+273,15)/(T_C_HP2-T_E_HP2)

COP_total=(Q_dot_condenser)/(W_dot_HP1+W_dot_HP2)


"A way to check if the overall energy balance is correct"
EB = Q_dot_oil_HP1 + Q_dot_oil_HP2 + Q_dot_C_HP2 + Q_dot_C_HP1 - (Q_dot_E_HP2 +
Q_dot_E_HP1 + W_dot_HP2  + W_dot_HP1) "EB should be zero"

"!plot script"


"T-Q condenser"
T_districtheating[1]=T_return_dh
T_districtheating[2]=T_con1_in_dh
T_districtheating[3]=T_con1_out_dh
T_districtheating[4]=T_mid_dh
T_districtheating[5]=T_oc1_dh
T_districtheating[6]=T_oc2_dh
T_districtheating[7]=T_con2_in_dh
T_districtheating[8]=T_con2_out_dh
T_districtheating[9]=T_forward_dh
Q_dot_dh[1]=0
Q_dot_dh[2]=Q_dot_sc_HP1
Q_dot_dh[3]=Q_dot_dh[2]+Q_dot_con_HP1
Q_dot_dh[4]=Q_dot_dh[3]+Q_dot_dsh_HP1
Q_dot_dh[5]=Q_dot_dh[4]+Q_dot_oil_HP1
Q_dot_dh[6]=Q_dot_dh[5]+Q_dot_oil_HP2
Q_dot_dh[7]=Q_dot_dh[6]+Q_dot_sc_HP2
Q_dot_dh[8]=Q_dot_dh[7]+Q_dot_con_HP2
Q_dot_dh[9]=Q_dot_condenser
T_oil_HP1[4]=T_oil1_HP1
T_oil_HP1[5]=T_oil2_HP1
T_sc_HP1[1]=T_HP1[5]
T_sc_HP1[2]=T_HP1[4]
T_con_HP1[2]=T_C_HP1
T_con_HP1[3]=T_C_HP1
T_dsh_HP1[3]=T_HP1[3]
T_dsh_HP1[4]=T_HP1[2]
T_oil_HP2[5]=T_oil1_HP2
T_oil_HP2[6]=T_oil2_HP2
T_sc_HP2[6]=T_HP2[5]
T_sc_HP2[7]=T_HP2[4]
T_con_HP2[7]=T_C_HP2
T_con_HP2[8]=T_C_HP2
T_dsh_HP2[8]=T_HP2[3]
T_dsh_HP2[9]=T_HP2[2]

"T-Q evaporator"
T_data[3]=T_return_data
T_data[2]=T_mid_data
T_data[1]=T_forward_data
Q_dot_data[1]=0
Q_dot_data[2]=Q_dot_E_HP1
```

```
Q_dot_data[3]=Q_dot_evaporator
T_evap_HP1[1]=T_E_HP1
T_evap_HP1[2]=T_E_HP1
T_evap_HP2[3]=T_E_HP2
T_evap_HP2[2]=T_E_HP2

"DH diagram"
h_dh[1]=h_return_dh
h_dh[2]=h_mid_dh
h_dh[3]=h_forward_dh
p_dh[1]=P_dh
p_dh[2]=P_dh
p_dh[3]=P_dh

"datacenter diagram"
h_data[3]=h_return_data
h_data[2]=h_mid_data
h_data[1]=h_forward_data
p_data[1]=P_datacenter
p_data[2]=P_datacenter
p_data[3]=P_datacenter
```

Appendix B

Model of the ammonia two-stage HP

```
"Two 2-stage heat pump connected in parallel with water as source"
"Assumptions:
-Evaporators capacity fixed
-temperatures are taken from the report for heat source and district heating
-assumptions are made regarding the temperature differences at condenser and evaporator
-evaporation and condensation are supposed happening at constant pressure
-no electrical or mechanical losses are considered for the compressor
-no superheating is considered"


"!Pinch method"
 $defaultarraysize 100
Procedure thigh(T_max;T_min;DELTA_T;N:deltaT[1..N])
$Arrays On
$common
T_max;T_min;DELTA_T_int;T_forward_dh;T_return_dh;N;Q_dot_C_HP1;Q_dot_oil_L_HP1;
Q_dot_oil_H_HP1;T_oil1_L_HP1;T_oil1_H_HP1;T_oil2_L_HP1;T_oil2_H_HP1;T_HP1[8];T_
HP1[5];T_HP1[2];T_HP1[3];T_cond2;Q_dot_sc_HP1;T_C_H_HP1;T_C_L_HP1;Q_dot_con_
HP1;Q_dot_dsh_H_HP1;Q_dot_dsh_L_HP1;T_pinch_H_HP1;Q_dot_C_HP2;Q_dot_oil_L_H
P2;Q_dot_oil_H_HP2;T_oil1_L_HP2;T_oil1_H_HP2;T_oil2_L_HP2;T_oil2_H_HP2;T_HP2[8]
;T_HP2[5];T_HP2[2];T_HP2[3];T_cond3;Q_dot_sc_HP2;T_C_H_HP2;T_C_L_HP2;Q_dot_co
n_HP2;Q_dot_dsh_H_HP2;Q_dot_dsh_L_HP2;T_pinch_H_HP2;Q_dot_condenser


"Hot streams temperature intervals definition"
Thot[1]:=T_max
i:=1
Repeat
Tlow[i]:=Thot[i]-DELTA_T_int
Thot[i+1]:=Thot[i]-DELTA_T_INT
i:=i+1
Until Thot[i]=T_HP1[8]
thig=Thot[i]
```

```
"!Heat transfered HP1"

"Oil cooling hot stream low stage"
i:=1
Repeat
If (Thot[i]<T_oil1_L_HP1) or (Thot[i]>T_HP1[2]) Then
Chot[i;1]=0
Else
m=floor((T_OIL2_L_HP1-T_OIL1_L_HP1)/(DELTA_T_INT))
Chot[i;1]=Q_dot_oil_L_HP1/m
C1tot=sum(Chot[1..i;1])
Endif
i=i+1
Until i=N

 If C1tot<>Q_DOT_OIL_L_HP1 Then   "to check that the hot stream is divided between the
intervals in the right way"
i:=1
 Repeat
If (Thot[i]<T_oil1_L_HP1) or (Thot[i]>T_HP1[2]) Then
Chot[i;1]=0
Else
l=ceil((T_OIL2_L_HP1-T_OIL1_L_HP1)/(DELTA_T_INT))
Chot[i;1]=Q_dot_oil_L_HP1/l
C1tot=sum(Chot[1..i;1])
Endif
i=i+1
Until i=N
Endif

"Oil cooling hot stream high stage"
i:=1
Repeat
If (Thot[i]<T_oil1_H_HP1) or (Thot[i]>T_HP1[5]) Then
Chot[i;9]=0

Else
r=floor((T_OIL2_H_HP1-T_OIL1_H_HP1)/(DELTA_T_INT))
Chot[i;9]=Q_dot_oil_H_HP1/r
C9tot=sum(Chot[1..i;9])
Endif
i=i+1
Until i=N

 If C9tot<>Q_DOT_OIL_H_HP1 Then   "to check that the hot stream is divided between the
intervals in the right way"
i:=1
 Repeat
If (Thot[i]<T_oil1_H_HP1) or (Thot[i]>T_HP1[5]) Then
Chot[i;9]=0
Else
z=ceil((T_OIL2_H_HP1-T_OIL1_H_HP1)/(DELTA_T_INT))
Chot[i;9]=Q_dot_oil_H_HP1/z
C9tot=sum(Chot[1..i;9])
Endif
i=i+1
Until i=N
Endif
```

```
"desuperheater of the lower stage"
i:=1
Repeat
If (Thot[i]<T_HP1[3]) or (Thot[i]>T_HP1[2])   Then
Chot[i;11]=0
Else
g=ceil((T_HP1[2]-T_HP1[3])/(DELTA_T_INT))
Chot[i;11]=Q_dot_dsh_L_HP1/g
C11tot=sum(Chot[1..i;11])
Endif
i=i+1
Until i=N

If C11tot<>Q_DOT_DSH_L_HP1 Then
i:=1
Repeat
If (Thot[i]<T_HP1[3]) or (Thot[i]>T_HP1[2])  Then
Chot[i;11]=0
Else
gg=floor((T_HP1[2]-T_HP1[3])/(DELTA_T_INT))
Chot[i;11]=Q_dot_dsh_L_HP1/gg
C11tot=sum(Chot[1..i;11])
Endif
i=i+1
Until i=N
Endif

"Subcooling hot stream"
i:=1
Repeat
If (Thot[i]>T_cond2) or (Thot[i]<T_HP1[8]) Then
Chot[i;2]=0
Else
p=ceil(((T_COND2-T_HP1[8]))/(DELTA_T_INT))
Chot[i;2]=Q_dot_sc_HP1/p
C2tot=sum(Chot[1..i;2])
Endif
i=i+1
Until i=N

If C2tot<>Q_DOT_SC_HP1 Then
i:=1
 Repeat
If (Thot[i]>T_cond2) or (Thot[i]<T_HP1[8]) Then
Chot[i;2]=0
Else
k=floor(((T_COND2-T_HP1[8]))/(DELTA_T_INT))
Chot[i;2]=Q_dot_sc_HP1/k
C2tot=sum(Chot[1..i;2])
Endif
i=i+1
Until i=N
Endif

"Condenser hot stream"
i:=1
Repeat
If (Thot[i]>T_C_H_HP1+DELTA_T_int) or (Thot[i]<T_cond2) Then
Chot[i;3]=0
Else
```

```
Chot[i;3]=Q_dot_con_HP1
C3tot=sum(Chot[1..i;3])
Endif
i=i+1
Until i=N

If C3tot<>Q_DOT_CON_HP1 Then   "to check that the hot stream is divided between the intervals in the right way"
i:=1
Repeat
If (Thot[i]>T_C_H_HP1+DELTA_T_int) or (Thot[i]<T_cond2) Then
Chot[i;3]=0
Else
Chot[i;3]=Q_dot_con_HP1/2
C3tot=sum(Chot[1..i;3])
Endif
i=i+1
Until i=N
Endif


