DTU Mechanical Engineering
Department of Mechanical Engineering

# Study of an integrated heating and cooling solution for DTU campus

Nerea Guimarey Docampo

Master's Thesis

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Master Thesis
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# Preface

As a requisite to obtain the Master's degree in Industrial Engineering at the Universitat Politècnica de València, Spain, this document, carrying the load of 30 ECTS, was written. It reports the results of a project developed at the *Danmarks Tekniske Universitet*, Denmark, under the Erasmus+ Programme.

Furthermore, it is hereby declared, under word of honour, that it is original and that all the non-original contributions were properly referenced with source identification.

# Acknowledgements

I would like to thank my supervisors, Brian Elmegaard and Torben Schmidt Ommen (DTU Mechanical Engineering), for giving me the opportunity to carry out this project with them during my exchange. Thanks to their guidance and support during the development of this thesis, especially in the moments of doubt and when I got stuck not knowing very well how to proceed. Thanks also to Emil, his role in the development of this thesis is equally important.

I would like to dedicate this thesis to my family, especially to my parents, whose support I have always had as soon as I decided to start this adventure abroad. Thanks to Ricardo, Ruth and Marcos in particular, you know the support you have given me during this thesis.

A special mention goes to those who have accompanied me during this stay abroad and who have become my friends and family during these 6 months. A0's kitchen has a special memory in my mind, with people who are a very important part of making this an unforgettable experience. No names are needed, you know who you are.

# Abstract

This thesis investigates the possibility of installing a heat pump to produce heating and cooling simultaneously in order to supply part of the annual demand. In the development of this study, the possibility of using a single-stage cycle or a two-stage cycle is analysed, with ammonia being the refrigerant used. Both cycles have the same design input parameters. These include heating and cooling supply and return temperatures, among other technical parameters of thermodynamic cycles. . The analysis of the performance of both cycles is carried out in the EES (Engineering Equation Solver) software, where it is concluded that the two-stage cycle has better results, and will be used for the heat pump in the development of the optimization model.

When the parameters to be used in the heat pump are known, an optimization model is developed using the GAMS (General Algebraic Modeling System) software, where the objective function is to obtain the minimum total annual cost of the production system. A linear model is developed, which results in the optimum capacities of the heat pump and possible storage tanks for subsequent heating and cooling supply. In addition to including the production of heating and cooling by means of the heat pump, there is the possibility of part of both productions being carried out by the production systems currently in operation on the campus. The optimization model uses the results obtained in EES as input parameters in addition to the hourly heating and cooling demands expected to be obtained during a year on campus. In addition to these values, other factors have been taken into account, such as investment and production costs.

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# List of Abbreviations and Symbols

## Greek letters

| | |
|---|---|
| $\rho$ | Density |

## Indexes

| | |
|---|---|
| $t$ | Time |
| $r$ | Return |
| $s$ | Supply |
| $s$ | Supply |
| $elec$ | Electricity |
| $OM$ | Operations and Maintenance |
| $Comp$ | Compressors of the current system |

## Acronyms

| | |
|---|---|
| CHP | Combined Heat and Power |
| cond | Condenser |
| evap | Evaporator |
| sc | Subcooling |
| dsh | Desuperheating |
| c | Condensation |
| e | Evaporation |
| sh | Superheating |
| DH | District Heating |
| DC | District Cooling |
| HP | Heat Pump |
| RES | Renewable Energy Sources |
| COP | Coefficient of Performance |
| VCC | Vapour Compression Cycle |

# Chapter 1

# Introduction

## 1.1 Literature review

As part of the fight against climate change, the European Union sets standards that member countries must meet. These standards are announced in the European Green Deal and each member country publish its owns in the National Energy and Climate Plan (NECP), based on the Green Deal. For Denmark, this plan was published on December 2019, and it has important objectives for the future. The main aim is to reduce the greenhouse gases by 70 % by 2030, compared to 1990 levels, and to reach net zero emissions by 2050, as a member of the Green Deal of the UE [1]. Other objectives are also to ensure a 55 % share of renewable energy in the final consumption by 2030 and at least a 90 % of the District Heating (DH) consumption produce by other energy sources than coal, oil or gas by 2030 [2]. Currently, DH represents up to 64 % of the total heating in the country, providing heating in a large-scale production, where up to 61 % of it it is produced by green energy [3]. In order to achieve a high percentage of district heating production based on green energy, heat pumps play an important role.

Heat pumps (HP) use energy, which can be electricity, to transfer heat from a low temperature heat source to a high temperature heat sink; in this way, they can use the power generated from the renewable energy sources (RES). This would be beneficial in a future scenario with a high share of renewable energies in the electricity grid, where the HP can use the excess electricity for heating production purposes.

With the new focus on the 4th Generation District Heating, with lower flow temperatures, the heat pumps increases its efficiency. The 4GDH is based in temperatures around 50 °C for supply and 25 °C for return flows, which makes the heat losses reduce and a wide possibility of low temperature heat sources [4].

The possibility to unify the heating and cooling production is also an advantage, since it can be achieved a very high efficiency for both systems, and it would not be the first time to carry out a project like this. The largest heating and cooling plant produced by a heat pump is located in Helsinki [5]. This shows the combination of DH and DC is possible and it also beneficial for the potential emissions and energy savings it could be achieved.

## 1.2 Problem statement

The main aim of this thesis is to demonstrate the feasibility of installing a vapour compression heat pump and the integrated supply of district heating and district cooling for the DTU Campus. This will be carry out with the development of a thermodynamic model.

The approach on this thesis will be focus on the development of a model with EES that can integrate the heating and cooling demand on the DTU Campus. The assumptions to develop the thermodynamic model will be based on the available data for the different plants that provide heating and cooling to DTU campus. The current demands and supply temperatures will be taken into account. If there is enough data from the current plants during a certain periods of time, the model will be verify for the heating and cooling season and its working will be analyse.

# Chapter 2

# Basic principles

This chapter describes the basic principles and operation that should be known to understand the background of the project.

## 2.1 Heat pumps

A heat pump (HP) is a device that extracts heat from a low temperature heat source (T1), and transfers it to a heat sink at a higher temperature (T2). This transfer needs to accomplish the second law of thermodynamics: *No process is possible whose sole result is the transfer of heat from a cooler to a hotter body* [6]. This heat transfer would not be possible without using a external power that is supplied to the system. This external power is usually electricity that runs a compressor. The basic sketch from figure 2.1 follows the first law of thermodynamics, where, the heat rejected to the heat sink, $Q_2$, is equal to the heat extracted from the heat source, $Q_1$ plus the necessary work to operate the cycle, $W_{in}$ (equation 2.1).

$$Q_2 = Q_1 + W_{\text{in}} \tag{2.1}$$

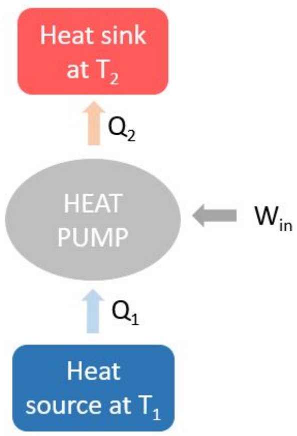



Fig. 2.1: Working principle of a heat pump

The use of HP for district heating makes low temperature sources suitable for heating purposes, considering that increases the temperature of a waste heat flow or a heat source to an useful high temperature heat. Although it is usually used for the purpose of generating heating and/or domestic hot water, there is the possibility of using a heat pump to generate heating and cooling simultaneously. In this project, this is the configuration that will be developed.

The efficiency of this system is evaluated through the coefficient of performance (COP), both when heating or cooling are produced. The maximum COP of a machine that transfers heat from one focus point to another is measured by the Carnot cycle. It only depends on the evaporation and condensation temperatures of the cycle. In the equation 2.2 is shown the maximum COP for heating and cooling purposes for the heat pump of the figure 2.1. It can be proved that, with a smaller temperature lift between the heat source and the heat sink, the COP of the system is higher ($T_2$-$T_1$).

$$COP_{\text{max, H}} = \frac{T_2}{T_2 - T_1} \qquad COP_{\text{max, C}} = \frac{T_1}{T_2 - T_1} \tag{2.2}$$

The COP of the real system will be lower than the one obtained by the Carnot cycle due to irreversibilities that happens in the real cycle. When measuring the efficiency of heat production, the COP is obtained as the ratio of the thermal energy delivered to the heat sink and the energy consumed. For cooling, the calculation is similar but using the heat extracted from the heat source instead.

$$COP_{\text{H}} = \frac{\dot{Q}_2}{\dot{W}} \qquad COP_{\text{C}} = \frac{\dot{Q}_1}{\dot{W}} \tag{2.3}$$

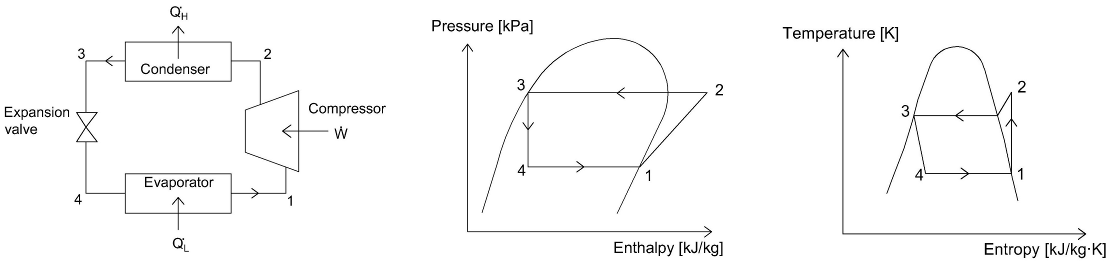
Getting a high value of the COP in a heat pump means a lower energy consumption for the same heating or cooling energy delivered than a heat pump with lower COP.

The performing principle of a heat pump relies on the vapour compression cycle (VCC). This cycle is based on the compression and expansion of the heat transfer fluid, called refrigerant, which goes through all components of a HP: evaporator, compressor, condenser and expansion valve.

## 2.2 Vapour compression cycle

The vapour compression cycle (VCC) is main cycle used for refrigeration purposes. The refrigerant goes through different phases that take place in the four components of the HP. In this cycle, the external energy used to make the heat transfer is electricity. The figure 2.2 shows the sketches of the principal components of the cycle and the P-h and T-s diagrams.

In the ideal cycle, the process that the refrigerant performs is the following.




(a) Components of a VCC (b) P-h ideal diagram (c) T-s ideal diagram

Fig. 2.2: The vapour compression cycle

- Compressor: the refrigerant enters the compressor as saturated vapour. In the compressor its pressure, and consequently its temperature, rises, reaching high values. All this takes place at constant entropy.

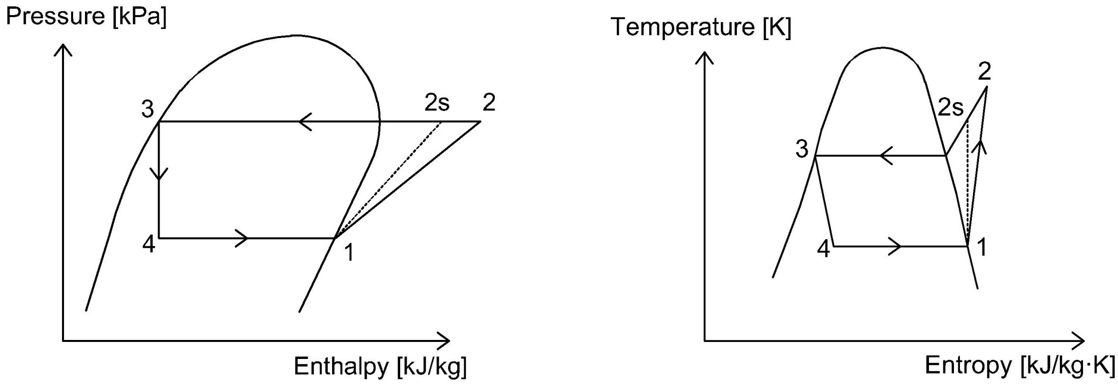
- Condenser: after leaving the compressor, the refrigerant, which is in the superheated vapour state, passes through the condenser, where it exchanges the heat it has just gained with another fluid, warming it up, until the refrigerant reaches the saturated liquid state. The process occurs at constant pressure.

- Expansion valve: after this, it returns to the initial pressure through the expansion valve, which reduces the pressure at constant enthalpy.

- Evaporator: the cycle continues in the evaporator, where the refrigerant absorbs heat from the heat source, heating up at constant temperature until it reaches the saturated vapour state, closing the cycle.

In the real cycle, the process through the compressor does not happen at constant entropy. This makes the temperature at the exit of it higher than in the ideal cycle. The performance of the compressor is evaluated by its isentropic efficiency. This indicator expresses the work that would be done in the ideal process respect to the actual work that the compressor must do, being this second one greater than the first one. The figure 2.3 shows the real diagrams for the VCC of P-h and T-s.

$$\eta_{\text{is}} = \frac{\text{isentropic work}}{\text{actual work}} = \frac{(h_{2s} - h_1)}{(h_2 - h_1)} \tag{2.4}$$




(a) P-h diagram (b) T-s diagram

Fig. 2.3: The real vapour compression cycle

# 2.3 District heating and district cooling

## 2.3.1 District heating

District heating is an energy service based on transfer heat from available heat sources to consumers that demand heating. District heating is a efficient method to supply heat for space heating and domestic hot water. The heat is distributed by a network of subterranean pipes that supplies the demand heat at any moment. The principal form of generating the heat is by using heat sources that would be wasted if they are not used for this purpose. The most common heat sources with which district heating works are the excess heat from industrial processes, the heat produced in waste incineration plants, excess heat from thermal power stations (known as Combined Heat and Power (CHP)) between others.

There are many designs that have been used in the development of the district heating technology. For heating, the heat transfer medium can be either steam or hot water. For district cooling the distribution medium is chilled water. Throughout its history, the main focus of the development of the heat distribution technologies is to reduction cost of several factors:

- Investment costs
- Space demand
- Installation time
- Running costs

District heating technology has been improved over the years, resulting in different generations that differentiates from the previous one by the improvements made.

### 2.3.1.1 Generations

In the history of the district heating it can be distinguish five generations of district heating.

**1st generation**

The fist generation of district heating was based on steam as the heat distribution medium. The distribution pipes where located in concrete ducts, with one large pipe which administrated the steam, an another smaller pipe for the condensate return. The main problem with the return pipe was the corrosion, being the the disconnection from the system the common solution. The first systems were introduced in USA in the 1880s. Due to the high temperatures of those systems, there were significant heat losses and accidents related to the high pressure pipes, which leaded to a very low system efficiency. Currently, it is considered an outdated technology [4].

**2nd generation**

The heat carrier in the second generation is pressurised hot water with temperatures usually over 100°C. The system consisted in two steel pipes, one for the flow and the other for the return, with similar size and both located in concrete ducts, as the fists generation. The advance in the ducts respect the previous generation was the reduction of the size and more compact ducts were developed. The pipes were insulated on site, usually with wool insulation. The heat losses were high over time due to the loss of the insulation properties when the insulation began to be damaged. However, the efficiency of the system was higher than in the previous generation due to the lower operating temperatures. This systems appeared in the 1930s [4].

**3rd generation**

The heat carrier is still pressurized hot water. The difference of this generation with the previous one lays in the fact that the temperature of the water is below 100°C. These systems are composed by prefabricated and pre-insulated pipes buried directly into the ground (with no ducts) with prefabricated compact substations. This generation focus on lower construction cost and energy efficiency (reducing the operative temperatures). The security of the supply (due to the oil crisis in the 70s) became one of its aims, and with the reduction of the temperatures, other heat sources besides oil were introduced into the system, as combined heat and power (CHP), biomass or waste incineration plants. The reduction of the temperatures made possible to improve in savings in thermal distribution losses. One of the most important improvements of the system are the systems' metering and monitoring, increasing their efficiency and performing. This generation is called the "Scandinavian" district heating technology because many of the first district heating component manufacturers were based in Scandinavia. This generation started to be implemented in the 1970s and they are still on operation [4].

**4th generation**

With the 4th generation still in development, it will be the generation of the energy efficiency and the renewable energies. Supply low-temperature district heating

Transforming the system towards a more sustainable one, with a high share of fluctuating renewable sources. Reducing the building energy consumption as well as the temperatures of the system, which leads to a higher efficiency but also a flexibility on the district heating system towards potential new heat sources. Large heat storage is introduced to compensate the intermediate availability of renewable sources and the increment of share renewable energy for heat and electricity [4].

#### 5th generation

The challenge of the 5th generation of district heating is to be able to provide both heating and cooling services simultaneously to different buildings in the same pipes. The distribution temperatures are close to ambient ground temperature in order to minimize the heat loses [7].

### 2.3.2 District cooling

District Cooling is a distributed system, that works in a similar way than DH. It supplies cooling by demand through a network of pipes, delivering cold water. The purpose of the cooling can be either commercial or private. With the increasing demand for refrigeration in many cities around the world, replacing A/C by DC is able to achieve much greater efficiency in the process of generating and delivering cold using DC than if traditional air conditioning systems were used.

By using DC in companies and spaces with high cold demand, it can be reach a cost saving of up to 40% and reduce the CO2 emissions by up to 70% compared to the use of traditional A/C systems [8], making the DC more climate-friendly than traditional systems.

The method of using DC is different in summer and in winter. In the winter months, most DC plants use cold water from a nearby source, such as a lake or the sea, to produce the necessary cooling in a process that results in low CO2 emissions. Whereas, in summer, the temperature of the source is not cold enough to produce the necessary cold directly. During this months, the use of external energy is necessary to cool down the water from the source.

# Chapter 3

# Data analysis

This chapter describes the analysis carried out with the operating data of the different heating and cooling plants on the campus of the DTU in Lyngby.

## 3.1 Data compiled

The analysis prepared was based on a previous analysis carried out for the period of operation between 23rd January 2017 and 23rd January 2018. This analysis was carried out and provided by Emil Dybro Korsgaard Jacobsen. To obtain more recent data on the operation of heating and cooling plants, the website «bmsnet.cas.dtu.dk» is used, where different measures necessary for the analysis are recorded among many others. In addition, the DTU's climate station is used for outside ambient temperature measurements, which are recorded and accessible via the web page «climatestationdata.byg.dtu.dk».

The data used in the analysis are energy measurements from the different heating and cooling plants. On the Lyngby-DTU campus, there is a combined heat and power plant (CHP), which supplies the necessary heat to the entire campus. To supply the necessary cooling, there are four plants with different power capacities.

- Heating plant B415 (Unknown capacity)
- Cooling plant O114 (Capacity: 2MW)
- Cooling plant B415 + Free Cooling (Capacity: 2MW / FC: 0,4MW)
- Cooling plant B349K (Capacity: 1 MW)
- Cooling plant B346k + Free Cooling (Capacity: 0,6MW / FC: 0,56MW)

It is known that the actual cooling capacity of the campus is not enough to supply all the cooling demand in the hot season. This is why 2MW more will be installed in May 2020, in addition to a further 3MW planned.

### 3.1.1 Previous analysis

In the study conducted by Emil, a period of time from January 23, 2017 to January 23, 2018 is analyzed.

#### 3.1.1.1 Heating Plant

In the case of the heating plant (B415), there is data collected over the entire period analysed. Thus, a correlation is obtained that relates the demand for heat on campus with the outside ambient temperature. The result can be observed in figure 3.1.

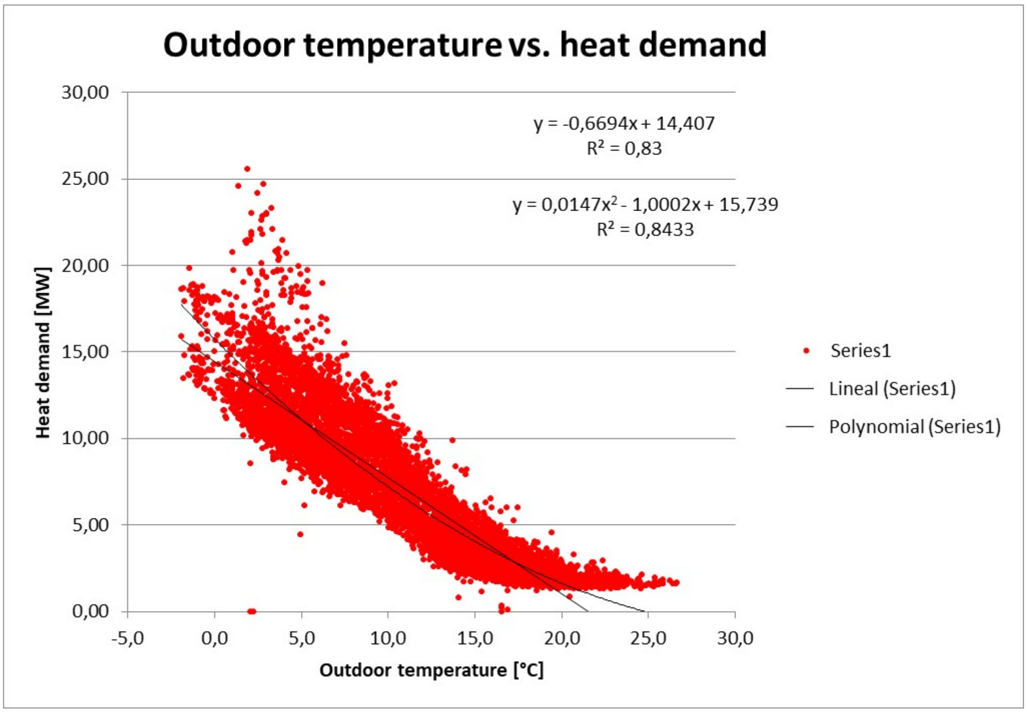



Fig. 3.1: Correlation between the outside ambient temperature and heat demand

In the figure it can be seen that the correlations are not valid for heat demands less than 1,5 MW, which is the minimum base load to maintain the hot temperature in several water tanks. The heat demand correlation based on the outside ambient temperature is made up of piecewise linear functions.

#### 3.1.1.2 Cooling Plants

The case of refrigeration plants is more complicated, as there is a lack of data for some time of the period represented. The cooling demand is composed of the sum of the outputs of all the cooling plants listed above. As the total demand depends on the data of more than one plant, it is more difficult to obtain the necessary data for the whole period analysed, and this is what happens.

For the refrigeration plant B415-3 (the compressor number 3), there is a data gap from May 8, 2017 to September 11, 2017. This gap is filled based on the data available before and after the gap as well as the knowledge of the operation of the B415 plant in pairs. Furthermore, while in the case of heating demand, correlations were obtained from data between 23 January 2017 and 23 January 2018, in the case of cooling demand correlations are obtained from data between 8 May 2017 and 23 January 2018, a shorter time period. This is due to the lack of data or records in one of the other refrigeration plants that supply the total demand.

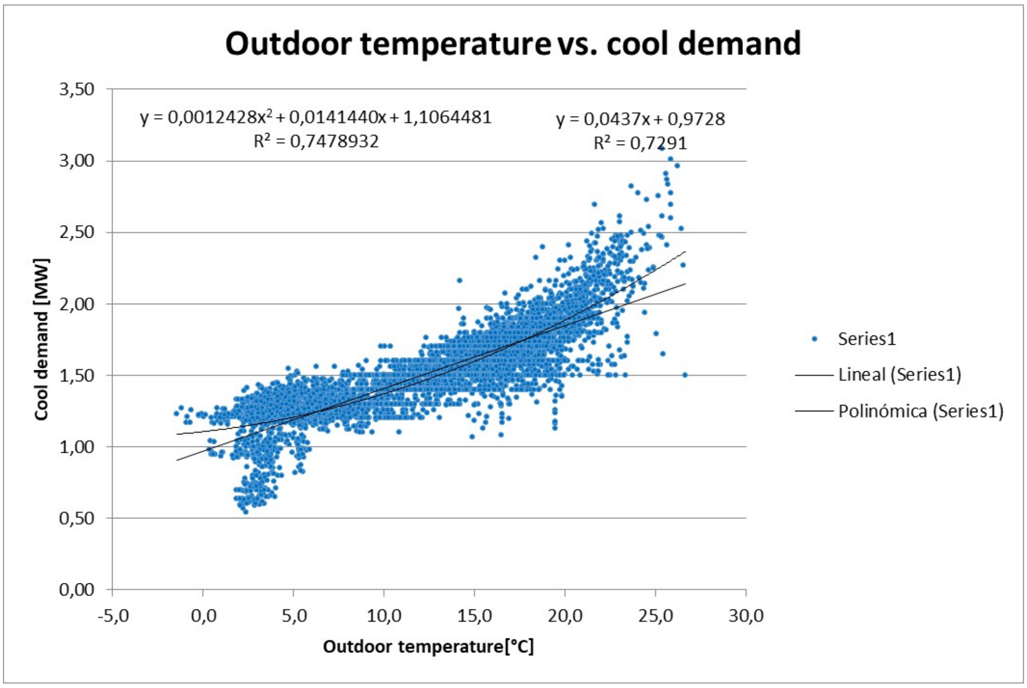



Fig. 3.2: Correlation between the outside ambient temperature and cool demand

The correlations of the cooling demand are composed of piecewise functions, one linear and one polynomial, depending on the outside ambient temperature, as it can be seen on figure 3.2.

#### 3.1.1.3 Results

In order to be able to represent heating and cooling consumption in relation to the outside temperature for the period of a full year (23 January 2017 to 23 January 2018), the gap in cooling consumption is filled with the correlation obtained from the analysis of the data. In this way, the curve shown in figure 3.3 is obtained.

If you compare the figure 3.3, the data from the actual heating and cooling consumption records, with figure 3.4, the data obtained from the calculated correlations, the difference between the two representations is not very large. The biggest difference you can see is that the

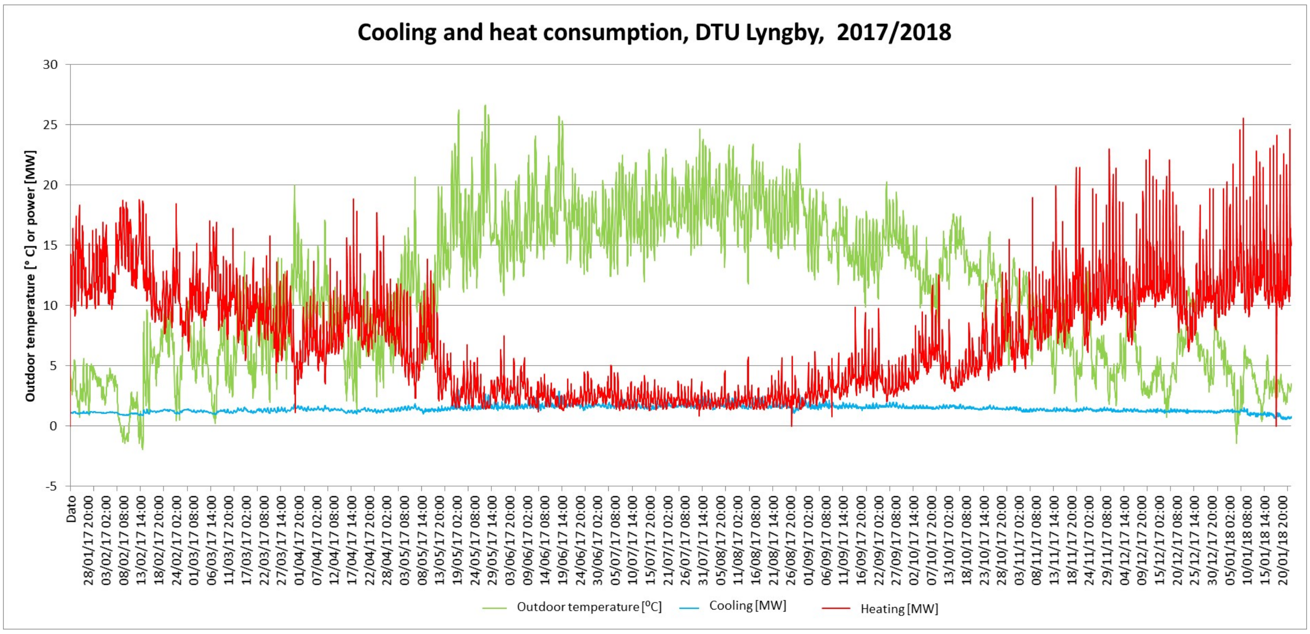



Fig. 3.3: Representation of heating and cooling consumption together with the outside temperature 2017-2018

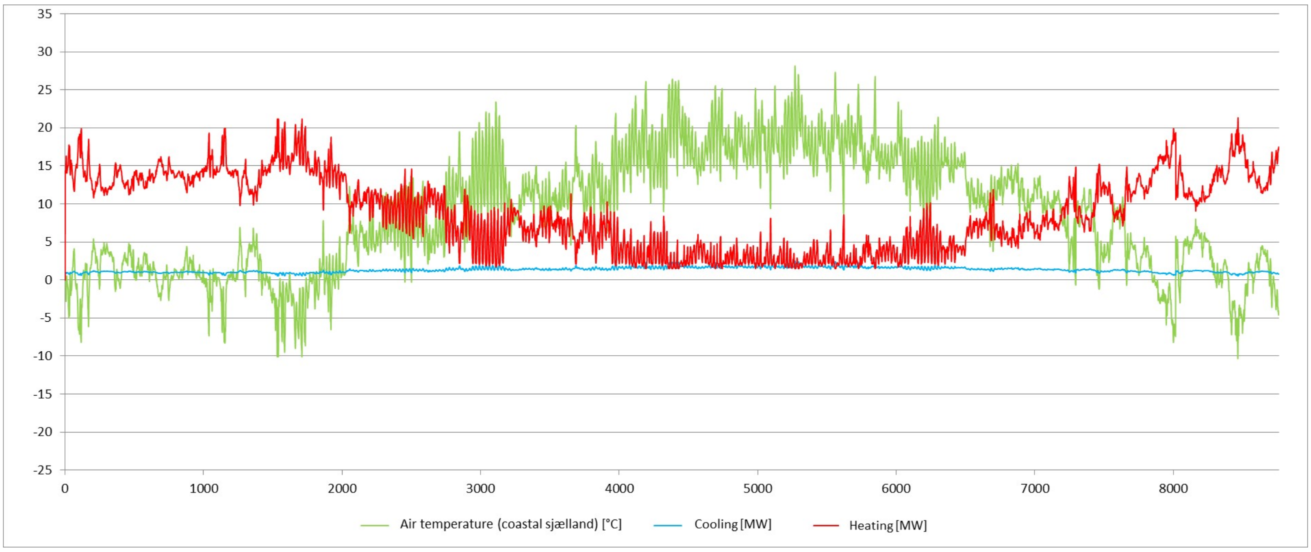



Fig. 3.4: Representation of the resulting correlations of heating and cooling with the outside temperature

correlations do not take into account the higher consumption peaks, but instead reflect the average consumption for cooling and heating.

In addition, it is necessary and significant to mention that since this study of heating and cooling consumption on the DTU-Lyngby campus was conducted, cooling consumption has doubled (at the cooling station). That is why more cooling capacity needs to be installed.

### 3.1.2 Latest Analysis

After studying the analysis in the period 2017/2018, a study of more recent data from the same plants is carried out in order to visualise any changes in the behaviour of heating and cooling demands. In order to carry out this study in greater depth, we have chosen to combine the data from the previous analysis carried out by Emil and the most recent data available. In this way, theoretically, it would be possible to visualize, among other aspects, the increase in the cooling demand (consumption) over time.

#### 3.1.2.1 Heating Plant

In the case of the heating plant there is a large gap without data collected by the system. From the last measurement provided by Emil's study, to the next measurement recorded by the system there is a difference of more than a year. Specifically, the gap covers from 23 January 2018 to 27 November 2019, making it difficult to observe any variation in heating consumption,as shown in figure 3.5.

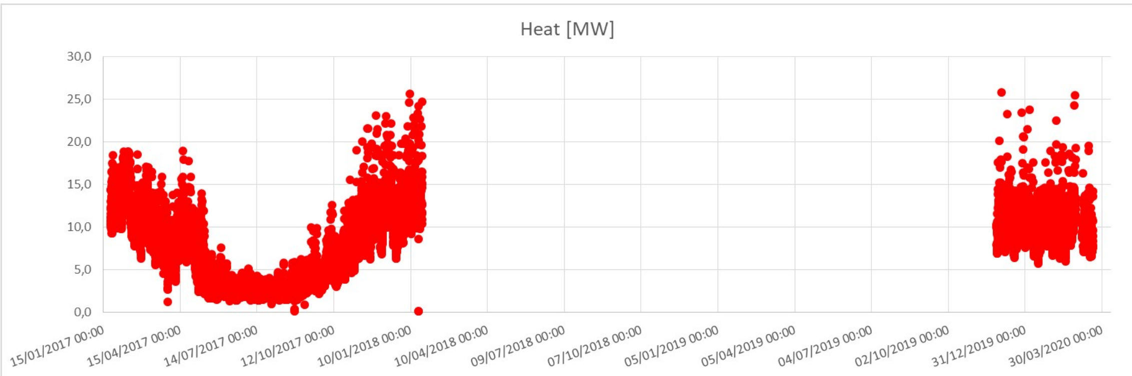


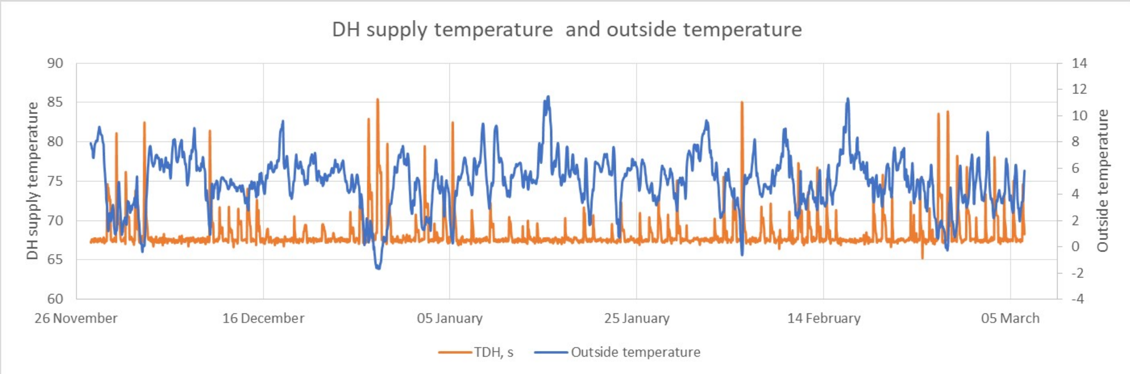

Fig. 3.5: Representation of the available points of the heating plant B415

The most recent data available on heating consumption covers the period from 27 November 2019 to 6 April 2020, with data more recent than that also available.

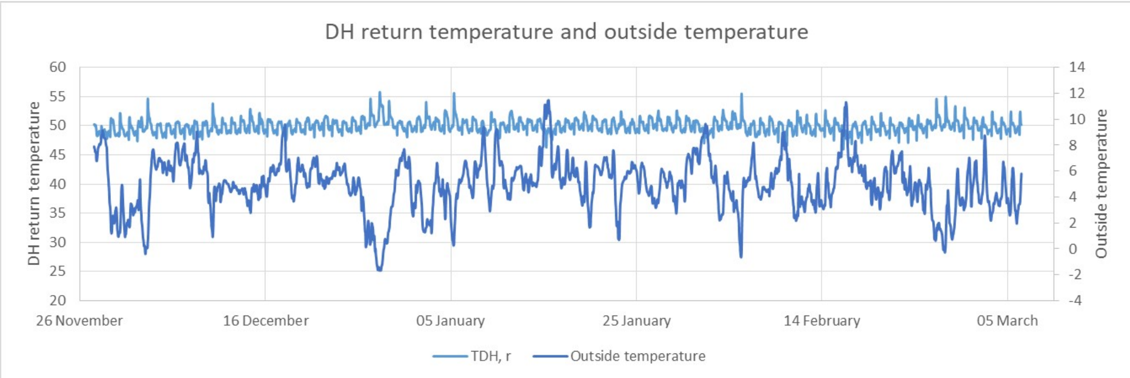
#### Heating temperatures

Within the study of the heating plant, an analysis of the supply and return temperatures are also evaluated.

The evolution of the supply and return temperatures for the DH can be observed in figures 3.6a and 3.6b through several months. For the DH supply temperature there is a minimum, around 68 - 70°C. This is the minimum temperature at which DH is supplied on campus whenever the outside temperature is high. It can be seen that when the outside temperature decreases, there are temperature peaks for DH supply, reaching values of 85°C. Whenever there is a minimum peak for the outside temperature, the DH temperature coincides with a maximum peak, with intermediate peaks occurring when the outside temperature reaches intermediate values. On the other hand, the values for the DH return temperature remains remarkably constant at around 50°C, independent of the outside temperature.




(a) DH supply temperature vs outside temperature




(b) DH return temperature vs outside temperature

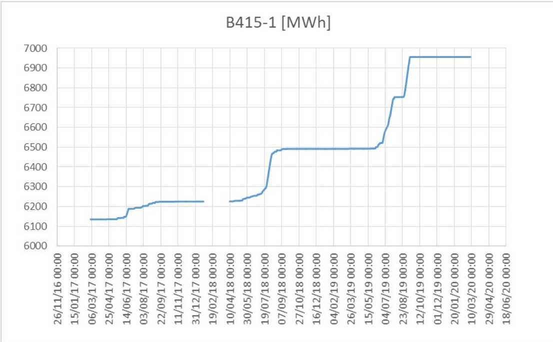
Fig. 3.6: Analysis of DH supply and return temperatures with the outside temperature

In summary, it can be said that the variation of the DH supply temperature is highly influenced by the external temperature experienced at any given time, varying from a minimum of around 70°C to a maximum of around 85°C, while on the other hand, the DH return temperature remains practically unchanged in the 50°C.

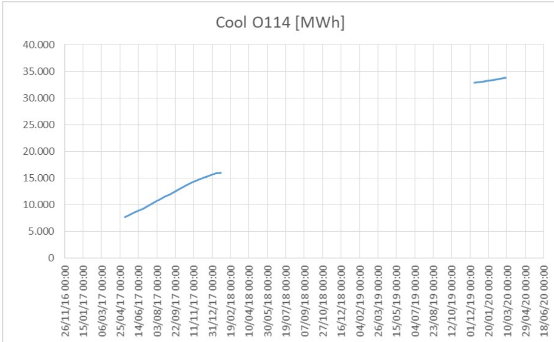
#### 3.1.2.2 Cooling Plants

There are different data gaps in different refrigeration plants. Overall, the period of time where data is available from all the different refrigeration plants is very limited, even less than the period available for the heating plant. Although in some of the refrigeration plants the data set available is quite large, in others the points recorded are scarce. As the total heating demand is not supplied by a single plant, but the sum of all of them, it is complicated to know the total cooling consumption without having data from all plants.

The different sizes of the lack of data in the different refrigeration plants can be seen in figure 3.7. While in figure 3.7a it can be seen that the gap is small,from 23 January 2018 to 9 April 2018, and the unavailable data could be estimated, in figure 3.7b, the gap is too large to estimate a trend in consumption that does not move away from what could be the actual consumption of that plant.




(a) Cooling plant B415 (compressor 1)




(b) Cooling plant O114

Fig. 3.7: Representations of the available points for two of the cooling plants

If we add up the consumption of all the plants to observe the demand for cooling that exists on the campus and represent only those points where there is data from all the plants, the result can be seen in figure 3.8. The gap in recorded measurements covers from January 23, 2018 (last day studied in Emil's analysis) to December 12, 2019 (first available measurement of plant O114).

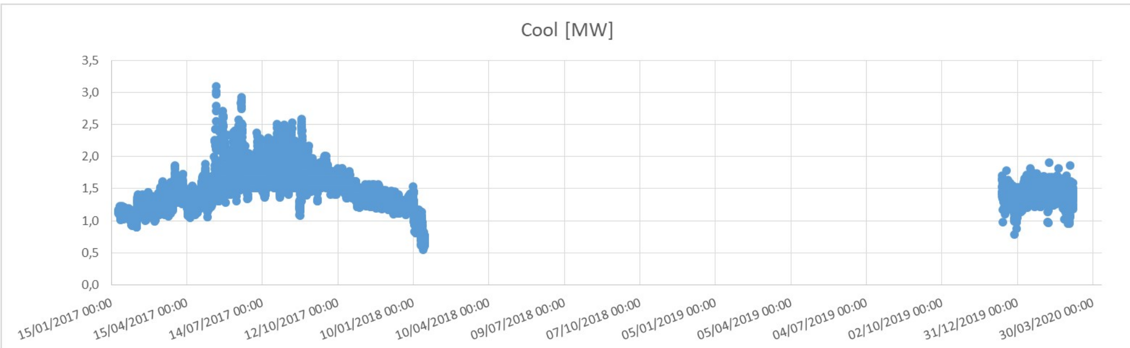



Fig. 3.8: Representation of consumption data available for all refrigeration plants

One of the most important aspects that should be visible in Figure 3.8, is that the increase in cooling consumption during cooling seasons should be seen. Because the gap covers the two cooling stations in 2018 and 2019 in particular, this increase cannot be seen in the available data.

### Cooling temperatures

The same analysis of the cooling supply and return temperatures has been attempted. In this case, it has not been possible to carry out such a study. Among the causes is that the cooling data on campus is for a short period and not during the summer months. Therefore, the campus cooling plants do not provide useful data to work with.

#### 3.1.2.3 Results

If the correlations for heating and cooling consumption are recalculated by adding the most recent data available to the data from the study carried out by Emil, the results of consumption as a function of the outside temperature are not very different. This is basically because the number of data added is not very large, so the result of the functions obtained in the correlations is not very different from those obtained in Emil's analysis. The result of these new correlations in the period from 23 January 2017 to 18 March 2020 can be seen in Figure 3.9.