"Desuperheating hot stream high stage"
i:=1
Repeat
If (Thot[i]<T_C_H_HP1+DELTA_T_int) or (Thot[i]>T_HP1[5])   Then
Chot[i;4]=0
Else
d=ceil((T_HP1[5]-T_C_H_HP1)/(DELTA_T_INT))
Chot[i;4]=Q_dot_dsh_H_HP1/d
C4tot=sum(Chot[1..i;4])
Endif
i=i+1
Until i=N

 If C4tot<>Q_DOT_DSH_H_HP1 Then
i:=1
Repeat
If (Thot[i]<T_C_H_HP1+DELTA_T_int) or (Thot[i]>T_HP1[5])  Then
Chot[i;4]=0
Else
q=floor((T_HP1[5]-T_C_H_HP1)/(DELTA_T_INT))
Chot[i;4]=Q_dot_dsh_H_HP1/q
C4tot=sum(Chot[1..i;4])
Endif
i=i+1
Until i=N
Endif

"!Heat transfered HP2"

"Oil cooling hot stream low stage"
i:=1
Repeat
If (Thot[i]<T_oil1_L_HP2) or (Thot[i]>T_HP2[2]) Then
Chot[i;5]=0
Else
h=floor((T_OIL2_L_HP2-T_OIL1_L_HP2)/(DELTA_T_INT))
Chot[i;5]=Q_dot_oil_L_HP2/h
C5tot=sum(Chot[1..i;5])
Endif
```

```
i=i+1
Until i=N

 If C5tot<>Q_DOT_OIL_L_HP2 Then   "to check that the hot stream is divided between the
intervals in the right way"
i:=1
 Repeat
If (Thot[i]<T_oil1_L_HP2) or (Thot[i]>T_HP2[2]) Then
Chot[i;5]=0
Else
w=ceil((T_OIL2_L_HP2-T_OIL1_L_HP2)/(DELTA_T_INT))
Chot[i;5]=Q_dot_oil_L_HP2/w
C5tot=sum(Chot[1..i;5])
Endif
i=i+1
Until i=N
Endif

"desuperheater of the lower stage"
i:=1
Repeat
If (Thot[i]<T_HP2[3]) or (Thot[i]>T_HP2[2])   Then
Chot[i;12]=0
Else
v=ceil((T_HP2[2]-T_HP2[3])/(DELTA_T_INT))
Chot[i;12]=Q_dot_dsh_L_HP2/v
C12tot=sum(Chot[1..i;12])
Endif
i=i+1
Until i=N

 If C12tot<>Q_DOT_DSH_L_HP2 Then
i:=1
Repeat
If (Thot[i]<T_HP2[3]) or (Thot[i]>T_HP2[2])  Then
Chot[i;12]=0
Else
vv=floor((T_HP2[2]-T_HP2[3])/(DELTA_T_INT))
Chot[i;12]=Q_dot_dsh_L_HP2/vv
C12tot=sum(Chot[1..i;12])
Endif
i=i+1
Until i=N
Endif


"Subcooling hot stream"
i:=1
Repeat
If (Thot[i]>T_cond3) or (Thot[i]<T_HP2[8]) Then
Chot[i;6]=0
Else
y=ceil(((T_COND3-T_HP2[8]))/(DELTA_T_INT))
Chot[i;6]=Q_dot_sc_HP2/y
C6tot=sum(Chot[1..i;6])
Endif
i=i+1
Until i=N

If C6tot<>Q_DOT_SC_HP2 Then
```

```
i:=1
 Repeat
If (Thot[i]>T_cond3) or (Thot[i]<T_HP2[8]) Then
Chot[i;6]=0
Else
t=floor(((T_COND3-T_HP2[8]))/(DELTA_T_INT))
Chot[i;6]=Q_dot_sc_HP2/t
C6tot=sum(Chot[1..i;6])
Endif
i=i+1
Until i=N
Endif


"Condenser hot stream"
i:=1
Repeat
If (Thot[i]>T_C_H_HP2+DELTA_T_int) or (Thot[i]<T_cond3) Then
Chot[i;7]=0
Else
Chot[i;7]=Q_dot_con_HP2
C7tot=sum(Chot[1..i;7])
Endif
i=i+1
Until i=N

If C7tot<>Q_DOT_CON_HP2 Then   "to check that the hot stream is divided between the
intervals in the right way"
i:=1
Repeat
If (Thot[i]>T_C_H_HP2+DELTA_T_int) or (Thot[i]<T_cond3) Then
Chot[i;7]=0
Else
Chot[i;7]=Q_dot_con_HP2/2
C7tot=sum(Chot[1..i;7])
Endif
i=i+1
Until i=N
Endif


"Desuperheating hot stream high stage"
i:=1
Repeat
If (Thot[i]<T_C_H_HP2+DELTA_T_int)   Then
Chot[i;8]=0
Else
c=ceil((T_max-T_C_H_HP2)/(DELTA_T_INT))
Chot[i;8]=Q_dot_dsh_H_HP2/c
C8tot=sum(chot[1..i;8])
Endif
i=i+1
Until i=N

 If C8tot<>Q_DOT_DSH_H_HP2 Then
i:=1
Repeat
If (Thot[i]<T_C_H_HP2+DELTA_T_int)   Then
Chot[i;8]=0
Else
```

```
f=floor((T_max-T_C_H_HP2)/(DELTA_T_INT))
Chot[i;8]=Q_dot_dsh_H_HP2/f
C8tot=sum(Chot[1..i;8])
Endif
i=i+1
Until i=N
Endif

"Oil cooling hot stream high stage"
i:=1
Repeat
If (Thot[i]<T_oil1_H_HP2) or (Thot[i]>T_HP2[5]) Then
Chot[i;10]=0
Else
b=floor((T_OIL2_H_HP2-T_OIL1_H_HP2)/(DELTA_T_INT))
Chot[i;10]=Q_dot_oil_H_HP2/b
C10tot=sum(Chot[1..i;10])
Endif
i=i+1
Until i=N

 If C10tot<>Q_DOT_OIL_H_HP2 Then   "to check that the hot stream is divided between the
intervals in the right way"
i:=1
Repeat
If (Thot[i]<T_oil1_H_HP2) or (Thot[i]>T_HP2[5]) Then
Chot[i;10]=0
Else
s=ceil((T_OIL2_H_HP2-T_OIL1_H_HP2)/(DELTA_T_INT))
Chot[i;10]=Q_dot_oil_H_HP2/s
C10tot=sum(Chot[1..i;10])
Endif
i=i+1
Until i=N
Endif

"Total hot stream for every temperature interval"
i:=1
Repeat
CThot[i]=Chot[i;1]+Chot[i;2]+Chot[i;3]+Chot[i;4]+Chot[i;5]+Chot[i;6]+Chot[i;7]+Chot[i;8]+Chot[
i;9]+Chot[i;10]+Chot[i;11]+Chot[i;12]
i=i+1
Until i=N
i:=1

Repeat
Qsum=sum(CThot[1..i])
i=i+1
Until i=N


"Sum of hot streams for plotting"
i:=1
j:=1
Repeat
If  i=1 Then
Qheat[i]=Q_dot_condenser
Else
Qheat[i]=Q_dot_condenser-sum(CThot[1..j])
Endif
```

```
i=i+1
j=i-1
Until i=N+1


"Cold stream temperature intervals definition"
i:=1
Repeat
Tcold[i]=40*Qheat[i]/Q_dot_condenser +T_return_dh
i:=i+1
Until i=N+1


"Temperature difference between hot streams and cold one"
Duplicate j=1;N
deltaT[j]=Thot[j]-Tcold[j]
End


End


T_coNd2=T_C_H_HP1-DELTA_T_int
T_cond3=T_C_H_HP2-DELTA_T_int
T_max=T_HP2[5]
T_min=T_return_dh

DELTA_T_int = (T_max-T_HP1[8])/(N-1)
N=100


Call thigh(T_max;T_min;DELTA_T_int;N:deltaT[1..N])
"Call qdh1(N;Thigh[1..N];Tlow[1..N]; Q_dot_C_HP1;T_forward_dh;T_return_dh:Qcold1)"


DELTAT_min= 3 [K]

DELTAT_pinch=min(deltaT[1..N-1])


"!Initial considerations and inputs"

DELTAT_evaporator=5
{DELTAT_condenser_HP1=5
DELTAT_condenser_HP2=5
DELTAT_subcooling_HP1=5
DELTAT_subcooling_HP2=5
DELTAT_dsh_L_HP1=5
DELTAT_dsh_L_HP2=5}
R$='ammonia'
Q_dot_datacenter=30 [MW]
Q_dot_evaporator=0,9*Q_dot_datacenter

P_dh= 9 [bar]                              "Assumption of the pressure in the
district heating circuit"
P_datacenter=10 [bar]                      "Assumption of the pressure in the
datacenter circuit"

"Assumption of the temperature in the inlet of the oil cooling, founded in the catalogue"
T_oil1_L_HP1=80[C]
```

```
T_oil1_H_HP1=80 [C]
T_oil1_L_HP2=80 [C]
T_oil1_H_HP2=80 [C]

v_dot_oilflow_L_HP1=300*2
v_dot_oilflow_H_HP1=411*2
v_dot_oilflow_L_HP2=300
v_dot_oilflow_H_HP2=1067


"Assumption oil cycle and compressor"
eta_is_L_HP1=0,83
eta_is_H_HP1=0,78
eta_is_L_HP2=0,76
eta_is_H_HP2=0,80

"Temperature of the water in the datacenter circuit"
T_return_data=32 [C]
T_mid_data=28 [C]
T_forward_data=20 [C]

"!Source calculations"

"district heating"
T_return_dh=45 [C]
"T_mid_dh=70 [C]"
T_forward_dh=85 [C]
h_return_dh=enthalpy(Water;P=P_dh;T=T_return_dh)
h_mid_dh=enthalpy(Water;P=P_dh;T=T_mid_dh)
h_forward_dh=enthalpy(Water;P=P_dh;T=T_forward_dh)
Q_dot_C_HP1=Q_dot_sc_HP1+Q_dot_con_HP1+Q_dot_dsh_H_HP1
Q_dot_C_HP2=Q_dot_sc_HP2+Q_dot_con_HP2+Q_dot_dsh_H_HP2