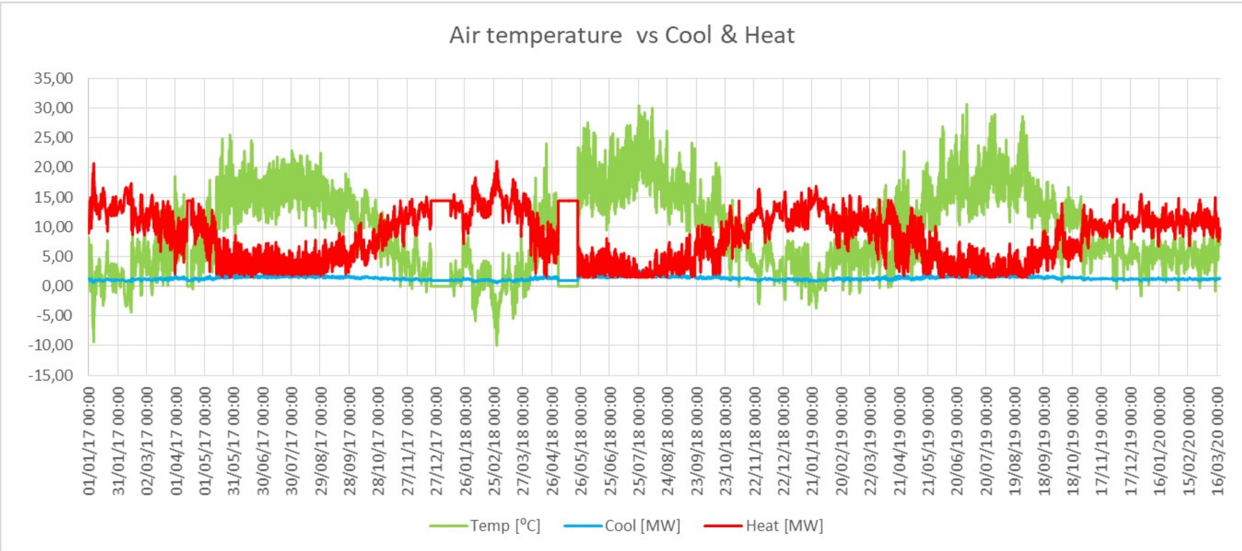



Fig. 3.9: Representation of heating and cooling consumption together with the outside temperature 2017-2020

In addition, the cooling consumption is shown in a separate figure, which is estimated with the correlations as a function of the outside temperature in Figure 3.10. In this figure it can be seen that there is no increase in cooling consumption. The reason for this may be the combination of two aspects. Firstly, the correlations are calculated with data that practically do not take into account more recent consumption. In addition, the correlations are characterized by representing an average load, without taking into account possible consumption peaks.

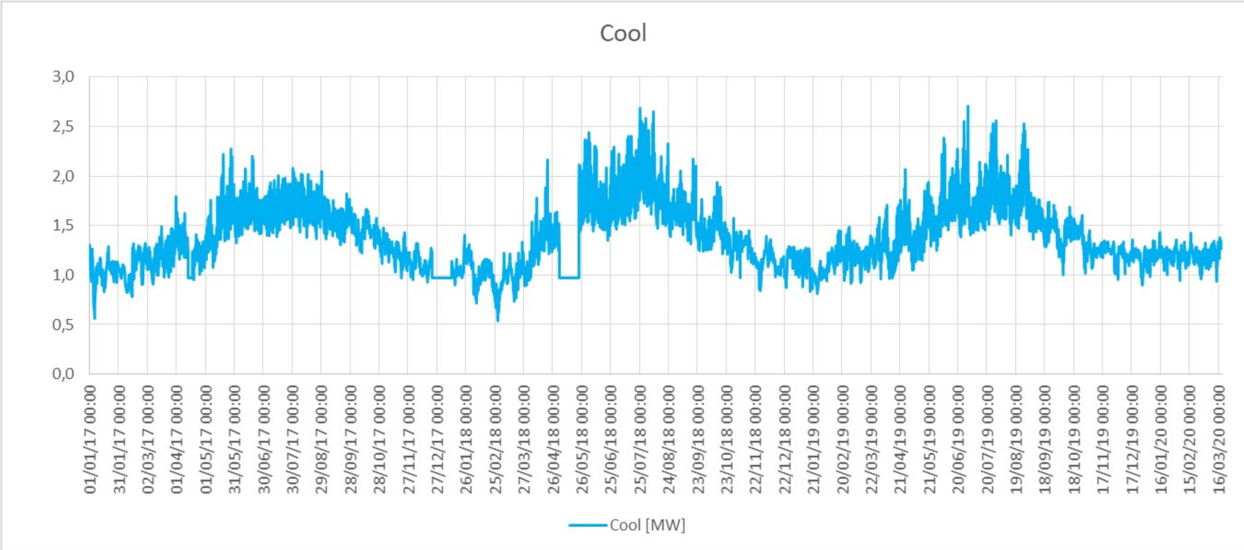



Fig. 3.10: Representation of cooling consumption 2017-2020

## 3.2 Conclusions

After analyzing the available data on the heating and cooling consumption of the different plants on the DTU-Lyngby campus, it can be concluded that it is not reliable to base the design of the integrated heating and cooling solution on them. In order to do so, a greater number of records should be available, so that the result would be more reliable.

# Chapter 4

# Methodology for thermodynamic modelling

This chapter describes the thermodynamic model used for the co-production of heating and cooling. As we speak of co-production, both heating and cooling outputs are supplied simultaneously, using the district cooling return flow as the heat source for the heat pump. As long as no storage tanks are installed in the supply system, both productions need to be consumed at the same time as they are produced. This will be studied in chapter 6.

The aim of developing the thermodynamic model and studying its results is to create a cost optimisation model as close to reality as possible. In this way it is possible to study different cycles and characteristics of the heat pump that can improve its performance, obtaining the best possible operating parameters. This is important because it influences the outcome of the cost optimisation model of the heating and cooling supply systems.

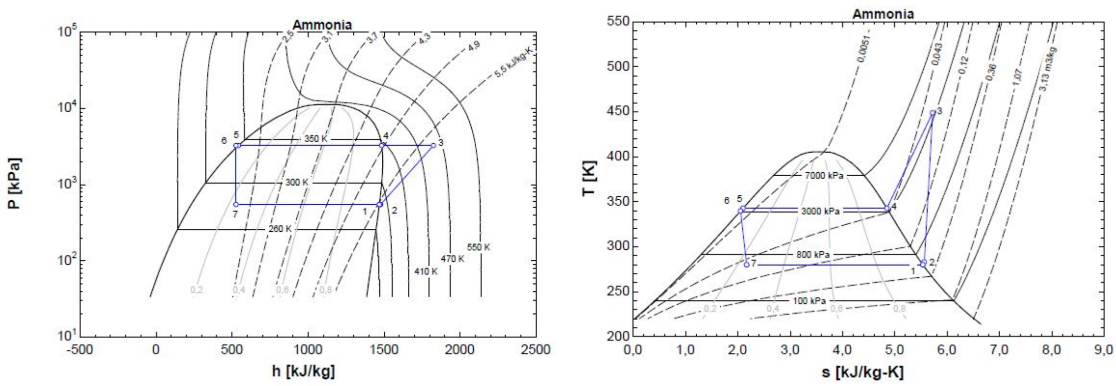
## 4.1 Thermodynamic cycle

The vapour compression cycle (VCC) is one of the most widely used cycles in heat pumps for both heating and cooling purposes. Its basic operating principle has been explained broadly in section 2.2, and the characteristics of the different cycles used in this study will be explained in detail below.

### 4.1.1 One-stage cycle

It is the simplest VCC that can be obtained in a heat pump. The cycle consists of an evaporator, a compressor, a condenser and an expansion valve. The one-stage cycle has a total of 7 different operating points which are represented by the P-h and T-s diagrams in figure 4.1.

The use of this type of cycle is limited by the pressure step that the compressor can withstand. For a large difference between the evaporation temperature and the condensing temperature, it is often necessary to add an intermediate stage, which results in the two-stage cycle.




(a) P-h diagram (b) T-s diagram

Fig. 4.1: Thermodynamic diagrams from an one-stage cycle

### 4.1.2 Two-stage cycle

Two-stage cycles are used to solve the problems that arise in the simple vapour compression cycle associated with a high pressure ratio. Some of these problems are lower isentropic efficiency in the compressor, as this is related to the pressure ratio, high compressor discharge temperature and higher energy consumption. There are two main solutions for a two-stage heat pump: a cascade cycle or a multi-stage cycle. The latter is used in this study. This type of cycle consists of using two or more compressors in series and a flash intercooler in the intermediate pressure. In the two-stage cycle, by adding more components, the number of resulting points is higher, with a result of 11 different thermodynamic points. All of them are represented in the P-h and T-s diagrams for a two-stage vapour compression cycle, which can be seen in figure 4.3. The intermediate pressure on a two-stage cycle is usually selected as the equation 4.1 shows.

$$p_{\text{int}} = \sqrt{p_{\text{high}} p_{\text{low}}} \tag{4.1}$$

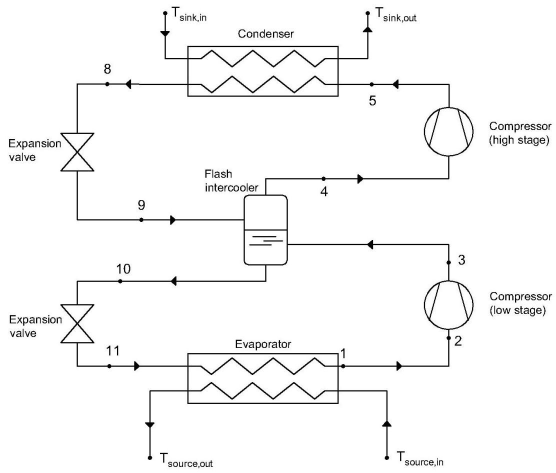
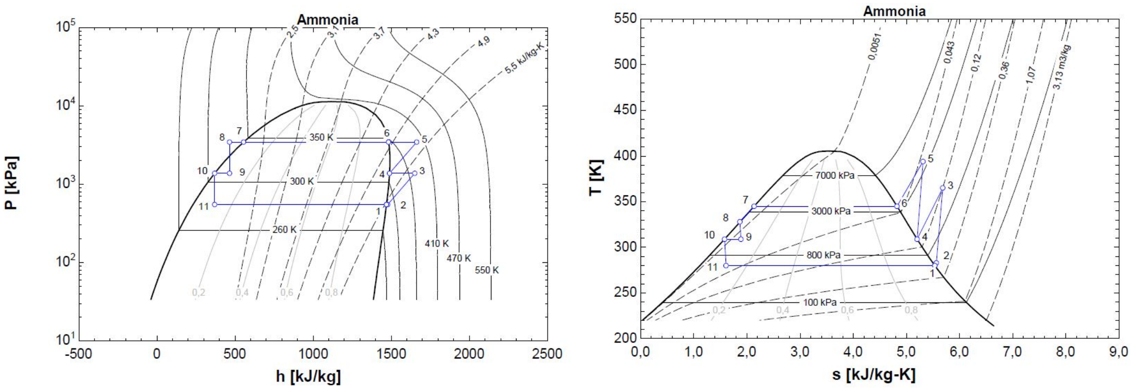



Fig. 4.2: Sketch of a two-stage cycle




(a) P-h diagram (b) T-s diagram

Fig. 4.3: Thermodynamic diagrams from a two-stage cycle

## 4.2 Components

As explained in section 2.2, the refrigerant goes through different stages in the vapour compression cycle. Each of these stages occurs in each of the components that constitute the heat pump. These elements connected in series compound the cycle. The main elements that constitute the heat pump are explained below in order to obtain the thermodynamic characteristics of each of the points of both one and two stage cycles.

### 4.2.1 Refrigerant

The refrigerant is the working fluid that undergoes the different changes in each of the stages of the heat pump. In order for a refrigerant to be used in a particular vapour compression cycle, it must meet a series of requirements both at the thermodynamic level and at the level of safety and contamination.

If a summary is made of the most important properties that a refrigerant must meet in order to be suitable for its use, some of them are as follows:

- Chemically stable and inert
- Critical point appropriate for the application
- High thermal conductivity
- Low freezing point
- Non-toxic, non-flammable, non-corrosive
- Environmentally friendly
- Easy leak detection
- Low cost

It is difficult for a refrigerant to meet all the necessary requirements, and the different types of refrigerants used from the first cooling cycles to the present day corroborate this fact. Due to the increasingly demanding requirements in terms of the damage to the environment from used refrigerants, some types of refrigerants have been eliminated from current refrigeration systems or are close to being eliminated. The current legislation limits two properties of refrigerants.

- Ozone Depletion Potential (ODP): represents the effect it has on the ozone layer. The range of values is from 0 to 1.
- Greenhouse Warming Potential (GWP): compares the climate impact of its emissions to the atmosphere with the emissions of the same amount of $CO_2$, considered the most harmful greenhouse gas. The range of values is from 0 to thousands.

Due to this restrictions, the natural refrigerants as ammonia became more interesting again. This is the refrigerant that the cycle in this project uses. Ammonia has no ozone depletion potential (ODP = 0) or greenhouse warming potential (GWP = 0) and has no expensive costs. Although it is toxic in very low concentrations, leak detection is effective as it has a characteristic odour even at lower concentrations than the harmful ones, which acts as an alarm signal for possible leaks. [9] [10]

### 4.2.2 Evaporator

The evaporator type selected for the cycle is a flooded evaporator, commonly used in ammonia cycles. In this type of evaporators a receiver is located at the inlet. This receiver separates the refrigerant leaving the isoenthalpic expansion valve into saturated liquid (x=0) and saturated vapour (x=1), ensuring that only liquid refrigerant enters the evaporator inlet, making the heat exchange more efficient, and only vapour enters to the compressor. At the outlet, the refrigerant

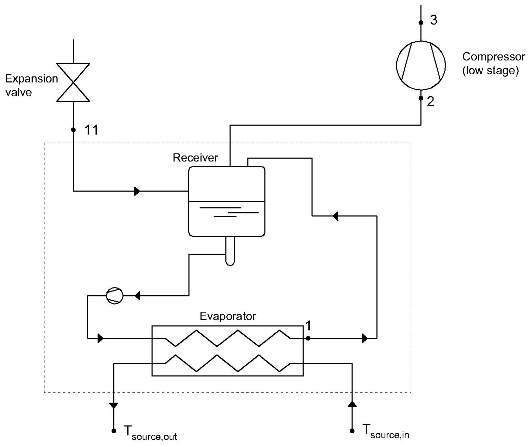



Fig. 4.4: Sketch of a flooded evaporator

flow is located in the two-phase region and flows back to the receiver. Figure 4.4 shows a sketch of a flooded evaporator.

Figure 4.5 shows the T-Q diagram of an evaporator. The heat source is cooled down from an inlet temperature $T_{source,in}$ to the outlet temperature $T_{source,out}$, while the evaporation occurs at a constant temperature $T_{evap}$, since it occurs in the two-phase region. The evaporation temperature is found from the evaporator pinch point. This point corresponds to the point where the smallest temperature difference between the DC flow and the refrigerant flow through the evaporator is met. This pinch point is located between the refrigerant inlet to the evaporator and the DC evaporator outlet flow, $T_{source,out}$.

$$T_{\text{evap}} = T_{\text{source,out}} - \Delta T_{\text{pinch, evap}} \tag{4.2}$$

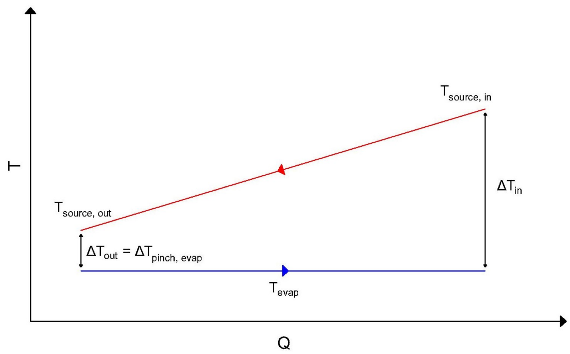



Fig. 4.5: T-Q diagram in a flooded evaporator

The cooling capacity of the evaporator can be calculated by the equation 4.3. To calculate the size of the evaporator at the design stage, its UA value (overall heat transfer coefficient multiplied by the area) is calculated.

$$Q_{\text{evap}} = \dot{m}_{\text{evap}} \cdot (h_{\text{out}} - h_{\text{in}}) \tag{4.3}$$

$$Q_{\text{evap}} = UA_{\text{evap}} \Delta T_{\text{lmtd, evap}} \tag{4.4}$$

Where $\Delta T_{\text{lmtd, evap}}$ is the logarithmic mean temperature difference in the evaporator defined by:

$$\Delta T_{\text{lmtd, evap}} = \frac{(T_{\text{source,in}} - T_{\text{evap}}) - (T_{\text{source,out}} - T_{\text{evap}})}{Ln(\frac{(T_{\text{source,in}} - T_{\text{evap}})}{(T_{\text{source,out}} - T_{\text{evap}})})} \tag{4.5}$$

### 4.2.3 Compressor

The purpose of the compressor in the cycle is to transfer the vapour from the evaporator to the condenser, where the pressure is higher than in the evaporator. This process takes place by increasing the gas pressure by reducing its volume [10]. Depending on the performance of the compressors, they are mainly divided into two types.

- Positive displacement: compress the gas intermittently. Gas compression occurs in a cyclical manner. A volume of gas is introduced into the compressor, it is comprised, discharged and the process starts again.
  - Reciprocating or piston type
  - Rotary
  - Scroll
  - Screw
- Dynamic: compress the gas continuously. The flow of the gas into and out of the compressor is not interrupted at any point in the process.
  - Centrifugal (radial)
  - Axial

The election of one or another type of compressor relies on the application of the cycle. For large refrigeration cycles, screw compressors are widely used.

The ideal process in a compressor is a reversible and adiabatic compression, where the change of entropy in the process is zero. In a real cycle, this process is not isentropic, and therefore, compressors have an isentropic efficiency that is is described by the equation 4.6, defined by the points of the simple cycle.

$$\eta_{is} = \frac{\text{Isentropic Compressor Work}}{\text{Actual Compressor Work}} = \frac{W_s}{W_a} \cong \frac{h_{3s} - h_2}{h_3 - h_2} \tag{4.6}$$

When designing the thermodynamic model it is assumed that the compressors are screw type, whose use is common in large cycles where ammonia is used as a refrigerant. The value of the isentropic efficiency of the compressor, in each stage in the case of the two-stage cycle, is calculated by the equation 4.7 [10]. In it, the pressure ratio, the built-in pressure ratio and the polytropic exponent influence the final value of the isentropic efficiency.

$$\eta_{\text{is}} = \eta_{\text{opt}} \frac{(p_{\text{out}}/p_{\text{in}})^{(\kappa-1)/\kappa} - 1}{\pi_{\text{bi}}^{(\kappa-1)/\kappa} - \frac{\kappa-1}{\kappa}\pi_{\text{bi}}^{-1/\kappa}(\pi_{\text{b}i} - p_{\text{out}}/p_{\text{in}}) - 1} \tag{4.7}$$

Where the built.in pressure ratio is calculated with the following equation and $v_i$ is the built-in volume ratio of the compressor and k the polytropic exponent.

$$\pi_i = v_i^k \tag{4.8}$$

### 4.2.4 Intercooler-Economizer

The intercooler in a device used in the intermediate pressure stage of a two-stage cycle. It is used to separate the two-phase refrigerant. At this way, it collects the refrigerant at the outlet of the expansion valve of the high pressure stage and at the outlet of the low pressure compressor. At the same time it ensures the entrance of liquid refrigerant in the expansion valve of the low pressure stage and the entrance of vapour in the high pressure compressor. The pressure in the intercooler is the selected for the intermediate stage (intermediate pressure) in the cycle. This pressure is selected by the equation 4.1. The energy balance in the intercooler follows the equation 4.9. The points represents the two-stage cycle in figure 4.2.

$$\dot{m}_{\text{high}} h_9 + \dot{m}_{\text{low}} h_3 = \dot{m}_{\text{high}} h_4 + \dot{m}_{\text{low}} h_{10} \tag{4.9}$$

### 4.2.5 Condenser

The principle of operation of a condenser is similar to the operation of an evaporator, with the difference that the condenser delivers the heat to a heat sink. Through the figure 4.6, it is possible to observe the different processes that are carried out inside a condenser, making that its sizing be divided in 3 parts: desuperheating, condensation and subcooling.

- Desuperheating: the refrigerant is cooled from the point at which it leaves the high pressure compressor until it reaches the saturated vapour state.
- Condensation: this process covers from the saturated vapour state until the saturated liquid state is reached in the refrigerant. This process takes place at a constant temperature.
- Subcooling: in this process, the refrigerant is cooled beyond the saturated vapour state.

In order to achieve a high value of the COP, the temperature in the condenser should be the lowest possible. To set the condensation temperature, it is established that the pinch point in the condenser occurs at the end of the desuperheating process (or what is the same, just before starting the condensation process at a constant temperature). In addition, in both the design and off-design conditions of the heat pump, the temperature difference in the subcooling is set. In some occasions,

$\Delta T_{in}$ can be equal to $\Delta T_{pinch,\ cond}$ if the difference in the subcooling is set free.

$$\Delta T_{\text{pinch, cond}} = T_6 - T_{16} \tag{4.10}$$

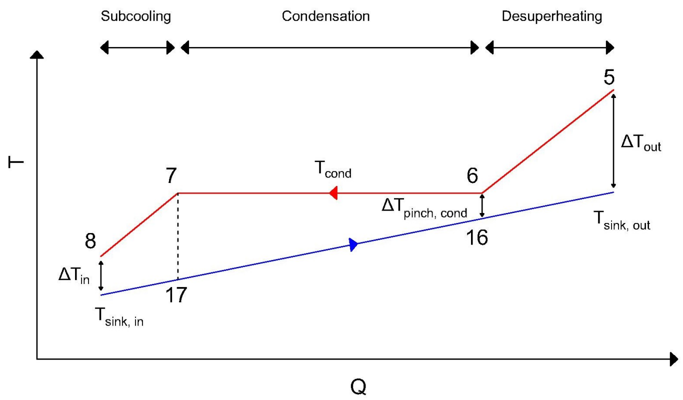



Fig. 4.6: T-Q diagram in a condenser

In the case of the condenser, to obtain its sizing, the calculation must be divided into each of the three parts that it is composed of. To do so, the logarithmic mean temperature difference is calculated in each of the sections.

$$Q_{\text{dsh}} = UA_{\text{dsh}}\Delta T_{\text{lmtd, dsh}} = \dot{m}(h_5 - h_6) \tag{4.11}$$

$$Q_{\text{c}} = UA_{\text{c}}\Delta T_{\text{lmtd, c}} = \dot{m}(h_6 - h_7) \tag{4.12}$$

$$Q_{\text{sc}} = UA_{\text{sc}}\Delta T_{\text{lmtd, sc}} = \dot{m}(h_7 - h_8) \tag{4.13}$$

$$\Delta T_{\text{lmtd, dsh}} = \frac{(T_5 - T_{\text{sink, out}}) - (T_6 - T_{16})}{Ln(\frac{(T_5 - T_{\text{sink, out}})}{(T_6 - T_{16})})} \tag{4.14}$$

$$\Delta T_{\text{lmtd, c}} = \frac{(T_6 - T_{16}) - (T_7 - T_{17})}{Ln(\frac{(T_6 - T_{16})}{(T_7 - T_{17})})} \tag{4.15}$$

$$\Delta T_{\text{lmtd, sc}} = \frac{(T_7 - T_{17}) - (T_8 - T_{\text{sink, in}})}{Ln(\frac{(T_7 - T_{17})}{(T_8 - T_{\text{sink, in}})})} \tag{4.16}$$

In this way, the size of the evaporator will be obtained as the sum of each of its component sections.

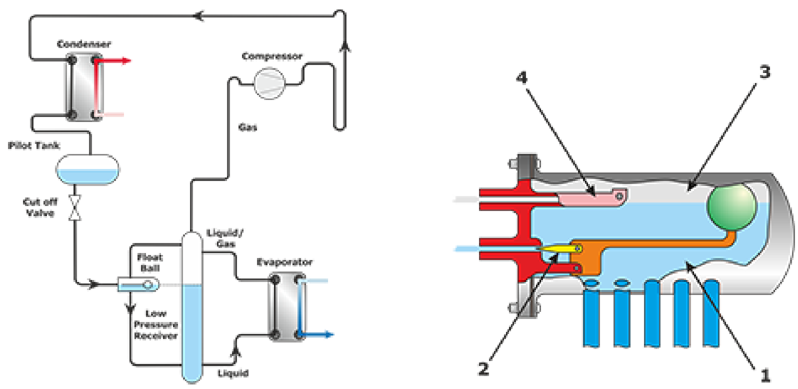
$$UA_{\text{cond}} = UA_{\text{dsh}} + UA_{\text{c}} + UA_{\text{sc}} \tag{4.17}$$

### 4.2.6 Expansion valve

The main purpose of the expansion valve is to maintain the pressure difference between the different stages of the heat pump cycle and to regulate the refrigerant flow that goes into the evaporator. The expansion devices can be classified into different types according to control they make.

- Thermostatic expansion valve
- External equaliser
- Electronic expansion valve
- Capillary tubes
- Low-pressure float valve
- High-pressure float valve

The most commonly used type of expansion device when utilizing flooded evaporators is a low-pressure float valve as well as in large ammonia cycles [10]. Typically the device is assembled in parallel with the receiver that separates the liquid and the gas before the entrance of the evaporator. The valve controls the level of refrigerant in the receiver connected to the evaporator so that if the level in the receiver increases, the valve closes the flow and the valve opens when the level of refrigerant decreases. The valve stores the low pressure liquid (1) and the steam generated in the evaporator (3). If the liquid level in the receiver increases, the valve closes (2), so that the amount of liquid in the evaporator is regulated. The generated vapour is directed to the compressor through the suction line (4). The operating diagram can be seen in figure 4.7. [11] [10]




(a) Cycle with a low-pressure float control valve

(b) Low-pressure float valve

Fig. 4.7: Cycle with a low-pressure float valve

## 4.3 Design parameters

After the analysis made in Chapter 3 on the operation of heating and cooling plants on campus, the main parameters from which the heat pump is designed can be defined. First of all, the boundary conditions under which the heat pump works are established. This mainly refers to the temperatures at which the heating and cooling systems operate on the campus.

As explained in section x, the supply and return temperatures of the district heating system vary according to the heating demand at each time of day. However, it is possible to set fairly clear design temperatures, which will be used for the design of the heat pump. The heating supply temperature is set at 70°C, and the return temperature is 50°C ($T_{DH,s}$ = 70°C; $T_{DH,r}$ = 50°C). In the case of the district cooling system, the design values for supply and return are based on the design data of the campus cooling plants [12]. Thus, the cooling supply has a temperature of 12°C, and the return has a temperature of 18°C in design conditions ($T_{DC,s}$ = 12°C; $T_{DC,r}$ = 18°C). Once the heating and cooling design temperatures are known, the next step is to obtain the condensing and evaporating conditions in the thermodynamic cycle. These conditions are defined from different temperature differences along the different phases of the refrigerant in the thermodynamic cycle, as well as refrigerant temperature differences with both heating and cooling flows. These phases are the superheating after passing through the evaporator, and the subcooling that happens in the last stage of the condenser. In superheating it is assumed that the fluid absorbs heat from the environment once it leaves the evaporator. This heat that is absorbed influences the conditions of the fluid inlet to the compressor, so it must be taken into account, but it does not influence the amount of heat exchanged in the evaporator and therefore does not influence the capacity of the evaporator and the DC. On the other hand, subcooling is another stage that takes place inside the condenser, as explained in subsection 4.2.5, and it does influence the final capacity of the condenser and the DH. The temperature increase and decrease in both stages is set at a magnitude of 3 degrees Kelvin ($\Delta T_{sh}$ = 3 K; $\Delta T_{sc}$ = 3 K).

In addition, the pinch temperature difference is defined for both the evaporator and the condenser. As explained in the corresponding sections, the pinch point defines the operating temperature of a heat exchanger. In this way, the working temperatures of the evaporator and the condenser can be obtained under design conditions. These values are set to a value of 5 degrees Kelvin for both heat exchangers ($\Delta T_{pinch,evap}$ = 5 K; $\Delta T_{pinch,cond}$ = 5 K).

The evaporator load selected for the design conditions is used to define the rest of the cycle characteristics in order to obtain the cycle performance results, as these do not depend on the chosen heat pump capacity. This is the reason for choosing to have a design load in the DC of 1MW. The actual evaporator and condenser capacities will be obtained later in the cost optimisation model. A summary of the design conditions can be observed in table 4.1.

Table 4.1: Design parameters for both one-stage and two-stage cycles.

| **Parameter** | **Design Value** | **Unit** | **Meaning** |
|---|---|---|---|
| $T_{DH,s}$ | 70 | ºC | Supply temperature for District Heating |
| $T_{DH,r}$ | 50 | ºC | Return temperature for District Heating |
| $T_{DC,s}$ | 12 | ºC | Supply temperature for District Heating |
| $T_{DC,r}$ | 18 | ºC | Return temperature for District Cooling |
| $\Delta T_{sh}$ | 3 | K | Superheating |
| $\Delta T_{sc}$ | 3 | K | Subcooling |
| $\Delta T_{pinch,evap}$ | 5 | K | Evaporator pinch difference temperature |
| $\Delta T_{pinch,cond}$ | 5 | K | Condenser pinch difference temperature |
| $Q_e$ | 1 | MW | Evaporator design load |

## 4.4 Off-design cycle

When the cycle is studied outside of design conditions, a series of changes must be made so that the model remains valid. After defining the operating conditions of the heat pump at full load and obtaining the sizing characteristics of the heat pump, it is necessary to study how the heat pump works when operating at part load. This is because the heat pump will not always be operating at full load, as the DC demand varies throughout the year. When the cooling load decreases, the UA values for the condenser must be corrected. The following formula is used for this purpose [13].

$$UA = UA_d * \left(\frac{Qe}{Qe_d}\right)^{0.8} \tag{4.18}$$

All other heat pump design values, such as subcooling or superheating, remain constant. The only values set free are the pinch difference temperatures in both the evaporator and condenser.

# Chapter 5

# Results for the thermodynamic cycle

As it was explain in chapter 4, two thermodynamic models were develop, one singe-stage cycle and another two-stage cycle. This chapter discusses the results of both cycles. In each cycle, the impact of the variation of different parameters with respect to the value they have in the design cycle is studied. The rest of the design values are maintained constant. The parameters that change during the off-design simulations are:

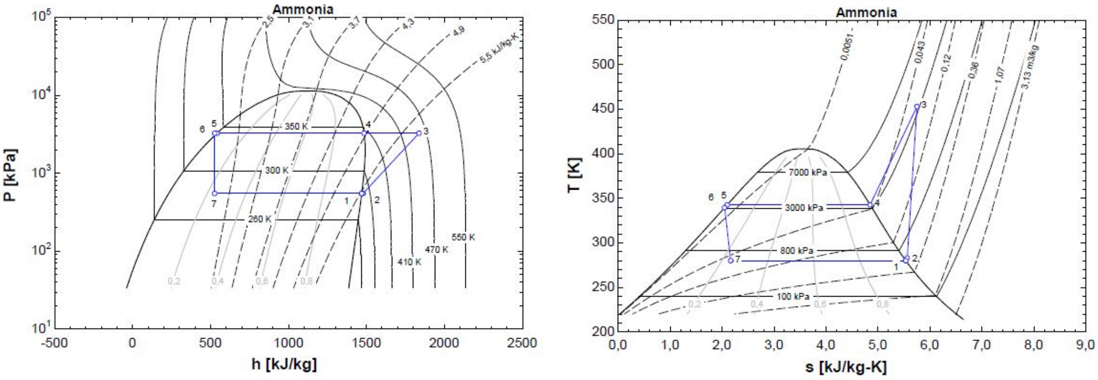
- Evaporator/cooling load: the load varies from 1MW until a partial load of 0.3MW.
- DH supply temperature: the variation goes from 343K to 363K.
- DC return temperature: the variation goes from 286K to 295K.

Both cycles were designed with a cooling load of 1MW in order to study their behaviour both in design and out of design conditions. This is due to the fact that the design cooling load is not known until the cost optimization is done. Furthermore, the operation of the heat pump does not depend on the design load of the evaporator.

## 5.1 Single-stage cycle

The values for the different parameters in the design single-stage cycle are detailed in chapter 4.

The result of the cycle under design conditions can be seen in figure 5.1, with the P-h and T-s diagrams, that show the thermodynamic values of the points that form the cycle. The evaporation and condensation temperatures result on 280K and 342.6K (7°C and 69°C respectively). An important value to have into account is the temperature at the compressor outlet, corresponding to point 3. This temperature can not be too high or the properties of the compressor can be damaged by such a high temperature. This point results on a temperature of 453K (180°C), which is an acceptable value. All this temperatures values can be seen as well in figure 5.2, where the T-Q diagrams for both evaporator and condenser are represented. In this diagrams, the pinch difference temperature can be observed as were explained in figures 4.5 and 4.6.

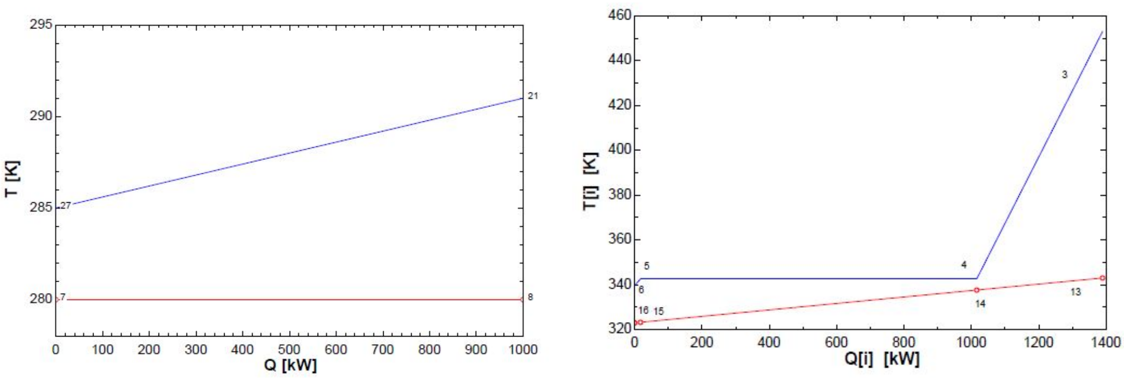



(a) P-h diagram

(b) T-s diagram

Fig. 5.1: Thermodynamic diagrams from the single-stage cycle

In the T-Q diagram, figure 5.2b, the different processes carried out in the condenser can be differentiated as it was explain in section 4.1. As can be assumed, the most important process carried out in the condenser occurs at a constant temperature of condensation, assuming 71.8% of the total heating load. The next process, with a weight of 27%, is the desuperheating process, leaving only 1.2% of the total heating load for the subcooling process. In the evaporator, figure 5.1b, as the superheating is not taken into account in the heat exchanging process for the cooling load, the 100% of the process occurs at constant temperature. In both figure, the pinch point is visible, between the 7 and 27 points in the evaporator and between 4 and 14 in the condenser.




(a) T-Q evaporator

(b) T-Q condenser

Fig. 5.2: Thermodynamic diagrams from the single-stage cycle

The result of the most representative values of the cycle in design conditions are displayed in the table 5.1.

Table 5.1: Design parameters for single stage cycle.

| **Parameter** | **Design Value** | **Unit** |
|---|---|---|
| Q_evap | 1 | MW |
| Q_cond | 1.39 | MW |
| UAe | 131.4 | kW/K |
| UAdsh | 11 | kW/K |
| UAc | 94.13 | kW/K |
| UAsc | 0.9699 | kW/K |
| UAcond | 106.1 | kW/K |
| Tevap | 280 | K |
| T3 | 453 | K |
| Tcond | 342.6 | K |
| Pevap | 551.1 | kPa |
| Pcond | 3272 | kPa |
| pr | 5.937 | - |
| n_is | 0.7729 | - |
| m_dot | 1.06 | kg/s |
| COP_H | 3.648 | - |
| COP_C | 2.625 | - |

In order to know how well the single-stage performs on off-design conditions, some modifications on the DH supply temperature, the DC return temperature and the cooling load were carried out.

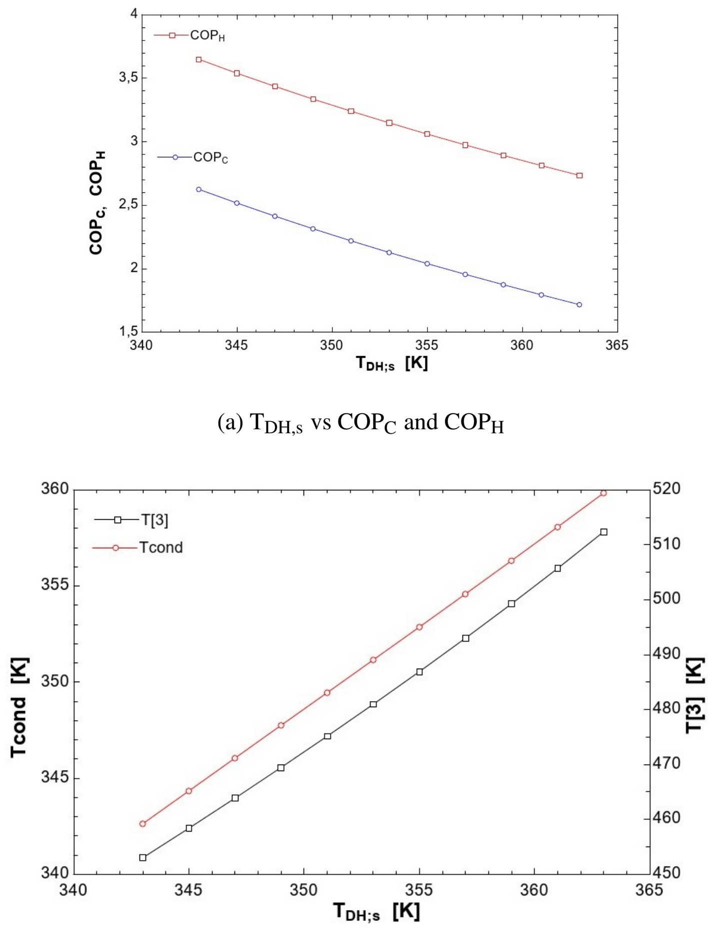
Because the DH supply temperature is highly dependent on the outside temperature (as the outside temperature decreases, the DH supply temperature increases) is very important to know how the COP of the HP varies with the variation of this parameter. For the data study carried out in chapter 3, it is known that the values of $T_{DH,s}$ varies from a minimum of 70°C until a maximum of almost 90°C (for the lowest outside temperature). As it can be seen in figure5.3b, the condensation temperature increases with an increase of the $T_{DH,s}$. This leads to a rise on the lift temperature between the evaporator and the condenser and a decrease on the efficiency of the cycle, observed in the figure 5.3a. In addition, the temperature increase at the compressor outlet follows the same trend as the increase in the condensing temperature, in figure 5.3b. As the DH supply temperature increases, both cooling and heating COP decrease.

The best values for both cooling and heating COP are obtained with the lowest DH supply temperature. Thus, for a value of $T_{DH,s}$ of 70°C, a $COP_H$=3.648 and a $COP_C$=2.625 are obtained.

A similar variation is carried out with the return flow of the DC, although non dependence was manifested with the outside temperature in chapter 3, at least not that strong as with the DH supply temperature. The behaviour of the COPs as a function of $T_{DC,r}$ is similar as with $T_{DH,s}$. As $T_{DC,r}$

increases, the evaporation temperature increases, making the temperature lift smaller and raising the COP for both processes, cooling and heating. In this case, a variation from 286K and 295K was performed in the return flow of the DC. The main difference between the variation of the $T_{DH,s}$ and the variation of $T_{DC,r}$ is the impact on the COP of cooling and heating. If a comparison between the figure 5.3a and the figure 5.4a is made, it can be observed that the gradient in the variation of the $T_{DC,r}$ is lower than when the $T_{DH,s}$ is varied. The cooling and heating COP decreases at a rate of 0.045 for each degree that $T_{DH,s}$ are increased, while, when $T_{Dc,r}$ is increased, the COP increases at a rate of 0.03 for each degree that $T_{DC,r}$ is increased.

As a final noteworthy aspect, the changes and variation in $T_{DH,s}$ does not affect the evaporation temperature at the same time as the variation in $T_{DC,r}$ also does not affect the condensation temperature of the cycle.

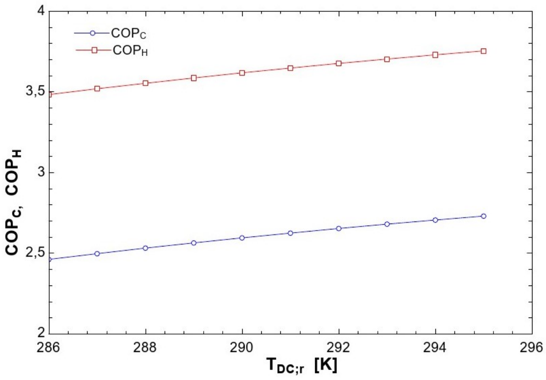



(a) $T_{DH,s}$ vs $COP_C$ and $COP_H$

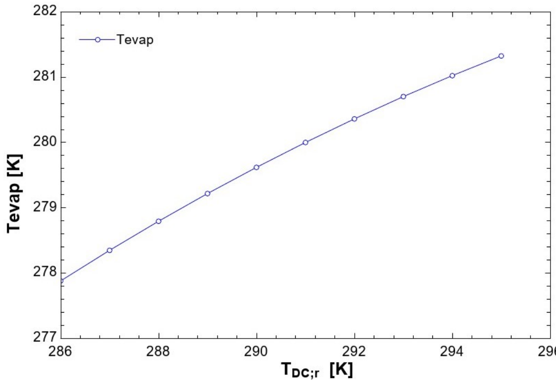
(b) $T_{DH,s}$ vs $T_{cond}$ and $T_3$

Fig. 5.3: Results on the variation of the $T_{DH,\,s}$ in the single-stage cycle

A variation of the cooling load was carried out from 1MW until 0.3MW to see how the cycle behaves in partial loads. The figures 5.5 and 5.6 show how the COP, the $T_{evap}$ and $T_{cond}$ and $\Delta T_{evap}$ and $\Delta T_{cond}$ change with the load.




(a) $T_{DC,r}$ vs $COP_C$ and $COP_H$




(b) $T_{DC,r}$ vs $T_{evap}$

Fig. 5.4: Results on the variation of the $T_{DC,\,r}$

As the load drops, the cooling and heating COP increase. This is because, as it can be seen in figure 5.6a, the pinch difference temperatures in both the evaporator and condenser decrease, or what is the same, the evaporation temperature increase and the condensation temperatures decrease (figure 5.6b), making the temperature lift between these two smaller.

If a comparison between the cooling and heating COP is carried out when the different parameters are varied, it can be observed that, the relation between the $COP_H$ and $COP_C$ is constant and following the $COP_H = COP_C + 1$ relation. This is due to the useless superheating that occurs after the evaporator. If this superheating was taken into account as a process that keep exchanging heat in the evaporator, the relation would not be accomplished.