"datacenter cooling"
h_forward_data=enthalpy(Water;P=P_datacenter;T=T_forward_data)
h_mid_data=enthalpy(Water;P=P_datacenter;T=T_mid_data)
h_return_data=enthalpy(Water;P=P_datacenter;T=T_return_data)
Q_dot_E_HP2=m_dot_data*(-h_mid_data+h_return_data)/1000
Q_dot_E_HP1=m_dot_data*(-h_forward_data+h_mid_data)/1000
Q_dot_condenser=Q_dot_C_HP1+Q_dot_oil_L_HP1+Q_dot_oil_H_HP1+Q_dot_C_HP2+Q
_dot_oil_L_HP2+Q_dot_oil_H_HP2+Q_dot_dsh_L_HP1+Q_dot_dsh_L_HP2
Q_dot_evaporator=Q_dot_E_HP1+Q_dot_E_HP2

"Temperatures fixed by deltaT pinch"
{T_C_L_HP1=42
"T_C_H_HP1=80"
T_C_L_HP2=46}
"T_C_H_HP2=85"
T_E_HP1=T_forward_data-DELTAT_evaporator
T_E_HP2=T_mid_data-DELTAT_evaporator

"Condensing and evaporating pressures"
P_C_L_HP1=pressure(R$;T=T_C_L_HP1;x=0)
P_C_H_HP1=pressure(R$;T=T_C_H_HP1;x=0)
P_C_L_HP2=pressure(R$;T=T_C_L_HP2;x=0)
P_C_H_HP2=pressure(R$;T=T_C_H_HP2;x=0)
P_E_HP1=pressure(R$;T=T_E_HP1;x=1)
P_E_HP2=pressure(R$;T=T_E_HP2;x=1)

"!HP1 cycle calculations"
```

```
"!Low pressure side"

"Evaporator exit"
T_HP1[1]=T_E_HP1
P_HP1[1]=P_E_HP1
h_HP1[1]=enthalpy(R$;P=P_E_HP1;x=1)
s_HP1[1]=entropy(R$;T=T_HP1[1];h=h_HP1[1])
Q_dot_E_HP1=m_dot_ref_L_HP1*(h_HP1[1]-h_HP1[11])/1000

"After compressor"
P_HP1[2]=P_C_L_HP1
s_is_L_HP1=s_HP1[1]
h_is_L_HP1=enthalpy(R$;P=P_C_L_HP1;s=s_is_L_HP1)
eta_is_L_HP1=(h_is_L_HP1-h_HP1[1])/(h_HP1_2W-h_HP1[1])
T_HP1_2W=temperature(R$;P=P_C_L_HP1;h=h_HP1_2W)
W_dot_L_HP1*1000=m_dot_ref_L_HP1*(h_HP1_2W-
h_HP1[1])

"Oil cooling"
T_HP1[2]=T_oil2_L_HP1
v_dot_oil_L_HP1=v_dot_oilflow_L_HP1*convert(l/min;m3/s)
rho_oil_L_HP1=density(Engine_Oil_10W; T=(T_oil1_L_HP1+T_oil2_L_HP1)/2)
m_dot_oil_L_HP1=v_dot_oil_L_HP1*rho_oil_L_HP1
h_oc2_in_L_dh=enthalpy(Water;P=P_dh;T=T_oc2_in_L_dh)
Q_dot_oil_L_HP1=m_dot_dh*(h_oc2_in_L_dh-h_mid_dh)/1000
-Q_dot_oil_L_HP1=m_dot_ref_L_HP1*(h_HP1[2]-h_HP1_2W)/1000
T_HP1[2]=temperature(R$;P=P_C_L_HP1;h=h_HP1[2])
cp_oil_L_HP1=cp(Engine_Oil_10W; T=(T_oil1_L_HP1+T_oil2_L_HP1)/2)
Q_dot_oil_L_HP1=m_dot_oil_L_HP1*cp_oil_L_HP1*(T_oil2_L_HP1-T_oil1_L_HP1)/1000

"Fixing mid temperature"
{T_C_L_HP1=(T_C_H_HP1+T_E_HP1)/2}

"After Low-stage desuperheater"
T_HP1[3]=T_sc2_out_dh+DELTAT_dsh_L_HP1
P_HP1[3]=P_C_L_HP1
h_HP1[3]=enthalpy(R$;P=P_C_L_HP1;T=T_HP1[3])
Q_dot_dsh_L_HP1=m_dot_ref_L_HP1*(h_HP1[2]-
h_HP1[3])/1000
Q_dot_dsh_L_HP1=m_dot_dh*(h_dsh2_in_L_dh-h_sc2_out_dh)/1000
T_dsh2_in_L_dh=temperature(Water;P=P_dh;h=h_dsh2_in_L_dh)

"After liquid separator"
h_HP1[10]=enthalpy(R$;P=P_C_L_HP1;x=0)
T_HP1[10]=t_sat(R$;P=P_C_L_HP1)
P_HP1[10]=P_C_L_HP1

"After expansion valve"
P_HP1[11]=P_E_HP1
h_HP1[11]=h_HP1[10]
T_HP1[11]=temperature(R$;P=P_E_HP1;h=h_HP1[10])


"!High pressure side"

m_dot_ref_H_HP1/m_dot_ref_L_HP1=(h_HP1[10]-h_HP1[3])/(h_HP1[9]-h_HP1[4])

"After vapor separator"
T_HP1[4]=t_sat(R$;P=P_C_L_HP1)
```

```
P_HP1[4]=P_C_L_HP1
h_HP1[4]=enthalpy(R$;P=P_C_L_HP1;x=1)
s_HP1[4]=entropy(R$;T=T_HP1[4];h=h_HP1[4])


"After compressor"
P_HP1[5]=P_C_H_HP1
s_is_H_HP1=s_HP1[4]
h_is_H_HP1=enthalpy(R$;P=P_C_H_HP1;s=s_is_H_HP1)
eta_is_H_HP1=(h_is_H_HP1-h_HP1[4])/(h_HP1_5W-h_HP1[4])
T_HP1_5W=temperature(R$;P=P_C_H_HP1;h=h_HP1_5W)
W_dot_H_HP1*1000=m_dot_ref_H_HP1*(h_HP1_5W-
h_HP1[4])


"Oil cooling"
T_HP1[5]=T_oil2_H_HP1
v_dot_oil_H_HP1=v_dot_oilflow_H_HP1*convert(l/min;m3/s)
rho_oil_H_HP1=density(Engine_Oil_10W; T=(T_oil1_H_HP1+T_oil2_H_HP1)/2)
m_dot_oil_H_HP1=v_dot_oil_H_HP1*rho_oil_H_HP1
h_oc2_in_H_dh=enthalpy(Water;P=P_dh;T=T_oc2_in_H_dh)
Q_dot_oil_H_HP1=m_dot_dh*(h_oc2_in_H_dh-h_oc2_out_L_dh)/1000
-Q_dot_oil_H_HP1=m_dot_ref_H_HP1*(h_HP1[5]-h_HP1_5W)/1000
T_HP1[5]=temperature(R$;P=P_C_H_HP1;h=h_HP1[5])
cp_oil_H_HP1=cp(Engine_Oil_10W; T=(T_oil1_H_HP1+T_oil2_H_HP1)/2)
Q_dot_oil_H_HP1=m_dot_oil_H_HP1*cp_oil_H_HP1*(T_oil2_H_HP1-T_oil1_H_HP1)/1000


"After High stage desuperheater"
P_HP1[6]=P_C_H_HP1
h_HP1[6]=enthalpy(R$;P=P_C_H_HP1;x=1)
T_HP1[6]=t_sat(R$;P=P_C_H_HP1)
Q_dot_dsh_H_HP1=m_dot_dh*(h_mid_dh-h_con1_out_dh)/1000
T_con1_out_dh=temperature(Water;P=P_dh;h=h_con1_out_dh)
Q_dot_dsh_H_HP1=m_dot_ref_H_HP1*(h_HP1[5]-h_HP1[6])/1000

"After condenser"
P_HP1[7]=P_C_H_HP1
T_HP1[7]=T_C_H_HP1
h_HP1[7]=enthalpy(R$;P=P_C_H_HP1;x=0)
Q_dot_con_HP1=m_dot_ref_H_HP1*(h_HP1[6]-h_HP1[7])/1000
Q_dot_con_HP1=m_dot_dh*(h_con1_out_dh-h_con1_in_dh)/1000
T_con1_in_dh=temperature(Water;P=P_dh;h=h_con1_in_dh)


"After subcooler"
T_HP1[8]=T_return_dh+DELTAT_subcooling_HP1
P_HP1[8]=P_C_H_HP1
h_HP1[8]=enthalpy(R$;P=P_C_H_HP1;T=T_HP1[8])
Q_dot_sc_HP1=m_dot_ref_H_HP1*(h_HP1[7]-h_HP1[8])/1000
Q_dot_sc_HP1=m_dot_dh*(h_sc1_out_dh-h_return_dh)/1000
T_sc1_out_dh=temperature(Water;P=P_dh;h=h_sc1_out_dh)

"Pinch and fix condensing temperature"
Q_dot_pinch_H_HP1=m_dot_ref_H_HP1*(enthalpy(R$;P=P_C_H_HP1;x=1)-
enthalpy(R$;P=P_C_H_HP1;x=0))/1000
Q_dot_pinch_H_HP1=m_dot_dh*(h_pinch_H_HP1-h_con1_in_dh)/1000
T_pinch_H_HP1=temperature(Water;P=P_dh;h=h_pinch_H_HP1)
T_C_H_HP1=T_pinch_H_HP1+DELTAT_condenser_HP1
```

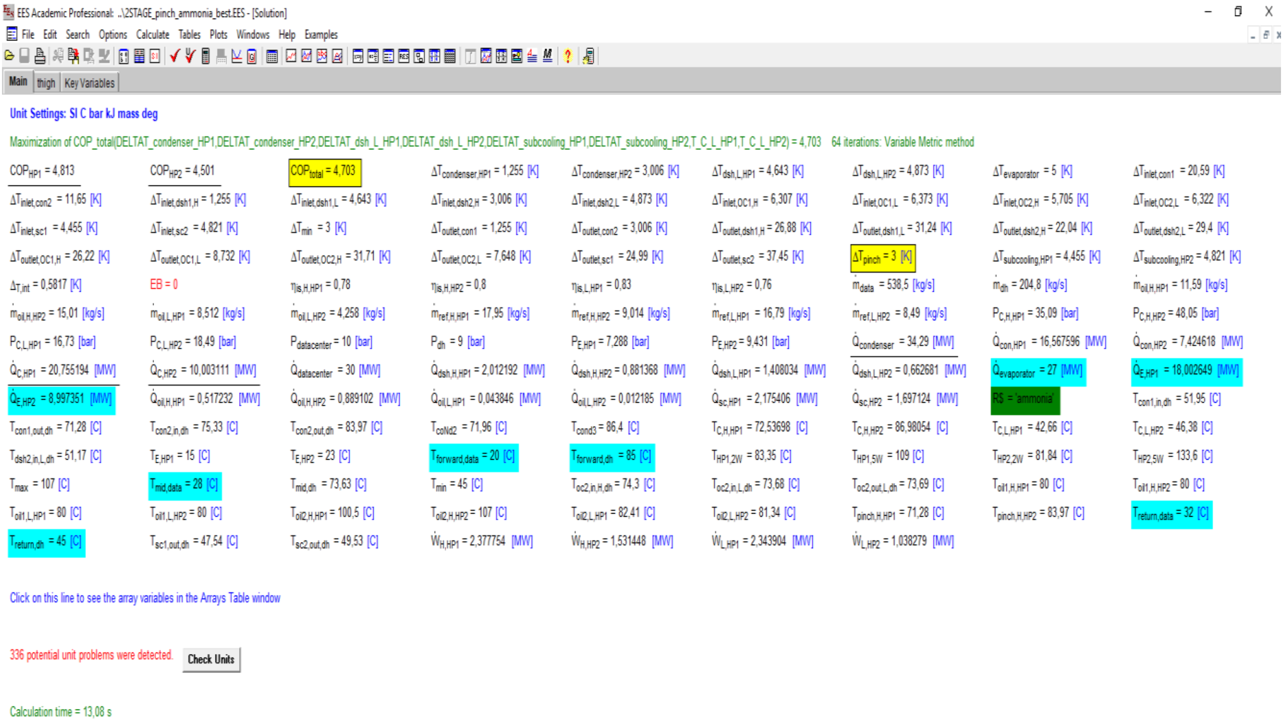
```
"After expansion valve"
P_HP1[9]=P_C_L_HP1
h_HP1[9]=h_HP1[8]
T_HP1[9]=temperature(R$;P=P_C_L_HP1;h=h_HP1[9])


 "performance"
COP_HP1=(Q_dot_C_HP1+Q_dot_oil_L_HP1+Q_dot_oil_H_HP1+Q_dot_dsh_L_HP1)/(W_
dot_L_HP1+W_dot_H_HP1)

"!HP2 cycle calculations"

"!Low pressure side"

 "Evaporator exit"
T_HP2[1]=T_E_HP2
P_HP2[1]=P_E_HP2
h_HP2[1]=enthalpy(R$;P=P_E_HP2;x=1)
s_HP2[1]=entropy(R$;T=T_HP2[1];h=h_HP2[1])
Q_dot_E_HP2=m_dot_ref_L_HP2*(h_HP2[1]-h_HP2[11])/1000


"After compressor"
P_HP2[2]=P_C_L_HP2
s_is_L_HP2=s_HP2[1]
h_is_L_HP2=enthalpy(R$;P=P_C_L_HP2;s=s_is_L_HP2)
eta_is_L_HP2=(h_is_L_HP2-h_HP2[1])/(h_HP2_2W-h_HP2[1])
T_HP2_2W=temperature(R$;P=P_C_L_HP2;h=h_HP2_2W)
W_dot_L_HP2*1000=m_dot_ref_L_HP2*(h_HP2_2W-h_HP2[1])

"Oil cooling"
T_HP2[2]=T_oil2_L_HP2
v_dot_oil_L_HP2=v_dot_oilflow_L_HP2*convert(l/min;m3/s)
rho_oil_L_HP2=density(Engine_Oil_10W; T=(T_oil1_L_HP2+T_oil2_L_HP2)/2)
m_dot_oil_L_HP2=v_dot_oil_L_HP2*rho_oil_L_HP2
h_oc2_out_L_dh=enthalpy(Water;P=P_dh;T=T_oc2_out_L_dh)
Q_dot_oil_L_HP2=m_dot_dh*(h_oc2_out_L_dh-h_oc2_in_L_dh)/1000
-Q_dot_oil_L_HP2=m_dot_ref_L_HP2*(h_HP2[2]-h_HP2_2W)/1000
T_HP2[2]=temperature(R$;P=P_C_L_HP2;h=h_HP2[2])
cp_oil_L_HP2=cp(Engine_Oil_10W; T=(T_oil1_L_HP2+T_oil2_L_HP2)/2)
Q_dot_oil_L_HP2=m_dot_oil_L_HP2*cp_oil_L_HP2*(T_oil2_L_HP2-T_oil1_L_HP2)/1000

"Fixing mid temperature"

{T_C_L_HP2=(T_C_H_HP2+T_E_HP2)/2}

"After Low-stage desuperheater"
T_HP2[3]=T_dsh2_in_L_dh+DELTAT_dsh_L_HP2
P_HP2[3]=P_C_L_HP2
h_HP2[3]=enthalpy(R$;P=P_C_L_HP2;T=T_HP2[3])
Q_dot_dsh_L_HP2=m_dot_ref_L_HP2*(h_HP2[2]-
h_HP2[3])/1000
Q_dot_dsh_L_HP2=m_dot_dh*(h_con1_in_dh-h_dsh2_in_L_dh)/1000


"After liquid separator"
h_HP2[10]=enthalpy(R$;P=P_C_L_HP2;x=0)
T_HP2[10]=t_sat(R$;P=P_C_L_HP2)
P_HP2[10]=P_C_L_HP2
```

```
"After expansion valve"
P_HP2[11]=P_E_HP2
h_HP2[11]=h_HP2[10]
T_HP2[11]=temperature(R$;P=P_E_HP2;h=h_HP2[10])

"!High pressure side"

m_dot_ref_H_HP2/m_dot_ref_L_HP2=(h_HP2[10]-h_HP2[3])/(h_HP2[9]-h_HP2[4])

"After vapor separator"
T_HP2[4]=t_sat(R$;P=P_C_L_HP2)
P_HP2[4]=P_C_L_HP2
h_HP2[4]=enthalpy(R$;P=P_C_L_HP2;x=1)
s_HP2[4]=entropy(R$;T=T_HP2[4];h=h_HP2[4])


"After compressor"
P_HP2[5]=P_C_H_HP2
s_is_H_HP2=s_HP2[4]
h_is_H_HP2=enthalpy(R$;P=P_C_H_HP2;s=s_is_H_HP2)
eta_is_H_HP2=(h_is_H_HP2-h_HP2[4])/(h_HP2_5W-h_HP2[4])
T_HP2_5W=temperature(R$;P=P_C_H_HP2;h=h_HP2_5W)
W_dot_H_HP2*1000=m_dot_ref_H_HP2*(h_HP2_5W-
h_HP2[4])


"Oil cooling"
T_HP2[5]=T_oil2_H_HP2
v_dot_oil_H_HP2=v_dot_oilflow_H_HP2*convert(l/min;m3/s)
rho_oil_H_HP2=density(Engine_Oil_10W; T=(T_oil1_H_HP2+T_oil2_H_HP2)/2)
m_dot_oil_H_HP2=v_dot_oil_H_HP2*rho_oil_H_HP2
h_con2_in_dh=enthalpy(Water;P=P_dh;T=T_con2_in_dh)
Q_dot_oil_H_HP2=m_dot_dh*(h_con2_in_dh-h_oc2_in_H_dh)/1000
-Q_dot_oil_H_HP2=m_dot_ref_H_HP2*(h_HP2[5]-h_HP2_5W)/1000
T_HP2[5]=temperature(R$;P=P_C_H_HP2;h=h_HP2[5])
cp_oil_H_HP2=cp(Engine_Oil_10W; T=(T_oil1_H_HP2+T_oil2_H_HP2)/2)
Q_dot_oil_H_HP2=m_dot_oil_H_HP2*cp_oil_H_HP2*(T_oil2_H_HP2-T_oil1_H_HP2)/1000


"After High stage desuperheater"
P_HP2[6]=P_C_H_HP2
h_HP2[6]=enthalpy(R$;P=P_C_H_HP2;x=1)
T_HP2[6]=t_sat(R$;P=P_C_H_HP2)
Q_dot_dsh_H_HP2=m_dot_dh*(h_forward_dh-h_con2_out_dh)/1000
T_con2_out_dh=temperature(Water;P=P_dh;h=h_con2_out_dh)
Q_dot_dsh_H_HP2=m_dot_ref_H_HP2*(h_HP2[5]-h_HP2[6])/1000

"After condenser"
P_HP2[7]=P_C_H_HP2
T_HP2[7]=T_C_H_HP2
h_HP2[7]=enthalpy(R$;P=P_C_H_HP2;x=0)
Q_dot_con_HP2=m_dot_ref_H_HP2*(h_HP2[6]-h_HP2[7])/1000
Q_dot_con_HP2=m_dot_dh*(h_con2_out_dh-h_con2_in_dh)/1000


"After subcooler"
T_HP2[8]=T_sc1_out_dh+DELTAT_subcooling_HP2
P_HP2[8]=P_C_H_HP2
h_HP2[8]=enthalpy(R$;P=P_C_H_HP2;T=T_HP2[8])
Q_dot_sc_HP2=m_dot_ref_H_HP2*(h_HP2[7]-h_HP2[8])/1000
```

```
Q_dot_sc_HP2=m_dot_dh*(h_sc2_out_dh-h_sc1_out_dh)/1000
T_sc2_out_dh=temperature(Water;P=P_dh;h=h_sc2_out_dh)


"Pinch and fix condensing temperature"
Q_dot_pinch_H_HP2=m_dot_ref_H_HP2*(enthalpy(R$;P=P_C_H_HP2;x=1)-
enthalpy(R$;P=P_C_H_HP2;x=0))/1000
Q_dot_pinch_H_HP2=m_dot_dh*(h_pinch_H_HP2-h_con2_in_dh)/1000
T_pinch_H_HP2=temperature(Water;P=P_dh;h=h_pinch_H_HP2)
T_C_H_HP2=T_pinch_H_HP2+DELTAT_condenser_HP2