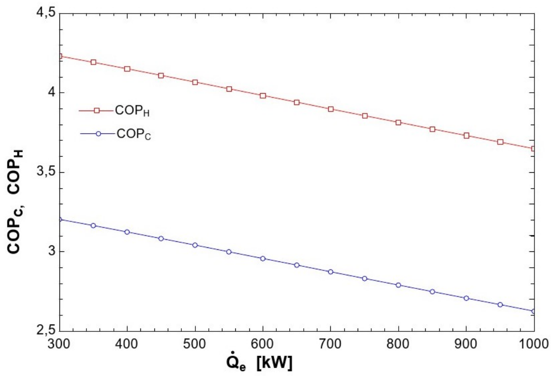
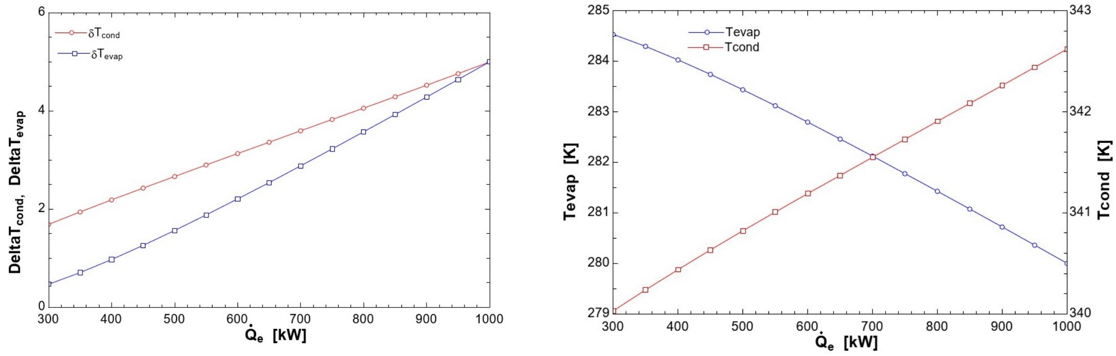



Fig. 5.5: Qe vs COP for the one-stage cycle

Another aspect that can be observed in figure 5.6a is that $\Delta T_{evap}$ is reduced until a value of almost zero for the lowest load while the lowest value of $\Delta T_{cond}$ does not drop below 1.6. The increasing of $T_{evap}$ follows an almost lineal tendency, as well as the decreasing tendency of $T_{evap}$, as it is the trend that also follows $\Delta T_{evap}$, approaching a value of zero.

The EES code that represents the single-stage cycle in both in design and out of design conditions can be found in the Appendix.




(a) $Q_e$ vs $\Delta T$ evaporator and condenser

(b) $Q_e$ vs $T_{cond}$ $T_{evap}$

Fig. 5.6: Results on the variation of the Qe for the one-stage cycle

## 5.2 Two-stage cycle

The design values for the two-stage cycle are the same than for the single-stage cycle, and they are detailed in chapter 4. The intermediate pressure of the cycle is chosen to obtain the maximum COP for both cooling and heating in design conditions. A simulation where the intermediate pressure was varied is shown in the figure 5.7, in which the different COP values for different intermediate pressure values are represented. It can be seen that both values for the heating and cooling COP increase as the intermediate pressure decreases until a maximum point for both is reached, from which, if the intermediate pressure continues to decrease, the efficiency for both processes begins to fall. The maximum efficiency values are achieved with a pressure of 1140 kPa for both process.

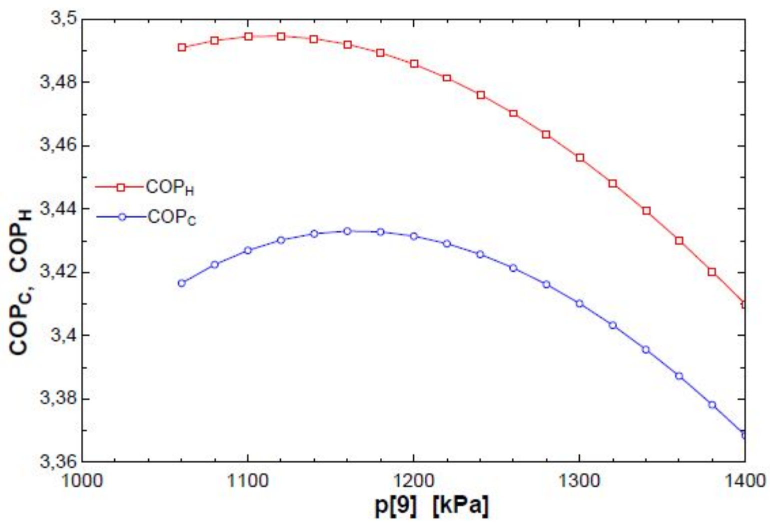
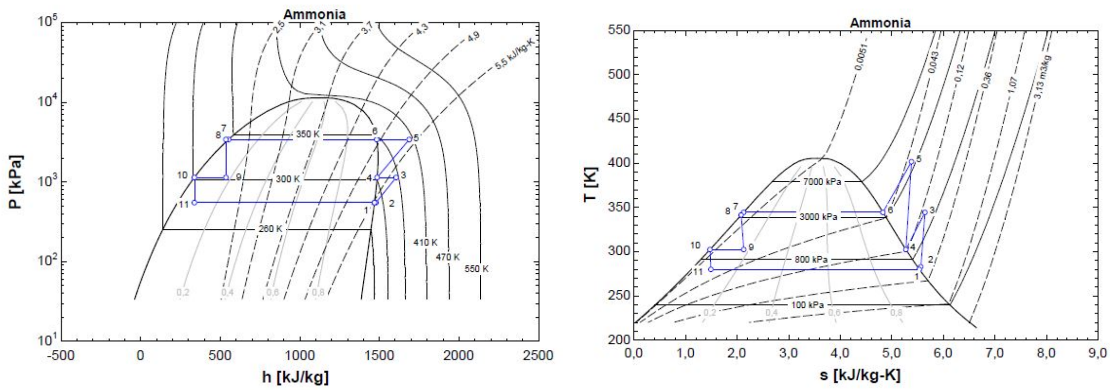



Fig. 5.7: Variation of the $COP_C$ and $COP_H$ with the intermediate pressure.

Once the intermediate pressure is chosen, the results of the cycle in design conditions can be obtained. The figure 5.8 shows the different points of the cycle in both P-h and T-s diagrams and its thermodynamic values. The evaporation temperature is the same than in the single-stage cycle as the way to calculate it is exactly the same in both cycles. By the contrary, the condensation temperature is slightly different than in the single-stage cycle. While in the single-stage the value was 342.6K, in the two-stage cycle this temperature has the value of 344.5K, a difference of 2K.




(a) P-h diagram (b) T-s diagram

Fig. 5.8: Thermodynamic diagrams from the two-stage cycle

The most representative values of the cycle in design conditions are displayed in the table 5.2.

Table 5.2: Design parameters for two-stage cycle.

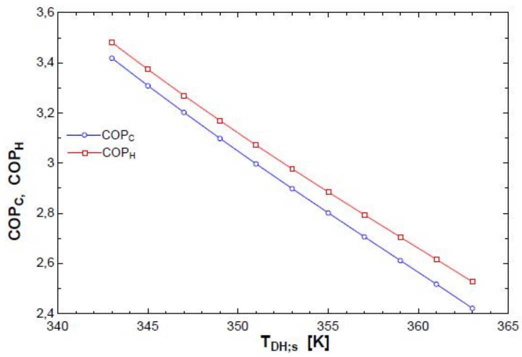
| Parameter | Design Value | Unit |
|---|---|---|
| Q_evap | 1 | MW |
| Q_cond | 1.358 | MW |
| UAe | 131.4 | kW/K |
| UAdsh | 11.06 | kW/K |
| UAc | 97.99 | kW/K |
| UAsc | 0.9885 | kW/K |
| UAcond | 110 | kW/K |
| Tevap | 280 | K |
| T3 | 401.7 | K |
| Tcond | 344.5 | K |
| Pevap | 551.1 | kPa |
| Pcond | 3412 | kPa |
| Pint | 1140 | kPa |
| pr_low | 2.068 | - |
| pr_high | 2.993 | - |
| n_is_low | 0.7867 | - |
| n_is_high | 0.7877 | - |
| m_dot_low | 0.8839 | kg/s |
| m_dot_high | 1.179 | kg/s |
| COP_H | 3.494 | - |
| COP_C | 3.432 | - |

When TDH,s and TDc,r vary in the two-stage cycle, the trends that follow the rest of the parameters of the cycle are similar to those that followed in the single-stage cycle.

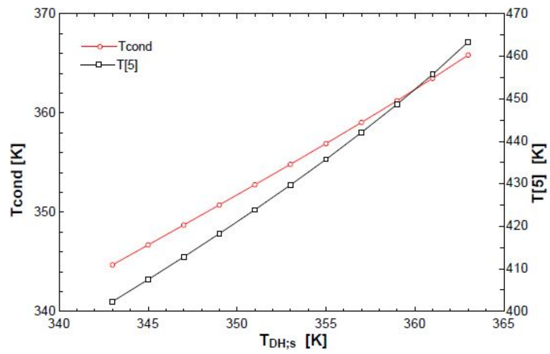
When TDH,s increases with respect to the design value, both heating and cooling COPs decrease, in figure x. This occurs because the condensation temperature increases with increasing TDH, as it did in the single-stage cycle. The same increasing trend follows the second stage compressor outlet temperature, just before the condenser inlet. The difference of both temperatures (in the single cycle and in the two-stage cycle), is that in the case of two-stage, the maximum temperature reached is lower.

The same trends as in the simple cycle follow the COP and the evaporation temperature when TDC,r varies. As Tevap increases, the step between condensing and evaporating temperatures be-comes smaller, resulting in a slight increase in COP. It can also be seen that the dependence of the COP with the variation of TDH,s is greater than with the variation of TDC,r, as it was in the simple cycle. In this case, the heating and cooling COPs vary by 0.05 for each degree that TDH,s varies, while when it is TDC,r that varies, the COP variation results in 0.025.

The difference between the two-stage cycle and the single-stage cycle is the values of the COPs. In this case, the values for the heating and cooling COP are more close.

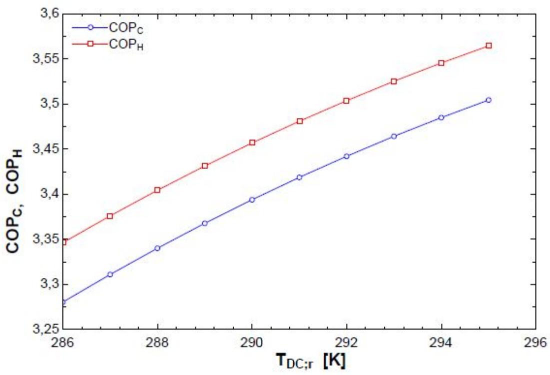



(a) $T_{DH,s}$ vs COP

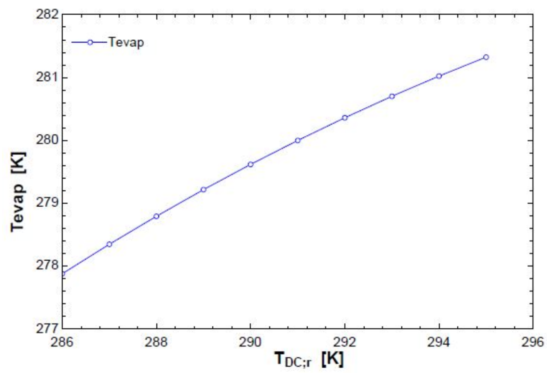



(b) $T_{DH,s}$ vs $T_{cond}$ and $T_5$

Fig. 5.9: $T_{DH,s}$ variations from the two-stage cycle



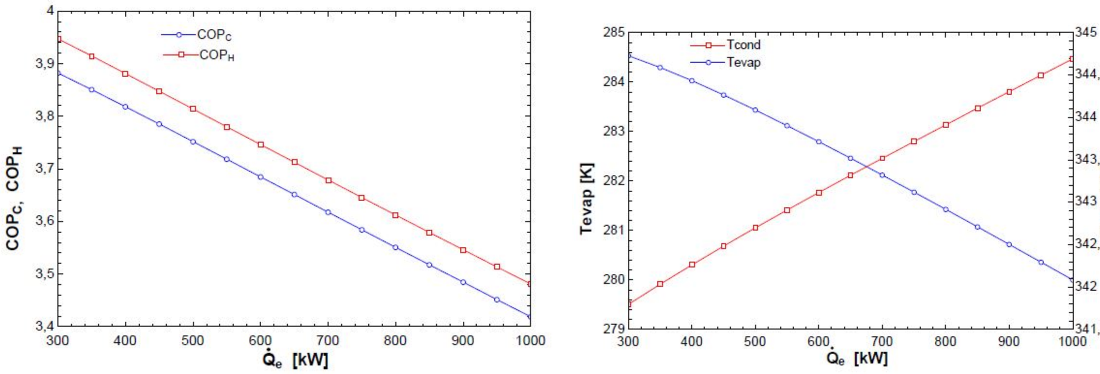

(a) $T_{DC,r}$ vs COP




(b) $T_{DC,r}$ vs $T_{evap}$

Fig. 5.10: $T_{DC,r}$ variations for the two-stage cycle

When the evaporator load is varied, the COP increase, in a more pronounced way for the heating process than for the cooling process. As in the single-stage cycle, the trend that follows the decrease in $T_{evap}$ when the load decreases is also lineal.




(a) $Q_e$ vs COP

(b) $Q_e$ vs $T_{cond}$ and $T_{evap}$

Fig. 5.11: $Q_e$ variations for the two-stage cycle

## 5.3 Conclusions

After the analysis of both cycles, single-stage and two-stages, the conclusion is that with the two-stage cycle, the performance of the heat pump is better. The COP values for heating and cooling purposes are better. In addition, when the heat pump is in an off-design condition because some parameter has changed, the deviation in efficiency that occurs is less when using a two-stage cycle than when the single-stage cycle is used.

# Chapter 6

# Methodology for the optimization model

This chapter investigates the feasibility of producing heating and cooling for the DTU campus in Lyngby from a heat pump.

A model is developed in order to investigate, from an economic point of view, the possibility of introducing, in the current production system of the Lyngby campus of the DTU, a form of co-production of heating and cooling by using a heat pump with a two-stage compression steam cycle. In this way, it is studied if it is economically viable to produce part of the demands of the university campus by means of this system.

To develop the optimization model, the General Algebraic Modelling System (GAMS) software is used, with mixed integer programming (MIP) problem solving. In this way, the model developed must be a model whose mathematical relationships between its different variables are linear.

The results of the two-stage cycle studies carried out in previous chapters 4 and 5 are used as input data for the optimization model so that the optimal heat pump capacity is achieved from the design capacity of the heat pump evaporator. The model compares the current system and the heat pump operating costs and find out the optimal capacity of the evaporator and the condenser and the hourly production for each system.

## 6.1 Objective function

The main objective of the optimization model is to find the heat pump solution that generates the lowest costs. To do this, the objective function minimizes the total annualized costs of the entire production system by running an annual simulation with an hourly timestep.

$$min\ z = \frac{inv_{\text{stor, DC}} + inv_{\text{stor, DH}} + inv_{\text{HP}}}{a_{\text{n, k}}} + \sum_{t=1}^{T} c_{\text{prod, j, t}} + c_{\text{OM}} \tag{6.1}$$

This function is composed of the different investment costs of the HP ($inv_{HP}$), the DH ($inv_{stor, DH}$) and DC ($inv_{stor, DC}$) storage tanks, the operating costs of producing heating and cooling ($c_{prod,j}$) and the costs of operation and maintenance of the whole system. Firstly, the differentiation to be made in the costs is between annual costs and annualised costs.

### 6.1.1 Annualised costs

The annualised costs are made up of the different initial investments that are necessary to be able to produce DH and DC. As mentioned above, the simulation of the optimization model is carried out over a year, so the solution lies in annualizing the investment costs by using equivalent annual cost (EAC). In this way, the annual cost of owning and operating the equipment over its lifetime is obtained. In order to calculate the EAC, the net present value (NPV) is divided by the annuity present value factor, $a_{n, k}$.

$$EAC = \frac{NPV}{a_{\text{n, k}}} \tag{6.2}$$

Where $a_{n, k}$ is calculated by the equation 6.3, with n being the number of years of project lifetime and k the discount rate.

$$a_{\text{n, k}} = \frac{1-(1+k)^{-n}}{k} \tag{6.3}$$

The simulation assumes a lifetime of the entire equipment of 20 years [14]. Thus, the investment costs are divided over the lifetime of the project. The interest rate k is calculated using the interest rate of the Danish government. For this project, 4% interest rate is used [15].

Within the investment costs found in the NPV, are the investment cost of the heat pump and the investment costs of the storage tanks for both DH and DC.

$$NPV = inv_{\text{stor, DC}} + inv_{\text{stor, DH}} + inv_{\text{HP}} \tag{6.4}$$

The equation for obtaining the cost of investing in HP has two terms. One that depends on the maximum capacity of the heat pump ($c_{size}$) and another fixed term ($c_{fixed}$). The value of both terms results in 15952685 (DKK) and 4031493 (DKK/MW) [16].

$$inv_{\text{HP}} = c_{\text{fixed}} + c_{\text{size}} \cdot Q_{\text{DC, max}} \tag{6.5}$$

On the other hand, the investment costs of the storage tanks only consist of a term that depends on the capacity of the tanks in MWh. The value of these terms is the same for both the DH tank and the DC tank.

Where i, in the equation 6.7, refers whether to the DC or the DH storage tanks. The investment cost of the storage tanks is obtained in DKK/m$^3$, with a value of 1500 DKK/m$^3$ [17]. The capacity of the tanks is obtained in the simulation in units of MWh, so it is necessary to make a modification of the units in the cost of the tanks. In order to transform the cost to DKK/MWh, the equation 6.6 is used first. For both supplies, heating and cooling, the temperature difference used corresponds to the difference between the supply and return temperatures. Thus, $\Delta T_{DC}$ = $T_{DC,\,r}$ - $T_{DC,\,s}$, and $\Delta T_{DH}$ = $T_{DH,\,s}$ - $T_{DH,\,r}$.

$$cost_{\text{stor, i}} = \frac{cost_{\text{tanksize, i}}}{\rho \cdot Cp \cdot \Delta T_{\text{i}} \cdot convert} \tag{6.6}$$

$$inv_{\text{stor, i}} = cost_{\text{stor, i}} \cdot stor_{\text{i, max}} \tag{6.7}$$

### 6.1.2 Yearly costs

The rest of the costs taken into account and that remain to be explained are the annual costs of the different production systems. These include the costs of the production of heating and cooling throughout the year, depending on demand and the operating and maintenance costs. These costs do not have to be annualised, as they are only taken into account over the period of one year.

$$Annual\ costs = \sum_{t=1}^{T} \left(c_{\text{prod, DC. t}} + c_{\text{prod, DH, t}}\right) + c_{\text{OM}} \tag{6.8}$$

$$c_{\text{prod, DC, t}} = \sum_{t=1}^{T} \left( \left( \frac{Q_{\text{DC, n, t}}}{COP_{\text{comp}}} + \frac{Q_{\text{DC, HP, t}}}{COP_{\text{C, HP, t}}} \right) \cdot c_{\text{elect}} \right) \tag{6.9}$$

$$c_{\text{prod, DH, t}} = \sum_{t=1}^{T} \left(Q_{\text{DH, n, t}} \cdot c_{\text{gas, heating}}\right) \tag{6.10}$$

The production cost of the cooling depends on three factors.

The first one is the price of the electricity ($c_{elect}$). After consulting the cost of electricity for DTU campus, an average value of 1200 DKK/MWh consumed is taken. Although the cost of electricity is not constant during the year and DTU trade electricity directly at Nordpool, this value is an acceptable approximation and a common value when designing projects [12].

The other two remain factors are the cooling production and the efficiency of each system, the current compressors system and the HP. The cooling production for each system is explained below, as well as the HP efficiency, through the COP. Finally, the COP of the compressors is estimated at a value of 4,25, based on the technical data provided by the manufacturers [12], as well as a study of the cooling production and the electricity consumption with the available data from chapter 3 [18].

The O&M costs are calculated through the equation 6.11. This equation takes into account the cost of an overhaul of the current refrigeration system and the cost of the HP.

The compressors that produce cooling in the current system need an overhaul every 14.000 and 100.000 hours of operating, depending on the type of compressors. These overhauls costs over 80.000 and 40.000 DKK each. To introduce this cost in the model without making the difference between the type of compressors, an average cost in the form of DKK/hour of operation of the current system has been obtained ($c_{OM,\,Comp}$). This cost takes the value of 2 (DKK/h) and is multiplied by the total operating hours of the compressors in the current system ($totalhours_{DC,\,n}$).

On the other hand, the O&M costs of the HP have two terms, a fixed cost per year and a cost that depends on the total annual heat production, $c_{\text{OM, HP, fixed}}$ and $c_{\text{OM, HP, prod}}$ respectively. The first one takes a value of 270 DKK per year, while the second has a value es 0,29 DKK per MWh produced during the year.

$$c_{\text{OM}} = (c_{\text{OM, Comp}} \cdot totalhours_{\text{DC, n}}) + (c_{\text{OM, HP, fixed}} + c_{\text{OM, HP, prod}} \cdot totalprod_{\text{DH, HP}}) \tag{6.11}$$

## 6.2 Model development

The objective function of the model has already been explained and detailed in the previous section, but for the model to work a series of restrictions and descriptions of how the system should work must be detailed. The model is based in the following sketch.

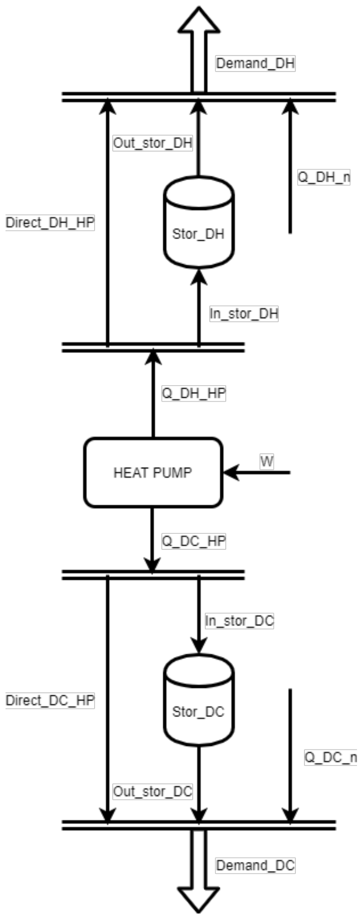



Fig. 6.1: Model diagram of the DH and DC production

The different constrains of the model are enumerated as following.