"After expansion valve"
P_HP2[9]=P_C_L_HP2
h_HP2[9]=h_HP2[8]
T_HP2[9]=temperature(R$;P=P_C_L_HP2;h=h_HP2[9])


 "performance"
COP_HP2=(Q_dot_C_HP2+Q_dot_oil_L_HP2+Q_dot_oil_H_HP2+Q_dot_dsh_L_HP2)/(W_
dot_L_HP2+W_dot_H_HP2)


COP_total=(Q_dot_condenser)/(W_dot_L_HP2+W_dot_H_HP2+W_dot_L_HP1+W_dot_H_
HP1)


EB=Q_dot_condenser-
(W_dot_L_HP2+W_dot_H_HP2+W_dot_L_HP1+W_dot_H_HP1+Q_dot_evaporator)


"!HEN pinch check"

DELTAT_inlet_sc1=T_HP1[8]-T_return_dh
DELTAT_outlet_sc1=T_HP1[7]-T_sc1_out_dh

DELTAT_inlet_sc2=T_HP2[8]-T_sc1_out_dh
DELTAT_outlet_sc2=T_HP2[7]-T_sc2_out_dh

DELTAT_inlet_dsh1_L=T_HP1[3]-T_sc2_out_dh
DELTAT_outlet_dsh1_L=T_HP1[2]-T_dsh2_in_L_dh

DELTAT_inlet_dsh2_L=T_HP2[3]-T_dsh2_in_L_dh
DELTAT_outlet_dsh2_L=T_HP2[2]-T_con1_in_dh

DELTAT_inlet_con1=T_HP1[6]-T_con1_in_dh
DELTAT_outlet_con1=T_HP1[7]-T_con1_out_dh

DELTAT_inlet_dsh1_H=T_HP1[6]-T_con1_out_dh
DELTAT_outlet_dsh1_H=T_HP1[5]-T_mid_dh

DELTAT_inlet_OC1_L=T_oil1_L_HP1-T_mid_dh
DELTAT_outlet_OC1_L=T_oil2_L_HP1-T_oc2_in_L_dh

DELTAT_inlet_OC2_L=T_oil1_L_HP2-T_oc2_in_L_dh
DELTAT_outlet_OC2_L=T_oil2_L_HP2-T_oc2_out_L_dh

DELTAT_inlet_OC1_H=T_oil1_H_HP1-T_oc2_out_L_dh
DELTAT_outlet_OC1_H=T_oil2_H_HP1-T_oc2_in_H_dh

DELTAT_inlet_OC2_H=T_oil1_H_HP2-T_oc2_in_H_dh
```

```
DELTAT_outlet_OC2_H=T_oil2_H_HP2-T_con2_in_dh

DELTAT_inlet_con2=T_HP2[6]-T_con2_in_dh
DELTAT_outlet_con2=T_HP2[7]-T_con2_out_dh

DELTAT_inlet_dsh2_H=T_HP2[6]-T_con2_out_dh
DELTAT_outlet_dsh2_H=T_HP2[5]-T_forward_dh


"!plot script"

"T-Q condenser"

T_districtheating[1]=T_return_dh
T_districtheating[2]=T_sc1_out_dh
T_districtheating[3]=T_sc2_out_dh
T_districtheating[4]=T_dsh2_in_L_dh
T_districtheating[5]=T_con1_in_dh
T_districtheating[6]=T_con1_out_dh
T_districtheating[7]=T_mid_dh
T_districtheating[8]=T_oc2_in_L_dh
T_districtheating[9]=T_oc2_out_L_dh
T_districtheating[10]=T_oc2_in_H_dh
T_districtheating[11]=T_con2_in_dh
T_districtheating[12]=T_con2_out_dh
T_districtheating[13]=T_forward_dh
Q_dot_dh[1]=0
Q_dot_dh[2]=Q_dot_sc_HP1
Q_dot_dh[3]=Q_dot_dh[2]+Q_dot_sc_HP2
Q_dot_dh[4]=Q_dot_dh[3]+Q_dot_dsh_L_HP1
Q_dot_dh[5]=Q_dot_dh[4]+Q_dot_dsh_L_HP2
Q_dot_dh[6]=Q_dot_dh[5]+Q_dot_con_HP1
Q_dot_dh[7]=Q_dot_dh[6]+Q_dot_dsh_H_HP1
Q_dot_dh[8]=Q_dot_dh[7]+Q_dot_oil_L_HP1
Q_dot_dh[9]=Q_dot_dh[8]+Q_dot_oil_L_HP2
Q_dot_dh[10]=Q_dot_dh[9]+Q_dot_oil_H_HP1
Q_dot_dh[11]=Q_dot_dh[10]+Q_dot_oil_H_HP2
Q_dot_dh[12]=Q_dot_dh[11]+Q_dot_con_HP2
Q_dot_dh[13]=Q_dot_condenser

T_sc_HP1[1]=T_HP1[8]
T_sc_HP1[2]=T_C_H_HP1
T_sc_HP2[2]=T_HP2[8]
T_sc_HP2[3]=T_C_H_HP2
T_dsh_L_HP1[3]=T_HP1[3]
T_dsh_L_HP1[4]=T_HP1[2]
T_dsh_L_HP2[4]=T_HP2[3]
T_dsh_L_HP2[5]=T_HP2[2]
T_con_HP1[5]=T_C_H_HP1
T_con_HP1[6]=T_C_H_HP1
T_dsh_H_HP1[6]=T_C_H_HP1
T_dsh_H_HP1[7]=T_HP1[5]
T_oil_L_HP1[7]=T_oil1_L_HP1
T_oil_L_HP1[8]=T_oil2_L_HP1
T_oil_L_HP2[8]=T_oil1_L_HP2
T_oil_L_HP2[9]=T_oil2_L_HP2
T_oil_H_HP1[9]=T_oil1_H_HP1
T_oil_H_HP1[10]=T_oil2_H_HP1
T_oil_H_HP2[10]=T_oil1_H_HP2
T_oil_H_HP2[11]=T_oil2_H_HP2
```

T_con_HP2[11]=T_C_H_HP2
T_con_HP2[12]=T_C_H_HP2
T_dsh_H_HP2[12]=T_C_H_HP2
T_dsh_H_HP2[13]=T_HP2[5]



HEN of the ammonia two-stage HP.

"Two 2-stage heat pump connected in parallel with water as source"
"Assumptions:
-Evaporators capacity fixed
-temperatures are taken from the report for heat source and district heating
-assumptions are made regarding the temperature differences at condenser and evaporator
-evaporation and condensation are supposed happening at constant pressure
-no electrical or mechanical losses are considered for the compressor
-no superheating is considered"

"!Initial considerations and inputs"

DELTAT_evaporator=5
R$='ammonia'
Q_dot_datacenter=30 [MW]
Q_dot_evaporator=0,9*Q_dot_datacenter

P_dh= 9 [bar] "Assumption of the pressure in the district heating circuit"
P_datacenter=10 [bar] "Assumption of the pressure in the datacenter circuit"

"Assumption of the temperature in the inlet of the oil cooling, founded in the catalogue"
T_oil1_L_HP1=80[C]

```
T_oil1_H_HP1=80 [C]
T_oil1_L_HP2=80 [C]
T_oil1_H_HP2=80 [C]

v_dot_oilflow_L_HP1=300*2
v_dot_oilflow_H_HP1=411*2
v_dot_oilflow_L_HP2=300
v_dot_oilflow_H_HP2=1067


"Assumption oil cycle and compressor"
eta_is_L_HP1=0,83
eta_is_H_HP1=0,78
eta_is_L_HP2=0,76
eta_is_H_HP2=0,80

"Temperature of the water in the datacenter circuit"
T_return_data=32 [C]
T_mid_data=28 [C]
T_forward_data=20 [C]

"!Additional inputs taken from previous simulations"

T_HP1[8]=49,45272
T_HP1[3]=54,16963
T_HP2[3]=56,0456
T_HP2[8]=52,36649
T_oil2_L_HP1=81,41288
T_oil2_L_HP2=81,34975
T_oil2_H_HP1=100,51605
T_oil2_H_HP2=107,04659

"!Source calculations"

"district heating"
T_return_dh=45 [C]
"T_mid_dh=70 [C]"
T_forward_dh=85 [C]
h_return_dh=enthalpy(Water;P=P_dh;T=T_return_dh)
{h_mid_dh=enthalpy(Water;P=P_dh;T=T_mid_dh)}
h_forward_dh=enthalpy(Water;P=P_dh;T=T_forward_dh)
Q_dot_C_HP1=Q_dot_sc_HP1+Q_dot_con_HP1+Q_dot_dsh_H_HP1
Q_dot_C_HP2=Q_dot_sc_HP2+Q_dot_con_HP2+Q_dot_dsh_H_HP2

"datacenter cooling"
h_forward_data=enthalpy(Water;P=P_datacenter;T=T_forward_data)
h_mid_data=enthalpy(Water;P=P_datacenter;T=T_mid_data)
h_return_data=enthalpy(Water;P=P_datacenter;T=T_return_data)
Q_dot_E_HP2=8,997
Q_dot_E_HP1=18
Q_dot_condenser=Q_dot_C_HP1+Q_dot_oil_L_HP1+Q_dot_oil_H_HP1+Q_dot_C_HP2+Q
_dot_oil_L_HP2+Q_dot_oil_H_HP2+Q_dot_dsh_L_HP1+Q_dot_dsh_L_HP2


"Temperatures fixed by deltaT pinch"
T_C_L_HP1=42,66                         "Mid temperatures fixed due to
compressors in RTselect have been already selected at that temperature"
T_C_H_HP1=72,54084
T_C_L_HP2=46,38
T_C_H_HP2=86,98673
```

```
T_E_HP1=T_forward_data-DELTAT_evaporator
T_E_HP2=T_mid_data-DELTAT_evaporator

"Condensing and evaporating pressures"
P_C_L_HP1=pressure(R$;T=T_C_L_HP1;x=0)
P_C_H_HP1=pressure(R$;T=T_C_H_HP1;x=0)
P_C_L_HP2=pressure(R$;T=T_C_L_HP2;x=0)
P_C_H_HP2=pressure(R$;T=T_C_H_HP2;x=0)
P_E_HP1=pressure(R$;T=T_E_HP1;x=1)
P_E_HP2=pressure(R$;T=T_E_HP2;x=1)