Through the equations 6.12 and 6.13, restrictions on heating and cooling demands are introduced. The model has to ensure the supply of heating and cooling demands at all timestep of the simulation. In this way, the demand that is requested hour by hour must be supplied by means of three modes. The first mode is the direct production of heating and/or cooling in the heat pump, in other words, the production of DH or DC at hour t provides part or all of the demand at hour t. If the direct production at the HP does not cover all the demand for DH or DC, the rest must be supplied by the storage tank ($out_{stor,\ i,\ t}$) or, as a last resort, there is the possibility to supply DH and DC by the existing system on campus ($Q_{i,\ n,\ t}$).

$$d_{\mathrm{DC,\ t}} = direct_{\mathrm{DC,\ HP,\ t}} + out_{\mathrm{stor,\ DC,\ t}} + Q_{\mathrm{DC,\ n,\ t}} \tag{6.12}$$

$$d_{\mathrm{DH,\ t}} = direct_{\mathrm{DH,\ HP,\ t}} + out_{\mathrm{stor,\ DH,\ t}} + Q_{\mathrm{DH,\ n,\ t}} \tag{6.13}$$

The equations from 6.14 to 6.16 show the model restrictions on DH and DC production in the heat pump. The total production of DH and DC in the HP at each timestep t should be equal to the direct demand supply ($direct_{i,\ HP,\ t}$) and the input flow to the respective storage tanks ($in_{stor,\ i,\ t}$). In addition, DH production at each timestep t, depends directly on DC production. Thus, if the evaporator of HP is running at part load, the DH output will vary according to the cooling load. This is reflected in equation 6.16.

$$Q_{\mathrm{DC,\ HP,\ t}} = direct_{\mathrm{DC,\ HP,\ t}} + in_{\mathrm{stor,\ DC,\ t}} \tag{6.14}$$

$$Q_{\mathrm{DH,\ HP,\ t}} = direct_{\mathrm{DH,\ HP,\ t}} + in_{\mathrm{stor,\ DH,\ t}} \tag{6.15}$$

$$Q_{\mathrm{DH,\ H,\ t}} = f(Q_{\mathrm{DC,\ HP,\ t}}) \tag{6.16}$$

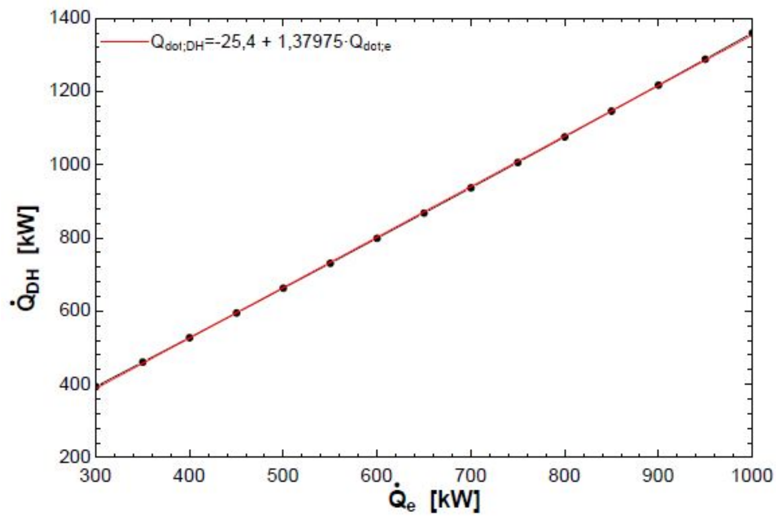



Fig. 6.2: Variation of the DH production with the load (DC production).

A minimum and a maximum capacity production in the HP is added in order to know if the HP is turned off at any time of the simulation time. The minimum production capacity is set on 0,5 MW, while the maximum production capacity is a value large enough to not be considered a constrain. In the equations 6.17 and 6.18, the variable $i_t$ show whether the HP is running at time t or not.

$$Q_{\text{DC, HP, t}} \geq Q_{\text{DC, HP, min}} \cdot i_{\text{t}} \tag{6.17}$$

$$Q_{\text{DC, HP, t}} \leq Q_{\text{DC, HP, max}} \cdot i_{\text{t}} \tag{6.18}$$

The restrictions for the storage tanks should give data on their level at each time step of the simulation. The restrictions for the storage tanks should give data on their level at each time step of the simulation. In this way, the level maintained at each t should be equal to the level at timestep t-1 and the result of the tank input and output flows at time t ($\text{in}_{\text{stor, i, t}}$ - $\text{out}_{\text{stor, i, t}}$).

$$stor_{\text{DC, t}} = stor_{\text{DC, t-1}} + in_{\text{stor, DC, t}} - out_{\text{stor, DC, t}} \tag{6.19}$$

$$stor_{\text{DH, t}} = stor_{\text{DH, t-1}} + in_{\text{stor, DH, t}} - out_{\text{stor, DH, t}} \tag{6.20}$$

Finally, in order to know the capacities of the heat pump and the DH and DC storage tanks, it is necessary to study the maximum values that reach these magnitudes during the annual simulation.

$$Q_{\text{DC, max}} \geq Q_{\text{DC, HP, t}} \tag{6.21}$$

$$stor_{\text{DC, max}} \geq stor_{\text{DC, t}} t = 1, \ldots, T \tag{6.22}$$

$$stor_{\text{DH, max}} \geq stor_{\text{DH, t}} \tag{6.23}$$

## 6.3 Input data

For the model to work properly, it is not only necessary to develop the equations that constitute it accurately. It is also equally important to correctly select the input data needed to obtain the desired results. The equations that conform the model and the different constrains linked to it have been explained in the previous section.

In this section we detail the different input data that are defined in the model according to their origin.

### 6.3.1 Technical data of the thermodynamic cycle

The possible cycles to be used have been studied in detail in chapters 4 and 5. In addition, the two-stage cycle has been selected as the one that gave the best results for the design of the heat pump. In this way, the technical data used in the development of the optimization model will be those corresponding to this cycle. The most important data when taking into account the performance of the cycle in its normal operation are the $\text{COP}_\text{C}$ and $\text{COP}_\text{H}$ values.

After the results obtained in chapter 5, it can be clarified that the variation of the $\text{COP}_\text{C}$ and $\text{COP}_\text{H}$ of the cycle depends more on the variation of the cycle load and the DH supply temperature.

Knowing this, it should be kept in mind that a linear model has been chosen to be developed, so the relationships between the different variables of the model must be linear. If the evaporator load is chosen as part of the decision variables of the model, the $COP_C$ and $COP_H$ values should not depend on the load so that the model remains linear. This can be observed in the following equations.

The COP of each cooling and heating process depends on the load by using a linear function *a·load + b*. In order to know the load at which the heat pump is operating at any given time, the total capacity of the heat pump must be known. This variable is one of the decision variables of the model.

$$load = \frac{Q_{\text{DC, HP, t}}}{Q_{\text{DC, max}}} \tag{6.24}$$

In turn, the objective function of the model, z, calculates the operating costs of the entire production system. To calculate the costs of heat pump production, the energy consumed by the compressor in the cycle is used. This has been explained in section x. The simplest way to obtain this work is by using the COP.

$$c_{\text{prod,DC, HP, t}} = W_{\text{comp, HP, t}} \cdot c_{\text{elect}} = (\frac{Q_{\text{DC, HP, t}}}{COP_{\text{C, HP, t}}}) \cdot c_{\text{elect}} \tag{6.25}$$

If in the above equation, equation 6.25, the term of the COP is replaced by the linear relationship that it has with the term of the charge defined above and in the x equation, the following expression is obtained.

$$c_{\text{prod,DC, HP, t}} = (\frac{Q_{\text{DC, HP, t}}}{a \cdot load + b}) \cdot c_{\text{elect}} = (\frac{Q_{\text{DC, HP, t}}}{a \cdot \frac{Q_{\text{DC, HP, t}}}{Q_{\text{DC, max}}} + b}) \cdot c_{\text{elect}} \tag{6.26}$$

The resulting expression in equation 6.26 is non-linear, since the term of $Q_{\text{DC, HP, t}}$ appears in both parts of the fraction. Therefore, it is necessary to relate the coefficients of performance of the heating and cooling processes of the heat pump with a variable that would make the model linear.

If one returns to the study carried out in chapter 5, it can be seen that the parameter that makes the value of the COPs vary most significantly, besides the load, is the DH supply temperature. Making the values dependent on the coefficient of performance of the $T_{DH,s}$ would make the model linear. Furthermore, as shown in chapter 3, the DH supply temperature is not constant throughout the year, since it depends heavily on the outdoor temperature at any given time of the year.

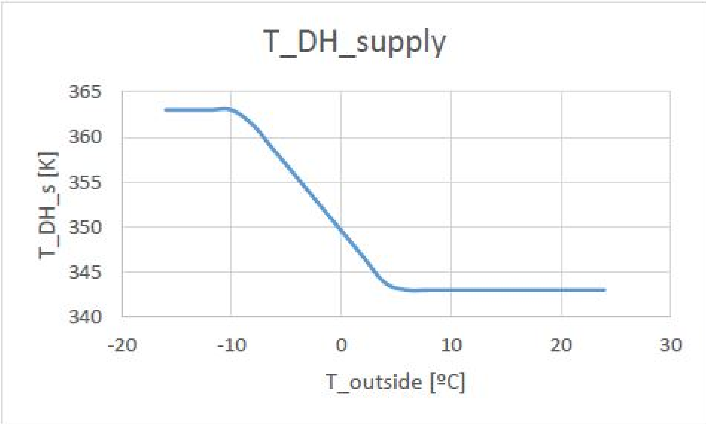



Fig. 6.3: Variation of $T_{DH,s}$ with the outside temperature.

Following the figure 6.3, the relation between the DH supply temperature and the outside temperature are settled as the following.

$$\begin{cases} T_{\text{DH, s, t}} = 343\,[K] & 4.5\,°C < T_{\text{out}} \\ T_{\text{DH, s, t}} = (-1.4584 \cdot T_{\text{out}} + 349.612)\,[K] & -10\,°C < T_{\text{out}} < 4.5\,°C \\ T_{\text{DH, s, t}} = 363\,[K] & T_{\text{out}} < -10\,°C \end{cases} \tag{6.27}$$

The variation of the $COP_C$ and $COP_H$ values as a function of the variation of the $T_{DH,s}$ has already been shown in chapter 5. This dependence can be written by means of a linear function of the type $a \cdot T_{DH,s} + b$, and is shown in figure 6.4 and in the equation 6.28.

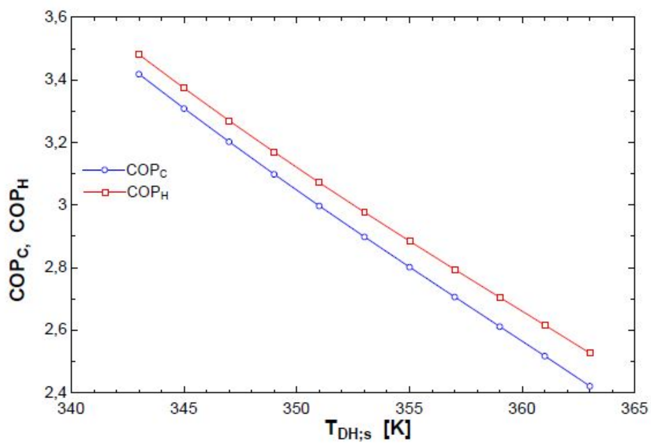



Fig. 6.4: Lineal dependency of both $COP_C$ and $COP_H$ with $T_{DH,s}$

$$\begin{cases} COP_{\text{C, HP, t}} = 19.932 - 0.0482713 \cdot T_{\text{DH, s, t}} \\ COP_{\text{H, HP, t}} = 20.9174 - 0.0510138 \cdot T_{\text{DH, s, t}} \end{cases} \tag{6.28}$$

### 6.3.2 Model parameters

Once the performance of the cycle is defined in the optimization model, it is needed to detail the rest of the parameters that are used as input data.

Firstly, the outside temperature is introduced. The data for this parameter is compiled from the DTU Climate Station [19], a web page where different climate parameters are recorded at the DTU campus in Lyngby. The desired time period is selected and the outdoor air temperature is obtained at the selected time step. In this case the time step must be one hour in a whole year's data recording period, resulting in figure 6.5.

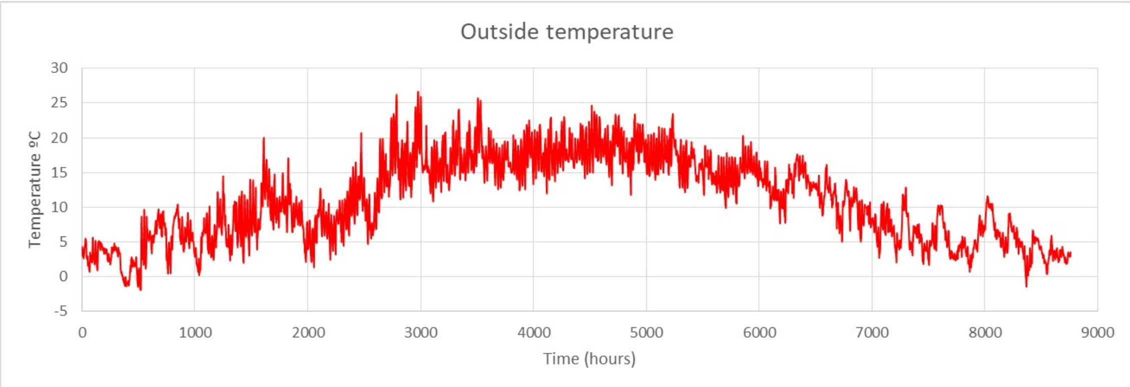



Fig. 6.5: Variation of the outside temperature through the year.

When the yearly data for the outside temperature is available, the DH supply temperature can be obtained using the equation 6.27. The result of applying it can be seen in figure 6.6. It can be seen that in the warmer months (central/summer months), the DH supply temperature is maintained at a constant value of 343K. On the other hand, in the extreme months corresponding to the winter months and the colder months, where the outside temperature is notably lower, the DH supply temperature varies much more from a minimum of 343K, in the hours where the temperature is higher, to a maximum of 352.5K, for the lower outside temperatures. These values may change depending on the outdoor temperatures reached each year. In any case, and due to the way in which the DH supply temperature has been defined, the maximum value that can be reached is 363K, by the equation 6.27.

One of the most important input data of the model is the heating and cooling demands to be supplied by the system. These demands have already been studied in chapter 3. The data are collected from the different registration points that the campus has in the current heating and cooling production systems. To obtain the data collected, the database managed by the DTU's Campus Services is used [18]. In the web page you select the reading points from where you want to obtain the data. In this case, the desired time step cannot be controlled, but the group of data can be easily manipulated and obtain a data set with the desired time step since the acquired format is CVS. The heating and cooling demands with which the optimization is carried out can be seen in figures 6.7 and 6.8.

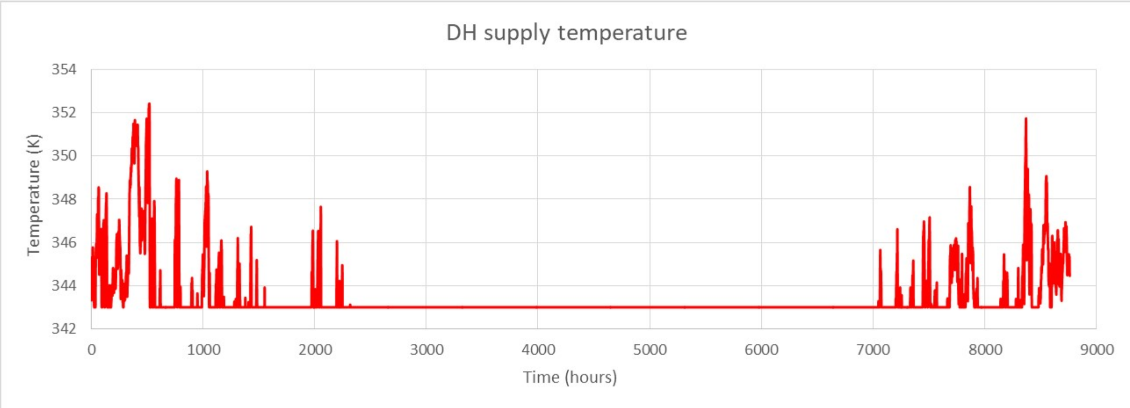



Fig. 6.6: Variation of the $T_{DH,\,s}$ through the year.

The demand for heating varies considerably throughout the year, with a maximum of 25MW in the coldest months and a minimum production of around 1.5 MW in the summer months. The average value of heating demand throughout the year is 7MW.

On the other hand, the values for cooling demand are based on the cooling demand showed in chapter 3. As it was discussed in chapter 3, it is known that the cooling demand has been increasing until reaching a peak of 8MW approximately. For this reason and also due to there is no recent registered data for the cooling demand, the data that will be used for this purpose in the model is the annual demand analysed in chapter 3 multiplied by a factor of 3.5. The basis on choosing this factor is to obtain a cooling demand of around the 8MW during the hottest months, in addition to some higher peak demand. Hence, the cooling demand has a maximum peak of almost 12 MW and a minimum of around 2 MW, with a range of demand during the summer months between 6MW and 8MW.

This study of the variation in heating and cooling demand values has already been carried out in more detail in chapter 3.

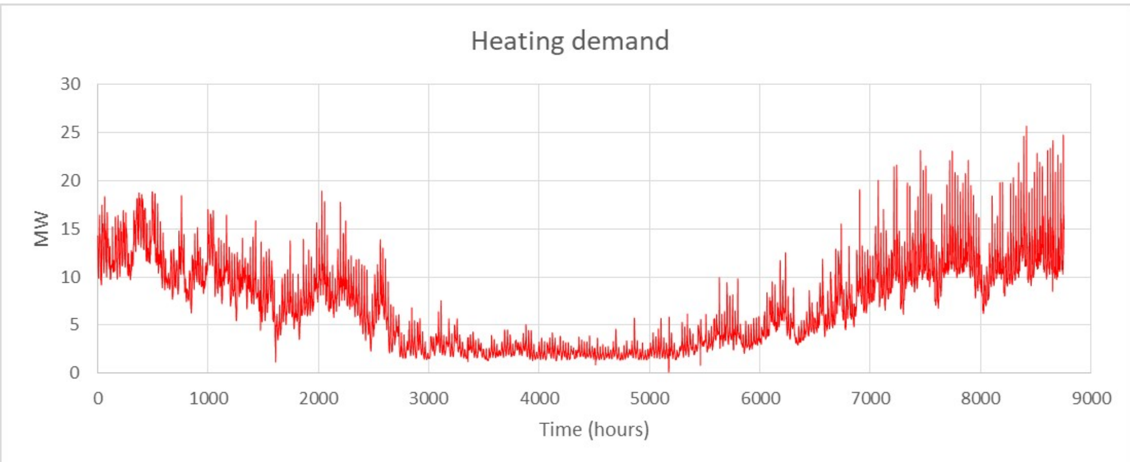



Fig. 6.7: Hourly demand for heating.

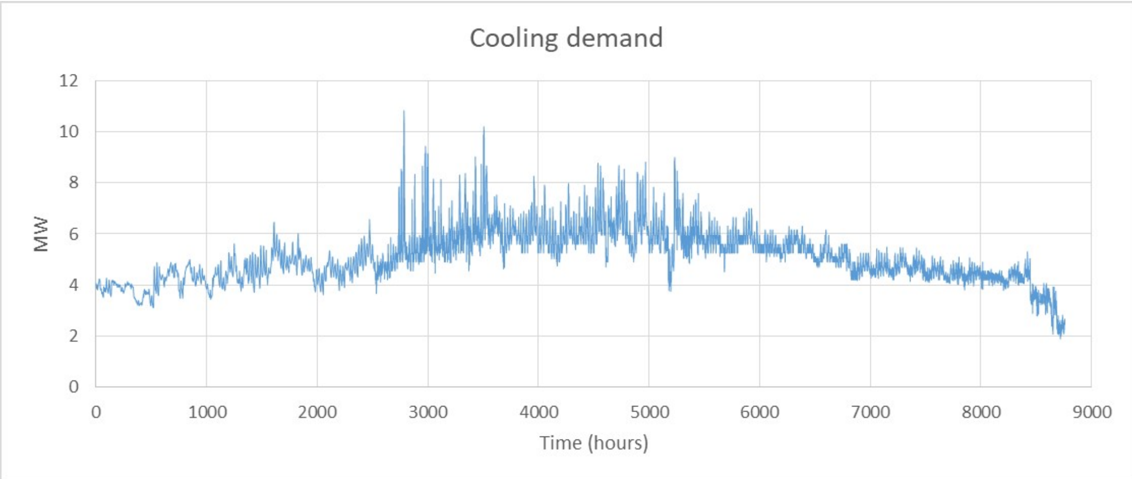



Fig. 6.8: Hourly demand for cooling.

# Chapter 7

# Results on the optimization model

This chapter presents the results obtained after the execution of the optimization model of the heating and cooling co-production plant.