"Mass flow assumptions"
m_dot_dh=m_dot_dh1+m_dot_dh2+m_dot_dh3+m_dot_dh4
m_dot_dh=m_dot_dh5+m_dot_dh6+m_dot_dh7+m_dot_dh8+m_dot_dh9+m_dot_dh11+m_d
ot_dh12+m_dot_dh10
m_dot_dh=m_dot_dh13+m_dot_dh15+m_dot_dh16+m_dot_dh14


"!HP1 cycle calculations"

"!Low pressure side"

"Evaporator exit"
T_HP1[1]=T_E_HP1
P_HP1[1]=P_E_HP1
h_HP1[1]=enthalpy(R$;P=P_E_HP1;x=1)
s_HP1[1]=entropy(R$;T=T_HP1[1];h=h_HP1[1])
Q_dot_E_HP1=m_dot_ref_L_HP1*(h_HP1[1]-h_HP1[11])/1000

"After compressor"
P_HP1[2]=P_C_L_HP1
s_is_L_HP1=s_HP1[1]
h_is_L_HP1=enthalpy(R$;P=P_C_L_HP1;s=s_is_L_HP1)
eta_is_L_HP1=(h_is_L_HP1-h_HP1[1])/(h_HP1_2W-h_HP1[1])
T_HP1_2W=temperature(R$;P=P_C_L_HP1;h=h_HP1_2W)
W_dot_L_HP1=2,344025 [MW]


"Oil cooling"
T_HP1[2]=T_oil2_L_HP1
v_dot_oil_L_HP1=v_dot_oilflow_L_HP1*convert(l/min;m3/s)
rho_oil_L_HP1=density(Engine_Oil_10W; T=(T_oil1_L_HP1+T_oil2_L_HP1)/2)
m_dot_oil_L_HP1=v_dot_oil_L_HP1*rho_oil_L_HP1
"h_oc2_in_L_dh=enthalpy(Water;P=P_dh;T=T_oc2_in_L_dh)"
Q_dot_oil_L_HP1=m_dot_dh9*(h_con2_in-dh-h_con1_out_dh)/1000
-Q_dot_oil_L_HP1=-0,043887
T_HP1[2]=temperature(R$;P=P_C_L_HP1;h=h_HP1[2])

cp_oil_L_HP1=cp(Engine_Oil_10W; T=(T_oil1_L_HP1+T_oil2_L_HP1)/2)


"After Low-stage desuperheater"
{T_HP1[3]=T_return_dh+DELTAT_dsh_L_HP1}
P_HP1[3]=P_C_L_HP1
h_HP1[3]=enthalpy(R$;P=P_C_L_HP1;T=T_HP1[3])
Q_dot_dsh_L_HP1=Q_dot_dsh_L_HP1_A+Q_dot_dsh_L_HP1_B
Q_dot_dsh_L_HP1=1,408134
Q_dot_dsh_L_HP1_A=m_dot_ref_L_HP1*(enthalpy(R$;T=T_C_H_HP1;P=P_C_L_HP1)-
h_HP1[3])/1000
```

```
"Q_dot_dsh_L_HP1_B=m_dot_ref_L_HP1*(h_HP1[2]-
enthalpy(R$;T=T_C_H_HP1;P=P_C_L_HP1))/1000"
Q_dot_dsh_L_HP1_A=m_dot_dh3*(h_con1_in_dh-h_return_dh)/1000
Q_dot_dsh_L_HP1_B=m_dot_dh7*(h_con2_in_dh-h_con1_out_dh)/1000


"After liquid separator"
h_HP1[10]=enthalpy(R$;P=P_C_L_HP1;x=0)
T_HP1[10]=t_sat(R$;P=P_C_L_HP1)
P_HP1[10]=P_C_L_HP1

"After expansion valve"
P_HP1[11]=P_E_HP1
h_HP1[11]=h_HP1[10]
T_HP1[11]=temperature(R$;P=P_E_HP1;h=h_HP1[10])


"!High pressure side"

m_dot_ref_H_HP1/m_dot_ref_L_HP1=(h_HP1[10]-h_HP1[3])/(h_HP1[9]-h_HP1[4])

"After vapor separator"
T_HP1[4]=t_sat(R$;P=P_C_L_HP1)
P_HP1[4]=P_C_L_HP1
h_HP1[4]=enthalpy(R$;P=P_C_L_HP1;x=1)
s_HP1[4]=entropy(R$;T=T_HP1[4];h=h_HP1[4])


"After compressor"
P_HP1[5]=P_C_H_HP1
s_is_H_HP1=s_HP1[4]
h_is_H_HP1=enthalpy(R$;P=P_C_H_HP1;s=s_is_H_HP1)
eta_is_H_HP1=(h_is_H_HP1-h_HP1[4])/(h_HP1_5W-h_HP1[4])
T_HP1_5W=temperature(R$;P=P_C_H_HP1;h=h_HP1_5W)
W_dot_H_HP1=2,377902 [MW]



"Oil cooling"
T_HP1[5]=T_oil2_H_HP1
v_dot_oil_H_HP1=v_dot_oilflow_H_HP1*convert(l/min;m3/s)
rho_oil_H_HP1=density(Engine_Oil_10W; T=(T_oil1_H_HP1+T_oil2_H_HP1)/2)
m_dot_oil_H_HP1=v_dot_oil_H_HP1*rho_oil_H_HP1
"h_oc2_in_H_dh=enthalpy(Water;P=P_dh;T=T_oc2_in_H_dh)"
Q_dot_oil_H_HP1_A=m_dot_dh10*(h_con2_in_dh-h_con1_out_dh)/1000
Q_dot_oil_H_HP1_B=m_dot_dh14*(h_forward_dh-h_con2_out_dh)/1000
Q_dot_oil_H_HP1=Q_dot_oil_H_HP1_A+Q_dot_oil_H_HP1_B
Q_dot_oil_H_HP1=0,517351 [MW]
T_HP1[5]=temperature(R$;P=P_C_H_HP1;h=h_HP1[5])

cp_oil_H_HP1=cp(Engine_Oil_10W; T=(T_oil1_H_HP1+T_oil2_H_HP1)/2)
Q_dot_oil_H_HP1_A=m_dot_oil_H_HP1*cp_oil_H_HP1*(T_C_H_HP2-T_oil1_H_HP1)/1000

"After High stage desuperheater"
P_HP1[6]=P_C_H_HP1
h_HP1[6]=enthalpy(R$;P=P_C_H_HP1;x=1)
T_HP1[6]=t_sat(R$;P=P_C_H_HP1)
Q_dot_dsh_H_HP1=Q_dot_dsh_H_HP1_A+Q_dot_dsh_H_HP1_B
Q_dot_dsh_H_HP1=2,012282 [MW]
Q_dot_dsh_H_HP1_A=m_dot_dh8*(h_con2_in_dh-h_con1_out_dh)/1000
```

```
Q_dot_dsh_H_HP1_B=m_dot_dh13*(h_forward_dh-h_con2_out_dh)/1000
Q_dot_dsh_H_HP1_A=m_dot_ref_H_HP1*(enthalpy(R$;T=T_C_H_HP2;P=P_C_H_HP1)-
h_HP1[6])/1000
"Q_dot_dsh_H_HP1_B=m_dot_ref_H_HP1*(h_HP1[5]-
enthalpy(R$;T=T_C_H_HP2;P=P_C_H_HP1))/1000"

"After condenser"
P_HP1[7]=P_C_H_HP1
T_HP1[7]=T_C_H_HP1
h_HP1[7]=enthalpy(R$;P=P_C_H_HP1;x=0)
"Q_dot_con_HP1=m_dot_ref_H_HP1*(h_HP1[6]-h_HP1[7])/1000"
Q_dot_con_HP1=m_dot_dh*(h_con1_out_dh-h_con1_in_dh)/1000
 Q_dot_con_HP1=16,566934 [MW]



"After subcooler"
{T_HP1[8]=T_return_dh+DELTAT_subcooling_HP1}
P_HP1[8]=P_C_H_HP1
h_HP1[8]=enthalpy(R$;P=P_C_H_HP1;T=T_HP1[8])
{Q_dot_sc_HP1=m_dot_ref_H_HP1*(h_HP1[7]-h_HP1[8])/1000}
Q_dot_sc_HP1=m_dot_dh4*(h_con1_in_dh-h_return_dh)/1000
 Q_dot_sc_HP1=2,175988 [MW]


"After expansion valve"
P_HP1[9]=P_C_L_HP1
h_HP1[9]=h_HP1[8]
T_HP1[9]=temperature(R$;P=P_C_L_HP1;h=h_HP1[9])


 "performance"
COP_HP1=(Q_dot_C_HP1+Q_dot_oil_L_HP1+Q_dot_oil_H_HP1+Q_dot_dsh_L_HP1)/(W_
dot_L_HP1+W_dot_H_HP1)

"!HP2 cycle calculations"

"!Low pressure side"

 "Evaporator exit"
T_HP2[1]=T_E_HP2
P_HP2[1]=P_E_HP2
h_HP2[1]=enthalpy(R$;P=P_E_HP2;x=1)
s_HP2[1]=entropy(R$;T=T_HP2[1];h=h_HP2[1])
Q_dot_E_HP2=m_dot_ref_L_HP2*(h_HP2[1]-h_HP2[11])/1000


"After compressor"
P_HP2[2]=P_C_L_HP2
s_is_L_HP2=s_HP2[1]
h_is_L_HP2=enthalpy(R$;P=P_C_L_HP2;s=s_is_L_HP2)
eta_is_L_HP2=(h_is_L_HP2-h_HP2[1])/(h_HP2_2W-h_HP2[1])
T_HP2_2W=temperature(R$;P=P_C_L_HP2;h=h_HP2_2W)
{W_dot_L_HP2*1000=m_dot_ref_L_HP2*(h_HP2_2W-h_HP2[1])}
W_dot_L_HP2=1,038520 [MW]


"Oil cooling"
T_HP2[2]=T_oil2_L_HP2
v_dot_oil_L_HP2=v_dot_oilflow_L_HP2*convert(l/min;m3/s)
```

```
rho_oil_L_HP2=density(Engine_Oil_10W; T=(T_oil1_L_HP2+T_oil2_L_HP2)/2)
m_dot_oil_L_HP2=v_dot_oil_L_HP2*rho_oil_L_HP2
"h_oc2_out_L_dh=enthalpy(Water;P=P_dh;T=T_oc2_out_L_dh)"
Q_dot_oil_L_HP2=m_dot_dh11*(h_con2_in_dh-h_con1_out_dh)/1000
-Q_dot_oil_L_HP2=-0,012267
T_HP2[2]=temperature(R$;P=P_C_L_HP2;h=h_HP2[2])

cp_oil_L_HP2=cp(Engine_Oil_10W; T=(T_oil1_L_HP2+T_oil2_L_HP2)/2)

"Fixing mid temperature"

{T_C_L_HP2=(T_C_H_HP2+T_E_HP2)/2}

"After Low-stage desuperheater"
{T_HP2[3]=T_return_dh+DELTAT_dsh_L_HP2}
P_HP2[3]=P_C_L_HP2
h_HP2[3]=enthalpy(R$;P=P_C_L_HP2;T=T_HP2[3])
Q_dot_dsh_L_HP2=Q_dot_dsh_L_HP2_A+Q_dot_dsh_L_HP2_B
Q_dot_dsh_L_HP2=0,662904 [MW]
Q_dot_dsh_L_HP2_A=m_dot_ref_L_HP2*(enthalpy(R$;T=T_C_H_HP1;P=P_C_L_HP2)-
h_HP2[3])/1000
"Q_dot_dsh_L_HP2_B=m_dot_ref_L_HP2*(h_HP2[2]-
enthalpy(R$;T=T_C_H_HP1;P=P_C_L_HP2))/1000"
Q_dot_dsh_L_HP2_A=m_dot_dh1*(h_con1_in_dh-h_return_dh)/1000
Q_dot_dsh_L_HP2_B=m_dot_dh5*(h_con2_in_dh-h_con1_out_dh)/1000


"After liquid separator"
h_HP2[10]=enthalpy(R$;P=P_C_L_HP2;x=0)
T_HP2[10]=t_sat(R$;P=P_C_L_HP2)
P_HP2[10]=P_C_L_HP2

"After expansion valve"
P_HP2[11]=P_E_HP2
h_HP2[11]=h_HP2[10]
T_HP2[11]=temperature(R$;P=P_E_HP2;h=h_HP2[10])

"!High pressure side"

m_dot_ref_H_HP2/m_dot_ref_L_HP2=(h_HP2[10]-h_HP2[3])/(h_HP2[9]-h_HP2[4])

"After vapor separator"
T_HP2[4]=t_sat(R$;P=P_C_L_HP2)
P_HP2[4]=P_C_L_HP2
h_HP2[4]=enthalpy(R$;P=P_C_L_HP2;x=1)
s_HP2[4]=entropy(R$;T=T_HP2[4];h=h_HP2[4])


"After compressor"
P_HP2[5]=P_C_H_HP2
s_is_H_HP2=s_HP2[4]
h_is_H_HP2=enthalpy(R$;P=P_C_H_HP2;s=s_is_H_HP2)
eta_is_H_HP2=(h_is_H_HP2-h_HP2[4])/(h_HP2_5W-h_HP2[4])
T_HP2_5W=temperature(R$;P=P_C_H_HP2;h=h_HP2_5W)
"W_dot_H_HP2*1000=m_dot_ref_H_HP2*(h_HP2_5W-h_HP2[4])"
W_dot_H_HP2=1,531449 [MW]



"Oil cooling"
```

```
T_HP2[5]=T_oil2_H_HP2
v_dot_oil_H_HP2=v_dot_oilflow_H_HP2*convert(l/min;m3/s)
rho_oil_H_HP2=density(Engine_Oil_10W; T=(T_oil1_H_HP2+T_oil2_H_HP2)/2)
m_dot_oil_H_HP2=v_dot_oil_H_HP2*rho_oil_H_HP2
Q_dot_oil_H_HP2=Q_dot_oil_H_HP2_A+Q_dot_oil_H_HP2_B
Q_dot_oil_H_HP2_A=m_dot_dh12*(h_con2_in_dh-h_con1_out_dh)/1000
Q_dot_oil_H_HP2_B=m_dot_dh15*(h_forward_dh-h_con2_out_dh)/1000
"-Q_dot_oil_H_HP2=m_dot_ref_H_HP2*(h_HP2[5]-h_HP2_5W)/1000 "
Q_dot_oil_H_HP2=0,889224 [MW]
T_HP2[5]=temperature(R$;P=P_C_H_HP2;h=h_HP2[5])

cp_oil_H_HP2=cp(Engine_Oil_10W; T=(T_oil1_H_HP2+T_oil2_H_HP2)/2)
Q_dot_oil_H_HP2_A=m_dot_oil_H_HP2*cp_oil_H_HP2*(T_C_H_HP2-T_oil1_H_HP2)/1000
"Q_dot_oil_H_HP2_B=m_dot_oil_H_HP2*cp_oil_H_HP2*(T_oil2_H_HP2-
T_C_H_HP2)/1000"


"After High stage desuperheater"
P_HP2[6]=P_C_H_HP2
h_HP2[6]=enthalpy(R$;P=P_C_H_HP2;x=1)
T_HP2[6]=t_sat(R$;P=P_C_H_HP2)
Q_dot_dsh_H_HP2=m_dot_dh16*(h_forward_dh-h_con2_out_dh)/1000
T_con2_out_dh=temperature(Water;P=P_dh;h=h_con2_out_dh)
T_con2_in_dh=temperature(Water;P=P_dh;h=h_con2_in_dh)
T_con1_out_dh=temperature(Water;P=P_dh;h=h_con1_out_dh)
T_con1_in_dh=temperature(Water;P=P_dh;h=h_con1_in_dh)
"Q_dot_dsh_H_HP2=m_dot_ref_H_HP2*(h_HP2[5]-h_HP2[6])/1000"
Q_dot_dsh_H_HP2=0,881343 [MW]


"After condenser"
P_HP2[7]=P_C_H_HP2
T_HP2[7]=T_C_H_HP2
h_HP2[7]=enthalpy(R$;P=P_C_H_HP2;x=0)
"Q_dot_con_HP2=m_dot_ref_H_HP2*(h_HP2[6]-h_HP2[7])/1000"
Q_dot_con_HP2=m_dot_dh*(h_con2_out_dh-h_con2_in_dh)/1000
 Q_dot_con_HP2=7,424237 [MW]


"After subcooler"
{T_HP2[8]=T_return_dh+DELTAT_subcooling_HP2}
P_HP2[8]=P_C_H_HP2
h_HP2[8]=enthalpy(R$;P=P_C_H_HP2;T=T_HP2[8])
Q_dot_sc_HP2=Q_dot_sc_HP2_A+Q_dot_sc_HP2_B
Q_dot_sc_HP2=1,697344 [MW]
Q_dot_sc_HP2_A=m_dot_ref_H_HP2*(enthalpy(R$;T=T_C_H_HP1;P=P_C_H_HP2)-
h_HP2[8])/1000
"Q_dot_sc_HP2_B=m_dot_ref_H_HP2*(h_HP2[7]-
enthalpy(R$;T=T_C_H_HP1;P=P_C_H_HP2))/1000"
Q_dot_sc_HP2_A=m_dot_dh2*(h_con1_in_dh-h_return_dh)/1000
Q_dot_sc_HP2_B=m_dot_dh6*(h_con2_in_dh-h_con1_out_dh)/1000


{"Pinch and fix condensing temperature"
Q_dot_pinch_H_HP2=m_dot_ref_H_HP2*(enthalpy(R$;P=P_C_H_HP2;x=1)-
enthalpy(R$;P=P_C_H_HP2;x=0))/1000
Q_dot_pinch_H_HP2=m_dot_dh*(h_pinch_H_HP2-h_con2_in_dh)/1000
T_pinch_H_HP2=temperature(Water;P=P_dh;h=h_pinch_H_HP2)
T_C_H_HP2=T_pinch_H_HP2+DELTAT_condenser_HP2
 }
```

```
"After expansion valve"
P_HP2[9]=P_C_L_HP2
h_HP2[9]=h_HP2[8]
T_HP2[9]=temperature(R$;P=P_C_L_HP2;h=h_HP2[9])


 "performance"
COP_HP2=(Q_dot_C_HP2+Q_dot_oil_L_HP2+Q_dot_oil_H_HP2+Q_dot_dsh_L_HP2)/(W_dot_L_HP2+W_dot_H_HP2)


COP_total=(Q_dot_condenser)/(W_dot_L_HP2+W_dot_H_HP2+W_dot_L_HP1+W_dot_H_HP1)


EB=Q_dot_condenser-(W_dot_L_HP2+W_dot_H_HP2+W_dot_L_HP1+W_dot_H_HP1+Q_dot_evaporator)


"!HEN pinch check"

DELTAT_inlet_sc1=T_HP1[8]-T_return_dh
DELTAT_outlet_sc1=T_HP1[7]-T_con1_in_dh

DELTAT_inlet_sc2_A=T_HP2[8]-T_return_dh
DELTAT_outlet_sc2_A=T_C_H_HP1-T_con1_in_dh

DELTAT_inlet_sc2_B=T_C_H_HP1-T_con1_out_dh
DELTAT_outlet_sc2_B=T_HP2[7]-T_con2_in_dh

DELTAT_inlet_dsh1_L_A=T_HP1[3]-T_return_dh
DELTAT_outlet_dsh1_L_A=T_C_H_HP1-T_con1_in_dh

DELTAT_inlet_dsh1_L_B=T_C_H_HP1-T_con1_out_dh
DELTAT_outlet_dsh1_L_B=T_HP1[2]-T_con2_in_dh

DELTAT_inlet_dsh2_L_A=T_HP2[3]-T_return_dh
DELTAT_outlet_dsh2_L_A=T_C_H_HP1-T_con1_in_dh

DELTAT_inlet_dsh2_L_B=T_C_H_HP1-T_con1_out_dh
DELTAT_outlet_dsh2_L_B=T_HP2[2]-T_con2_in_dh

DELTAT_inlet_con1=T_HP1[6]-T_con1_in_dh
DELTAT_outlet_con1=T_HP1[7]-T_con1_out_dh

DELTAT_inlet_dsh1_H_A=T_HP1[6]-T_con1_out_dh
DELTAT_outlet_dsh1_H_A=T_C_H_HP2-T_con2_in_dh