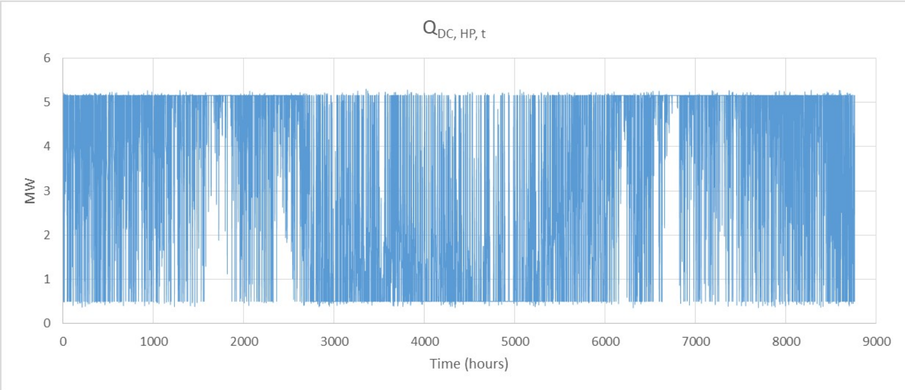
The presentation of the data is divided into two parts. First, an annual analysis is presented, showing the results of the simulation over the one-year time horizon. Then, an analysis of the plant's performance over a one-week period is made.

## 7.1 Results of the production system

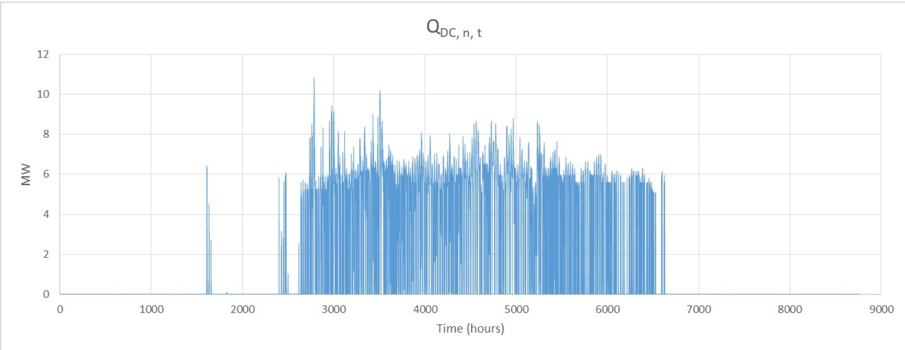
The results of the production of heating and cooling according to the annual demands are presented below. Both graphs of the results of the most representative variables at each time step of the simulation and the load duration curves are shown. In figures 7.1 and 7.2, the hourly production of heating and cooling in both the HP and the current system in campus can be seen. The reason why that figures 7.1a and 7.2a are very similar lies in the fact that the production of heating in the HP depends directly on the production of cooling, as it was expose in equation 6.16 and in figure 6.2. Therefore, the DH production at the HP will be proportional to the production of DC at any time of the simulation. The same is not applicable to DH and DC production on the current campus systems, as these productions depend on each demand and the supply that have not been covered by the HP production.

Another striking aspect of the figures is the amount of demand covered by the current campus systems, both for heating and cooling. It can be seen in figure 7.1b that the current cooling system is only necessary in the summer months, when the demand for cooling increases considerably with respect to the demand during the rest of the year.

On the other hand, the production of heating through the current system is much greater throughout the year and is reduced, reaching not to produce DH in the central months. This means that the HP is sufficient to cover the demand for DH in the summer months, since, as seen in figure 6.7, this is reduced considerably compared to the rest of the year.




(a) Production of DC in the HP

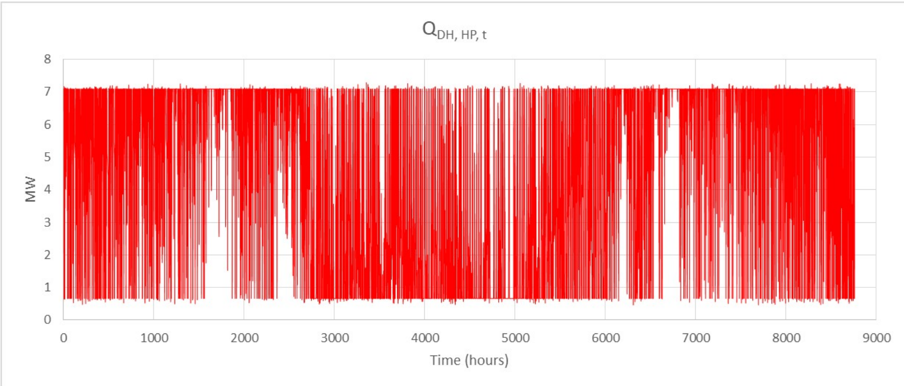



(b) Production of DC in the current system

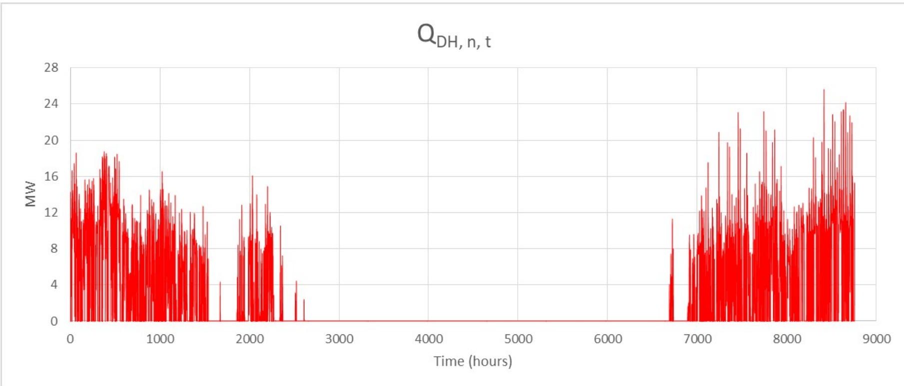
Fig. 7.1: Hourly DC production

In addition, the load duration curves in the HP for both DC and DH are shown in figure 7.3. As can be assumed, both curves have the same shape, but with different values. Around 4000 hours of HP operation at full load are achieved, which means a production of 5.15 MW for cooling purposes and 7 MW for heating purposes each hour for almost the half of the year. If you look at the cooling demand curve, figure 6.8, and compare it with the load duration curve of the DC production in the heat pump, figure 7.3a, you can see that the peak load of the HP, 5.15MW it has a close value to the average cooling demand, 5.82MW. This has as a result that the HP would be able to supply a large part of the annual cooling demand, despite a backup will be needed for cooling generation in the months where demand is greatest.

If the load duration curves for the current system for both heating and cooling are studied, in figure 7.4, some details and similarities can be observed. For both curves, the maximum operating hours reach around 2500 hours. The peak reached in the cooling system is around 11MW, matching the peak in the cooling demand during summer, where the compressors are turned on according to figure 7.1b. After the peak, the production is reduced considerably, maintaining an average of 5.8MW, until it reaches the 2500 hours. The load duration curve for the heating system

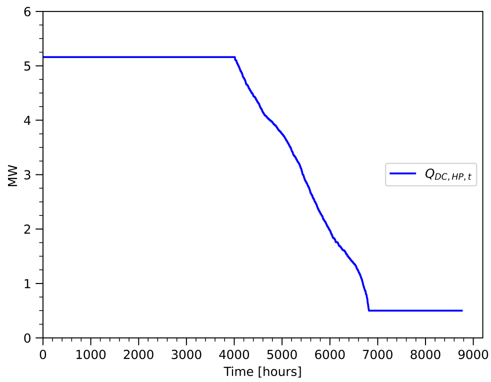



(a) Production of DH in the HP

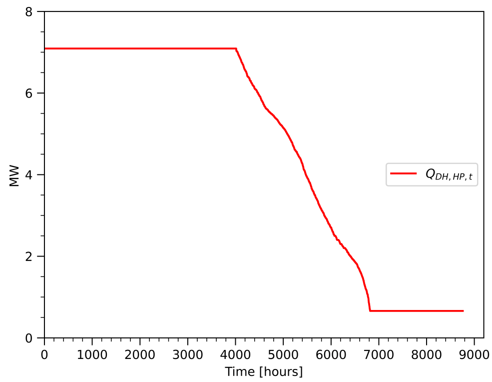



(b) Production of DH in the current system

Fig. 7.2: Hourly DH production




(a) DC production



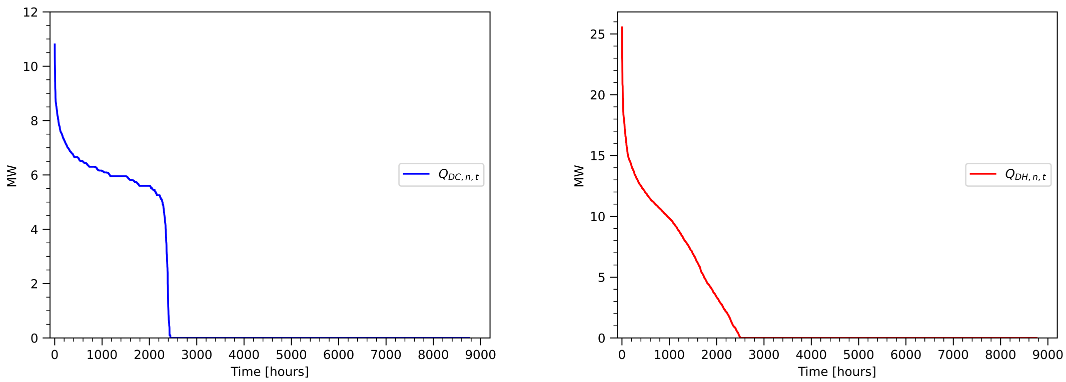

(b) DH production

Fig. 7.3: Load duration curves for DC and DH productions in the HP

is similar. The peak production reaches 25MW, and the decreasing of the load is made in a lineal tendency until the 2500 hours.

In this way, the HP serves as a complement to the DH supply and the DC supply of the current systems, supplying part of the demand and helping to reduce the operating hours of the current

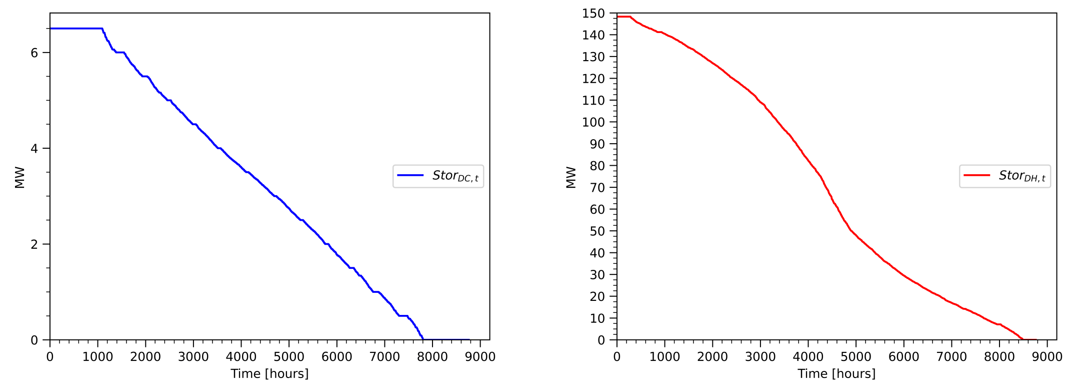
DH and DC supply systems and the operating costs of the system as a whole.




(a) DC production (b) DH production

Fig. 7.4: Load duration curves for DC and DH in the current systems

For the figure 7.5, the maximum capacity for both storage tanks can be obtained. The DC storage tank reaches a maximum capacity of 6.5MWh for more than 1100 hours, while the DH storage tank reaches a maximum capacity of 148MWh for just less than 500 hours. This great difference between both capacities is due to the large difference in values of both demands. However, both storage tanks are in use, with higher or lower storage level, for almost the whole year, with 7700 hours for the DC tank, and 8500 hours for the DH tank.



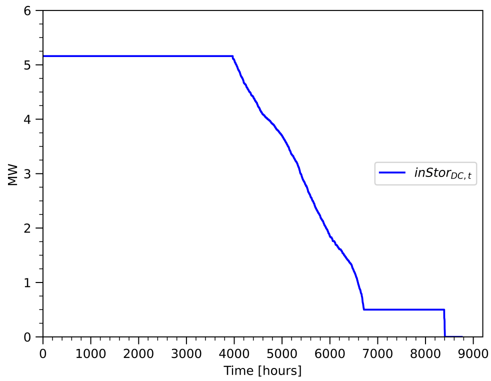

(a) DC storage tank (b) DH storage tank

Fig. 7.5: Load duration curves for the DC and DH level on the storage tanks

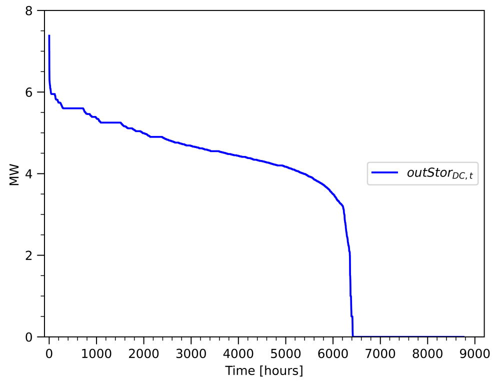
By studying the load duration curves of both DC and DH storage tanks, some conclusions can be drawn.

Firstly, for the DC storage tank, the load duration curves of the inlet and outlet flows of the storage tank can be seen in figure 7.6. If a comparison is made between both inlet and outlet flows and the load duration curve of the direct DC supply by HP, it is concluded that, most of the time, the cooling demand is supplied through the storage tank. In figure 7.6c, the output flow from the DC storage tank, matches almost perfectly the cooling demand during the winter months. The reason why the model decides to introduce the production from the HP to the tank at the same time that get the DC supply from the output of the tank instead of choosing for a direct supply is not clear. This may be the reason why the load duration curve for the DC direct supply is small, figure 7.7a.

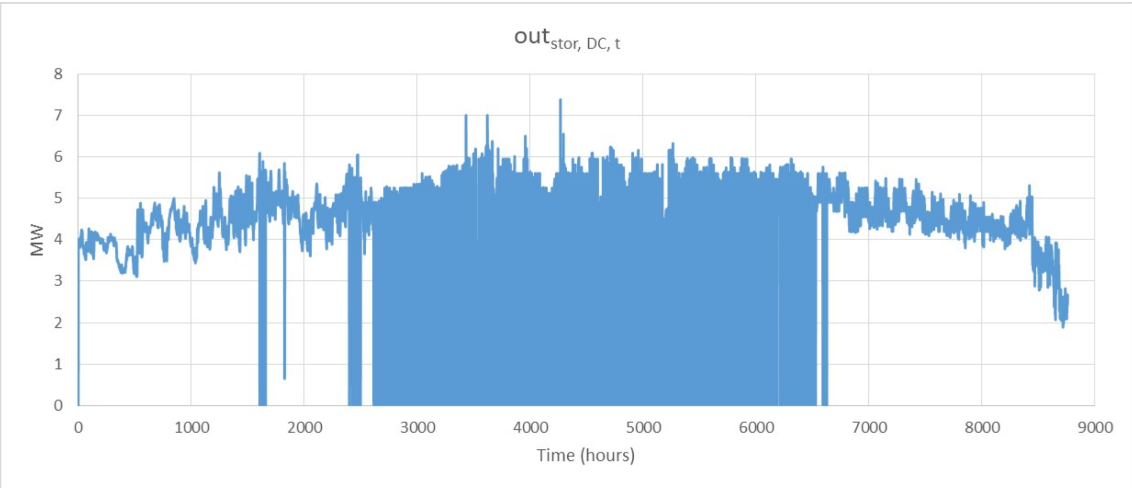
In figure 7.6a, the input flow reaches 4000 hours at full load. During those hours, the HP supplies its full load to the tank, 5.15MW. For the output flow, figure 7.6b, there is a peak of more than 7MW and, after that, the load decrease slowly and continuously until reaching around 6500 hours.




(a) Input flow for the DC storage tank in load duration curve



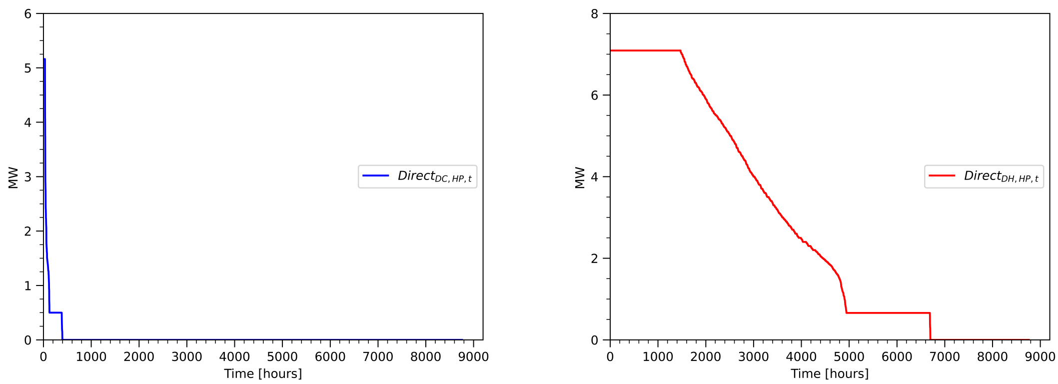

(b) Output flow for the DC storage tank in load duration curve




(c) Hourly output flow for the DC storage tank

Fig. 7.6: Load duration curves for DC storage tank for inlet and outlet and output flow in hourly representation

A similar study can be done for the DH storage tank. In this case, the inlet flow of the storage tank, figure 7.8a, reaches almost 4000 hours, being 1200 of those hours at full load. This means that all the DH production of the HP, 7 MW, is introduced into the storage tank during those 1200 hours. On the other hand, the output flow of the DH tank, figure 7.8b, barely reaches 4000 hours of operation. It can be seen that the output flow has a peak supply of almost 25 MW and then decreases considerably until reaching the value of zero in a constant tendency. This has an explanation. Because the HP has a reduced DH production capacity in comparison with the heating demand, the storage tank inlet flow must be much more constant and of longer duration to be able to supply, in a few time steps, the demand requested for the heating. For this reason, the size of the DH storage tank is much larger than the DC storage tank. During the hotter months, the HP does not stop to produce heating, and the heat produce goes into the storage tank, making necessary a large tank. The storage tank works as a help to the direct supply and the current heating system in order to match the heating demand. For this reason, the figure 7.6c has a similar shape than the heating demand with lower values.



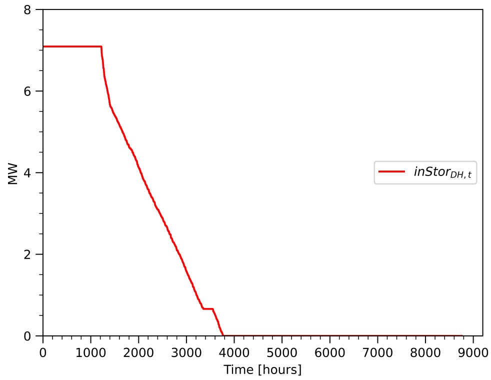

(a) Direct supply of DC (b) Direct supply of DH

Fig. 7.7: Load duration curves for direct supply of DC and DH

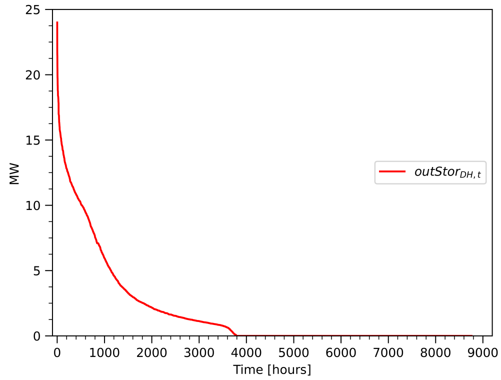
The final values of the overall system optimization can be seen in table 7.1.

The objective value of the model is the total annualised cost z. This takes a value of $3.412 \cdot 10^7$ DKK. From the results in the table, it can be seen that most of that cost comes from the production of the heating and cooling, which sum results in $3.1 \cdot 10^7$, a 90% of the annual cost of the system. The 10% rest corresponds to the annualised investment costs for the HP and storage tanks and the annual O&M costs.

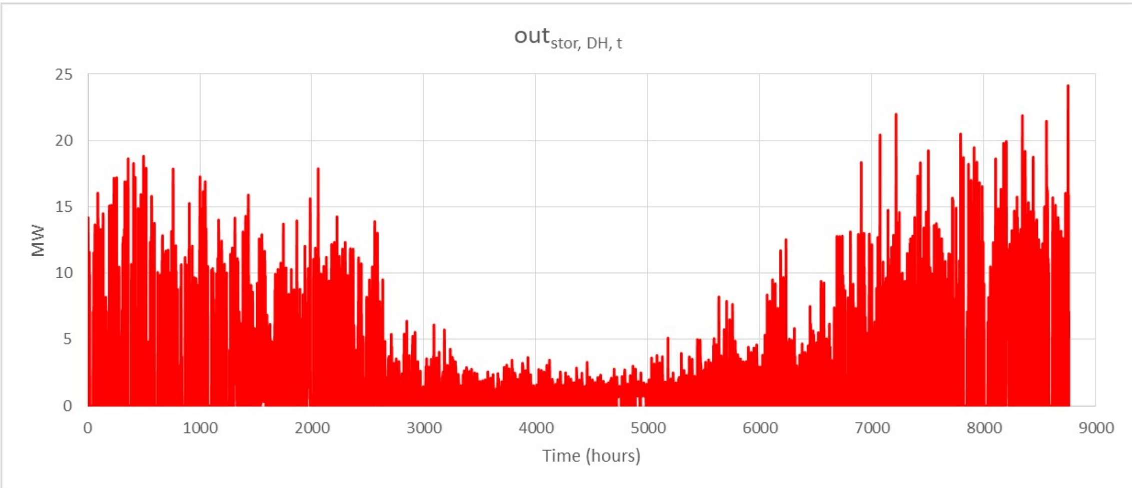
The total annual production of heating and cooling matches the total annual demand for each of them. The cooling demand through the year sums 44637 MWh, where 30000 MWh are produced by the HP and the rest of them by the current cooling system. On the other hand, the total heating demand for the year sums 61396 MWh, where 41170 MWh are produced by the HP and the current




(a) Input flow for the DH storage tank in load duration curve




(b) Input flow for the DH storage tank in load duration curve




(c) Hourly output flow for the DH storage tank

Fig. 7.8: Load duration curves for DC storage tank for inlet and outlet and output flow in hourly representation

heating system needs to supply the rest 20226 MWh. The total production and the demands are the exactly the same since the purpose of the model is to reduce the cost. The HP produces as much as the optimization model considers in order to reduce cost. If the total production by the HP does not cover all the demand, the rest of it is covered by the current heating and cooling system, but no more. This would be an unnecessary extra cost.