DELTAT_inlet_dsh1_H_B=T_C_H_HP2-T_con2_out_dh
DELTAT_outlet_dsh1_H_B=T_HP1[5]-T_forward_dh

DELTAT_inlet_OC1_L=T_oil1_L_HP1-T_con1_out_dh
DELTAT_outlet_OC1_L=T_oil2_L_HP1-T_con2_in_dh

DELTAT_inlet_OC2_L=T_oil1_L_HP2-T_con1_out_dh
DELTAT_outlet_OC2_L=T_oil2_L_HP2-T_con2_in_dh

DELTAT_inlet_OC1_H_A=T_oil1_H_HP1-T_con1_out_dh
DELTAT_outlet_OC1_H_A=T_C_H_HP2-T_con2_in_dh
```

DELTAT_inlet_OC1_H_B=T_C_H_HP2-T_con2_out_dh
DELTAT_outlet_OC1_H_B=T_oil2_H_HP1-T_forward_dh

DELTAT_inlet_OC2_H_A=T_oil1_H_HP2-T_con1_out_dh
DELTAT_outlet_OC2_H_A=T_C_H_HP2-T_con2_in_dh

DELTAT_inlet_OC2_H_B=T_C_H_HP2-T_con2_out_dh
DELTAT_outlet_OC2_H_B=T_oil2_H_HP2-T_forward_dh

DELTAT_inlet_con2=T_HP2[6]-T_con2_in_dh
DELTAT_outlet_con2=T_HP2[7]-T_con2_out_dh

DELTAT_inlet_dsh2_H=T_HP2[6]-T_con2_out_dh
DELTAT_outlet_dsh2_H=T_HP2[5]-T_forward_dh

"!plot script"

"T-Q condenser"

T_districtheating1[1]=T_return_dh
T_districtheating1[2]=T_con1_in_dh
T_districtheating2[2]=T_return_dh
T_districtheating2[3]=T_con1_in_dh
T_districtheating3[3]=T_return_dh
T_districtheating3[4]=T_con1_in_dh
T_districtheating4[4]=T_return_dh
T_districtheating4[5]=T_con1_in_dh

T_districtheating5[5]=T_con1_in_dh
T_districtheating5[6]=T_con1_out_dh

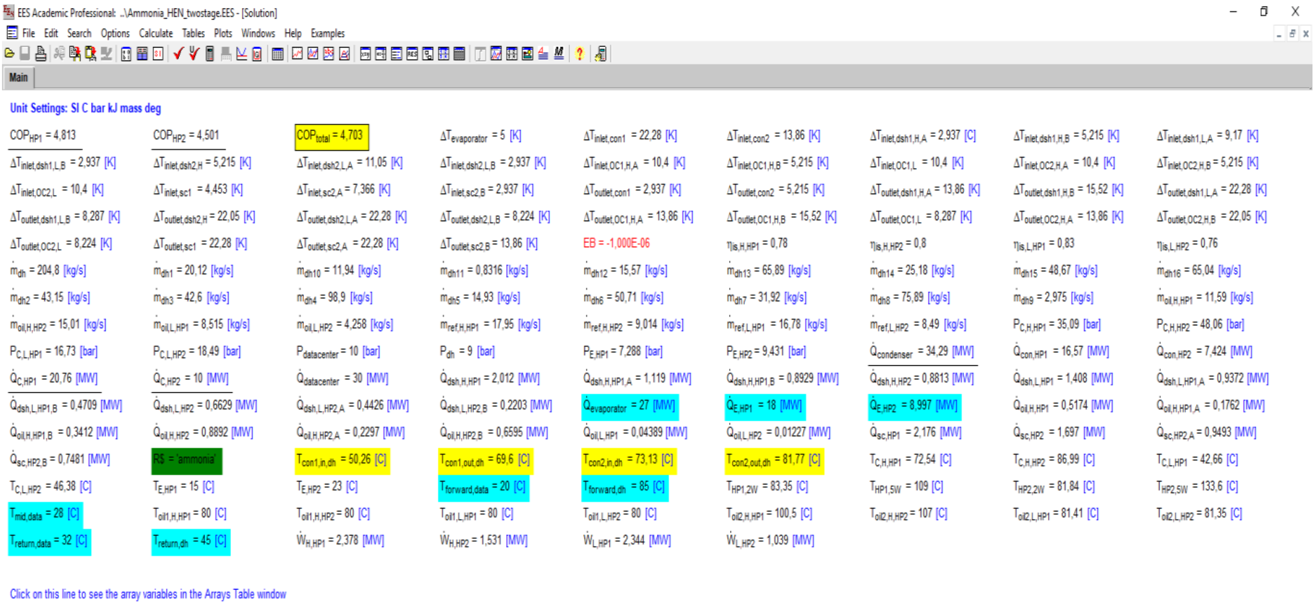
T_districtheating6[6]=T_con1_out_dh
T_districtheating6[7]=T_con2_in_dh
T_districtheating7[7]=T_con1_out_dh
T_districtheating7[8]=T_con2_in_dh
T_districtheating8[8]=T_con1_out_dh
T_districtheating8[9]=T_con2_in_dh
T_districtheating9[9]=T_con1_out_dh
T_districtheating9[10]=T_con2_in_dh
T_districtheating10[10]=T_con1_out_dh
T_districtheating10[11]=T_con2_in_dh
T_districtheating11[11]=T_con1_out_dh
T_districtheating11[12]=T_con2_in_dh
T_districtheating12[12]=T_con1_out_dh
T_districtheating12[13]=T_con2_in_dh
T_districtheating13[13]=T_con1_out_dh
T_districtheating13[14]=T_con2_in_dh

T_districtheating14[14]=T_con2_in_dh
T_districtheating14[15]=T_con2_out_dh

T_districtheating15[15]=T_con2_out_dh
T_districtheating15[16]=T_forward_dh
T_districtheating16[16]=T_con2_out_dh
T_districtheating16[17]=T_forward_dh
T_districtheating17[17]=T_con2_out_dh
T_districtheating17[18]=T_forward_dh
T_districtheating18[18]=T_con2_out_dh

```
T_districtheating18[19]=T_forward_dh


Q_dot_dh[1]=0
Q_dot_dh[2]=Q_dot_dsh_L_HP2_A
Q_dot_dh[3]=Q_dot_dh[2]+Q_dot_sc_HP2_A
Q_dot_dh[4]=Q_dot_dh[3]+Q_dot_dsh_L_HP1_A
Q_dot_dh[5]=Q_dot_dh[4]+Q_dot_sc_HP1
Q_dot_dh[6]=Q_dot_dh[5]+Q_dot_con_HP1
Q_dot_dh[7]=Q_dot_dh[6]+Q_dot_dsh_L_HP2_B
Q_dot_dh[8]=Q_dot_dh[7]+Q_dot_sc_HP2_B
Q_dot_dh[9]=Q_dot_dh[8]+Q_dot_dsh_L_HP1_B
Q_dot_dh[10]=Q_dot_dh[9]+Q_dot_dsh_H_HP1_A
Q_dot_dh[11]=Q_dot_dh[10]+Q_dot_oil_L_HP1
Q_dot_dh[12]=Q_dot_dh[11]+Q_dot_oil_H_HP1_A
Q_dot_dh[13]=Q_dot_dh[12]+Q_dot_oil_L_HP2
Q_dot_dh[14]=Q_dot_dh[13]+Q_dot_oil_H_HP2_A
Q_dot_dh[15]=Q_dot_dh[14]+Q_dot_con_HP2
Q_dot_dh[16]=Q_dot_dh[15]+Q_dot_dsh_H_HP1_B
Q_dot_dh[17]=Q_dot_dh[16]+Q_dot_oil_H_HP1_B
Q_dot_dh[18]=Q_dot_dh[17]+Q_dot_oil_H_HP2_B
Q_dot_dh[19]=Q_dot_dh[18]+Q_dot_dsh_H_HP2

T_dsh_L_HP2_A[1]=T_HP2[3]
T_dsh_L_HP2_A[2]=T_C_H_HP1
T_sc_HP2_A[2]=T_HP2[8]
T_sc_HP2_A[3]=T_C_H_HP1
T_dsh_L_HP1_A[3]=T_HP1[3]
T_dsh_L_HP1_A[4]=T_C_H_HP1
T_sc_HP1[4]=T_HP1[8]
T_sc_HP1[5]=T_C_H_HP1

T_con_HP1[5]=T_C_H_HP1
T_con_HP1[6]=T_C_H_HP1

T_dsh_L_HP2_B[6]=T_C_H_HP1
T_dsh_L_HP2_B[7]=T_HP2[2]
T_sc_HP2_B[7]=T_C_H_HP1
T_sc_HP2_B[8]=T_C_H_HP2
T_dsh_L_HP1_B[8]=T_C_H_HP1
T_dsh_L_HP1_B[9]=T_HP1[2]
T_dsh_H_HP1_A[9]=T_C_H_HP1
T_dsh_H_HP1_A[10]=T_C_H_HP2
T_oil_L_HP1[10]=T_oil1_L_HP1
T_oil_L_HP1[11]=T_oil2_L_HP1
T_oil_H_HP1_A[11]=T_oil1_H_HP1
T_oil_H_HP1_A[12]=T_C_H_HP2
T_oil_L_HP2[12]=T_oil1_L_HP2
T_oil_L_HP2[13]=T_oil2_L_HP2
T_oil_H_HP2_A[13]=T_oil1_H_HP2
T_oil_H_HP2_A[14]=T_C_H_HP2

T_con_HP2[14]=T_C_H_HP2
T_con_HP2[15]=T_C_H_HP2

T_dsh_H_HP1_B[15]=T_C_H_HP2
T_dsh_H_HP1_B[16]=T_HP1[5]
T_oil_H_HP1_B[16]=T_C_H_HP2
T_oil_H_HP1_B[17]=T_oil2_H_HP1
T_oil_H_HP2_B[17]=T_C_H_HP2
```

T_oil_H_HP2_B[18]=T_oil2_H_HP2
T_dsh_H_HP2[18]=T_C_H_HP2
T_dsh_H_HP2[19]=T_HP2[5]



DTU Mechanical Engineering
Section of Thermal Energy
Technical University of Denmark

Nils Koppels Allé, Bld. 403
DK-2800 Kgs. Lyngby
Denmark
Phone (+45) 4525 4131
Fax (+45) 4588 4325

www.mek.dtu.dk