The capacity of the storage tanks were discussed before. If the capacity in MWh is converted into m$^3$, a 935 m$^3$ capacity storage tank is necessary for the DC. Transforming this magnitude into a cubic storage, with equal size lengths, the result is 9.8 meters. A different magnitude has the DH storage tank, with 6400 m$^3$, seven times bigger than the one for DC. This tanks would result in a 18.6 meters dice shape tank.

Table 7.1: Results for the optimization model

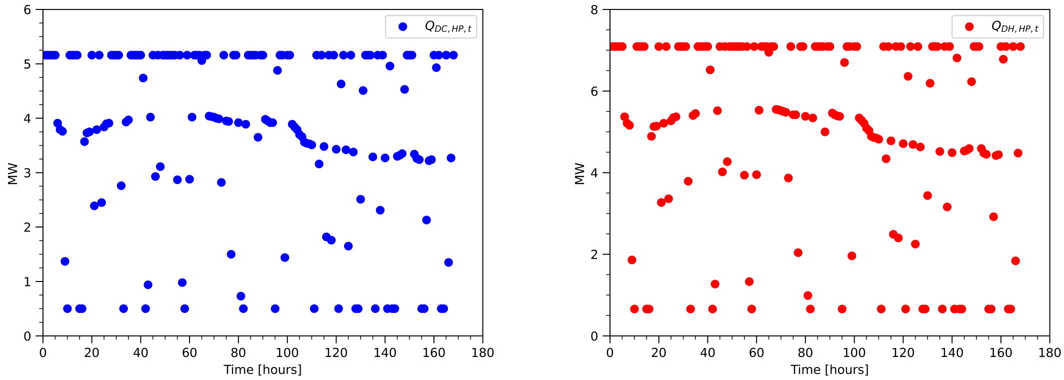
| Parameter | Unit | Value | Meaning |
|---|---|---|---|
| $z$ | DKK | $3{,}412 \cdot 10^7$ | Total cost of the co-production system |
| $d_{DC,\ total}$ | MWh | 44637 | Total annual demand of cooling |
| $d_{DH,\ total}$ | MWh | 61396 | Total annual demand of heating |
| $Q_{DC,\ HP,\ total}$ | MWh | 30000 | Total annual production of cooling by the HP |
| $Q_{DH,\ HP,\ total}$ | MWh | 41170 | Total annual production of heating by the HP |
| $Q_{DC,\ n,\ total}$ | MWh | 14643 | Total annual production of cooling by the current system |
| $Q_{DH,\ n,\ total}$ | MWh | 20226 | Total annual production of heating by the current system |
| $direct_{DC,\ HP,\ total}$ | MWh | 464,5 | Total annual direct supply of cooling by the HP |
| $direct_{DH,\ HP,\ total}$ | MWh | 24982 | Total annual direct supply of heating by the HP |
| $in_{stor,\ DC,\ total}$ | MWh | 29535 | Total annual flow into the DC storage tank |
| $in_{stor,\ DH,\ total}$ | MWh | 16187 | Total annual flow into the DH storage tank |
| $out_{stor,\ DC,\ total}$ | MWh | 29528 | Total annual flow out of the DC storage tank |
| $out_{stor,\ DH,\ total}$ | MWh | 16187 | Total annual flow out of the DH storage tank |
| $Q_{DC,\ max}$ | MW | 5,15 | Maximum capacity of the evaporator in the HP |
| $Q_{DH,\ max}$ | MW | 7,10 | Maximum capacity of the condenser in the HP |
| $hours_{HP,\ total}$ | h | 8760 | Total number of hours the HP is running |
| $hours_{DC,\ n,\ total}$ | h | 2443 | Total number of hours the DC current system is running |
| $c_{prod,\ DC,\ total}$ | DKK | $1{,}487 \cdot 10^7$ | Total cost of producing cooling |
| $c_{prod,\ DH,\ total}$ | DKK | $1{,}618 \cdot 10^7$ | Total cost of producing heating |
| $Q_{DC,\ stor}$ | MWh | 6,5 | Maximum capacity of the cooling storage tank |
| $Q_{DH,\ stor}$ | MWh | 148 | Maximum capacity of the heating storage tank |
| $Q_{DC,\ stor}$ | $m^3$ | 935 | Maximum capacity of the heating storage tank in $m^3$ |
| $Q_{DH,\ stor}$ | $m^3$ | 6398 | Maximum capacity of the heating storage tank in $m^3$ |

## 7.2 Weekly analysis

A weekly analysis is carried out in order to visualised the different operating points of the HP during a week.

As the HP load varies each hour, it is not easy to observe the different operating points through the year in figure 7.1a. This is the main reason of this section, to be able to see the variation of the load in the different time steps of the simulation. Two weeks have been selected in order to study the difference between the winter months, where the cooling demand is low, and the summer months, where the cooling demand is higher than the cooling capacity of the HP.

The first selected week is the second week of the year, and the operating points for the cooling and the heating production can be seen in figure 7.9. As it was explained before, the heating production depends directly of the cooling production. This is why the figure 7.9b has the same shape as figure 7.9a but with different values. The different values for the load in the HP goes from the maximum load 5.15MW until the minimum load imposed at which the HP is still in operation, 0.5MW. In the middle of the figure 7.9a, it can be observed some middle values that follow a tendency. This tendency matches with the demand values for cooling during these hours. In figure 7.10, the demand values for the same week are represented on top of the load variation.




(a) Operating points for DC production by the HP (b) Operating points for DH production by the HP

Fig. 7.9: Production of DC and DH in the 2nd week of the year

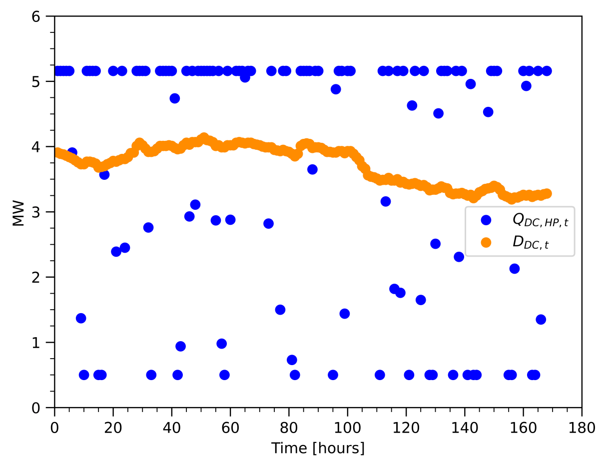
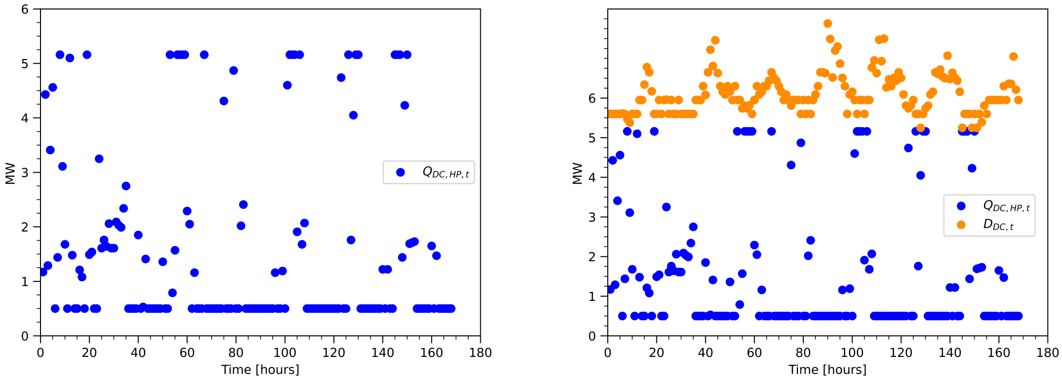



Fig. 7.10: DC production with the cooling demand for the 2nd week

The week to show the evolution during the summer months is the 26$^{th}$ week of the year, being this one the last week of June. In contrast to the winter week, during the summer week it can not be observed any tendency of the load that matches with the demand. The main reason of that is that the demand during summer is higher than the capacity of the HP, therefore, the cooling production of the HP is not enough to cover all the cooling demand. This, together with the low heating demand of those months, makes the HP load during those months lower than usual in the winter months. In figure 7.11a, it can be observed that the HP takes the lowest possible load values without shutting it down for the most of the hours during the week. It can be said that there is a slight tendency of the cooling load in the HP in some hours of the week. This tendency is in the lower part of the figure 7.11a, taking values of cooling production proportional to the cooling demand in those hours, the shape that these values is quite similar to the shape of the cooling demand points above. It was pointed out that one of the reasons of this phenomena is the low heating demand. The HP can not produce at full load during most of the time step since there is no heating demand that covers all the production.




(a) Operating points for DC production by the HP

(b) Operating points for DH production by the HP

Fig. 7.11: Production of DC and DH in the 26$^{th}$ week of the year

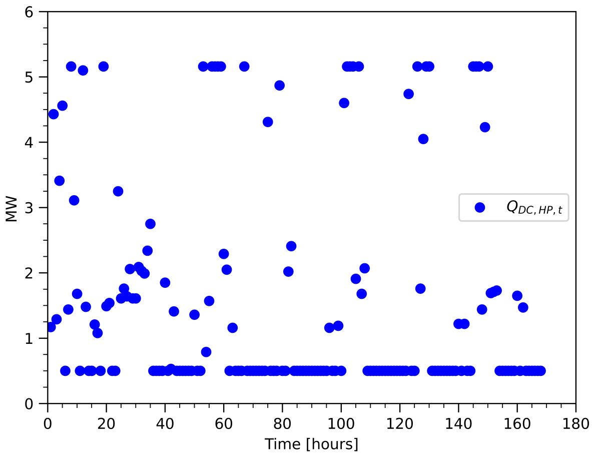



Fig. 7.12: DC production with the cooling demand for the 26$^{th}$ week

## 7.3 Sensitivity analysis

A variation of the electricity price is carried out in this section.

The effect of electricity price variation on the results of the optimization model has been studied. The range in which the price of electricity is considered goes from 800DKK/MWh to 1600DKK/MWh, being the design value initially chosen for the implementation of the model 1200DKK/MWh. The results of the influence of this variation can be seen in table 7.2.

As can be seen from the different values of the variables involved in the model, the variation in the price of electricity has its maximum influence on the result of the objective function of the model, the total costs of the heating and cooling production system. The capacity of the HP is not influenced by the price change. In addition, the total annual production of heating and cooling supplied by both the HP and the current supply system does not experience great changes in their values. The cooling and heating outputs in the HP when the price of electricity decreases ($Q_{\text{DC, HP, total}}$ and $Q_{\text{DH, HP, total}}$), experience an increase in 42MWh and 85 MWh respectively. If these amounts are compared with the magnitude of the total annual production, one can say that the increase in this case, of the annual production is practically insignificant. The increase in heating and cooling production by less than 100 MWh per year is less than 0.003%.

A similar analysis can be performed with the other variables indicated in table 7.2. The variation of their value with respect to the value obtained in the initial model is, as in the case of production in the HP, very low.

As a result, it is concluded that, the price of electricity has no major influence on the results of the optimization model other than increasing or reducing the total annual cost of the system. This is due to the simple fact that the electricity consumed is more expensive or cheaper, but it does not have a large influence on the magnitude of the total electricity consumption in the production system.

Table 7.2: Results for the electricity price variation

| **Parameter** | **Unit** | **Min Value** | **Design Value** | **Max Value** |
|---|---|---|---|---|
| $c_{\text{elect}}$ | DKK/MWh | 800 | 1200 | 1600 |
| z | DKK | $2.921 \cdot 10^7$ | $3.421 \cdot 10^7$ | $3.911 \cdot 10^7$ |
| $Q_{\text{DC, HP, total}}$ | MWh | 30042 | 30000 | 30033 |
| $Q_{\text{DH, HP, total}}$ | MWh | 41228 | 41170 | 41216 |
| $Q_{\text{DC, n, total}}$ | MWh | 14601 | 14643 | 14610 |
| $Q_{\text{DH, n, total}}$ | MWh | 20168 | 20226 | 20179 |
| $Q_{\text{DC, max}}$ | MW | 5,15 | 5,15 | 5,15 |
| $Q_{\text{DH, max}}$ | MW | 7,10 | 7,10 | 7,10 |
| $\text{hours}_{\text{HP, total}}$ | h | 8760 | 8760 | 8760 |
| $\text{hours}_{\text{DC, n, total}}$ | h | 2433 | 2443 | 2434 |
| $Q_{\text{DC, stor}}$ | MWh | 6,5 | 6,5 | 7 |
| $Q_{\text{DH, stor}}$ | MWh | 168 | 148 | 163 |
| $Q_{\text{DC, stor}}$ | $m^3$ | 935 | 935 | 1007 |
| $Q_{\text{DH, stor}}$ | $m^3$ | 7313 | 6398 | 7035 |

# Chapter 8

# Discussion

The project carried out during the development of this thesis has an interesting background. The inclusion of heat pumps in the production of district heating in recent years has increased in the Nordic countries, where the demand for heating during the cold months is high. This implies a shift of energy source towards the use of electricity. With the increase of renewable energy sources in the energy mix of the countries, this means a decrease of emissions in the production of district heating. The fact that this project tries to unify and simultaneously produce heating and cooling is a challenge. This is because the demand for district heating and district cooling is often not the same or comparable throughout the year. This is precisely what is happening on the Lyngby campus of DTU. While during the winter months, the demand for heating is high, the demand for cooling is much lower. On the other hand, in the summer months, the demand for cooling increases considerably while heating is at its annual minimum. It is not easy to meet these demands and therefore the production of heating and cooling.

Using a two-stage cycle for heat pump design makes performance better than using a single-stage cycle, because for a high temperature difference between evaporation and condensation, the two-stage cycle has a better response. In the design, the return from the district cooling is used as the heat source for the heat pump. The supply and return temperatures of the district cooling are in this case kept constant at 12°C and 18°C respectively, as design values, while the supply and return temperatures of the district heating have values of 70°C and 50°C respectively. After an analysis where the DC return temperature and the DH supply temperature vary, it is concluded that the DH has a greater impact on the HP performance.

When developing the optimization model, the dependence of the HP COP on the DH supply temperature is one of the input parameters. Although COP has a greater influence on the variation of the HP load, this makes the model non-linear, since the HP load is one of the decision variables. To try to match DH and DC output to the heat pump, the model includes the possibility of installing DH and DC storage tanks. In this way, the refrigeration production in those hours where the demand is low, is stored to be able to be consumed in later hours where the demand increases.

The same applies to heating. In addition, the model developed includes the possibility of producing DH and DC through the systems that currently exist on campus for this purpose. With all this, two different situations can be seen in the results.

Firstly, during the winter months, the production of refrigeration is supplied only by the production of the heat pump. While the heating production needs the support of the current system to meet the demand. Thus, during this period, the heat pump functions as a cost-saving method in the production of winter heating. In addition, the storage tank for DC is used to supply the stored energy at peak cooling times. During these months it is not necessary to provide cooling through the current production system.

On the other hand, during the summer months the situation is very different. The heating demand is drastically reduced, remaining at a minimum of approximately 2MW and 1.5MW. The demand for cooling is at the highest point of the year. This poses a problem for the operation of the heat pump, as it is a co-production system, if cooling is required and produced by the HP, heating will also be produced. This is why the DH storage tank is so large. All heating produced during the summer months that is not consumed, must be stored somewhere. It can also be seen that the heat pump output during these months is reduced as it is necessary to use the current system of refrigeration compressors to supply most of the demand during the summer. This is because it is not economically viable to store most of the heat produced for later consumption. This together makes the installed capacity of the heat pump similar to the average cooling demand during the year, although it does not make the use of compressors for DC production during the summer unnecessary.

As for the size of the storage tanks, these reach very different sizes, around 900 m$^3$ and 6400 m$^3$ for DC and DH respectively. These are of course large tanks. The installation of the DC tank, due to the volume it reaches, is more feasible than the installation of the DH tank. This does not mean that there is no DH storage of this size. Long-term DH storage is more and more common and has a larger capacity than the model.

## 8.1 Future work

After the realization of this project and the analysis of its results, the conclusion is reached of some modifications that can be carried out in future studies of the same situation.

As explained above, it is very difficult to match the demand for heating and cooling. This could be solved by studying the installation of a long-term storage, both for heating and cooling. Due to the geographical location of the Lyngby campus of the DTU, this option may be complicated by the large land requirements, but it would solve the problem of undemanding heating production during the summer and undemanding cooling production during the winter.

Another method could be to study the use of different heat pumps for heating and cooling production, so that the production of one does not depend on the production of another, or even the use of different heat sources for the HP. Even the use of reversible cycles can be analysed. This would mean that during the winter, only heating is provided, while in the summer, cooling is provided. In this hypothetical situation, the cooling demanded during the winter is not supplied and the same happens in summer with the heating. In this case, a system that combines the use of a small-scale HP for year-round co-production and a large-scale reversible HP can come up with a good solution to be studied.

# Chapter 9

# Conclusion

The main outcome of the development of this thesis is to find a solution that integrates a co-production heat pump for DH and DC supply at DTU's Lyngby university campus. The result obtained includes, besides the installation of a heat pump with a cooling production capacity of 5.15MW and a heating production capacity of 7.10MW, the installation of storage tanks for DC and DH of 935 $m^3$ and 6398 $m^3$ respectively, as well as the use of the existing production systems on the campus. The entire model developed includes operating costs in addition to O&M and investment in the new equipment.

The European project (Green Deal) of reduction of greenhouse gas emissions and higher final consumption of RES makes the investment in HP for the production of either DH, DC or both, a solution that helps to achieve the European objective.

In addition, coupling DH and DC production in a space where they are practically opposed throughout the year, is a difficult task using only co-production technology for a heat pump. It requires accurate data on heating and cooling demands and openness to study other possibilities for the design of production systems.

Despite these difficulties and the fact that an initial investment is required to install the new equipment, the heat pump is still a solution that lowers the costs of heating and cooling production on campus. This leads to the conclusion that heat pump installation is a good solution that can be studied in more detail and depth when supplying possible future increases in demand.

 *Conclusion*

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## Appendix A

# Appendix - Thermodynamic Models in EES

```
R$='Ammonia'

"Supply and return temperatures for DH and DC"
T_DH_s=343
T_DH_r=323
T_DC_s=285
T_DC_r=291

DeltaT_sh=3                                   "Superheating after evaporator"
DeltaT_sc=3                                   "Subcooling condenser"

"Pinch temperatures for evaporator and condenser"
DeltaT_evap=5
DeltaT_cond=5


k=1,3
vi=3
pii_bi=vi^k
n_is_opt=0,8                                  "Isentropic efficiency"
n_is=n_is_opt*( pr^((k-1)/k)-1)/(pii_bi ^((k-1)/k)-((k-1)/k)*pii_bi^(-1/k )*(pii_bi-pr)-1)

"-----CYCLE-----"

"State 1 - After Evaporator"

T[1]=T_DC_s-DeltaT_evap
p[1]=p_sat(R$;T=T[1])
x[1]=1
h[1]=enthalpy(R$;P=p[1];x=x[1])
s[1]=entropy(R$;P=p[1];x=x[1])

"Stage 2 - After Superheating"

T[2]=T[1]+DeltaT_sh
p[2]=p[1]
h[2]=enthalpy(R$;T=T[2];P=p[2])
s[2]=entropy(R$;T=T[2];P=p[2])

"State 3 - After Compressor"

s[30]=s[2]
T[30]=temperature(R$;P=p[3];s=s[30])
h[30]=enthalpy(R$;s=s[30];P=p[3])
h[3]=h[2]+(h[30]-h[2])/(n_is)
T[3]=temperature(R$;P=p[3];h=h[3])
p[3]=p[5]
s[3]=entropy(R$;T=T[3];P=P[3])

"State 4, 5 & 6 - Condenser"

T[5]=T[14]+DeltaT_cond
h[5]=enthalpy(R$;T=T[5];x=0)

p[6]=p[5]
h[6]=enthalpy(R$;T=T[6];P=p[5])

{T[6]=T_DH_r+DeltaT_cond                      "T[6] fixed by pinch temperature (DeltaT_cond) in condenser"
p[6]=pressure(R$;T=T[6]+T_subcool;x=0)
T_subcool=T[5]-T[6]}

T[6]=T_DH_r+DeltaT_pinch_sc                   "T[6] fixed by subcooling difference temperature (DeltaT_sc)"
p[6]=pressure(R$;T=T[6]+DeltaT_sc;x=0)
DeltaT_sc=T[5]-T[6]

h[4]=enthalpy(R$;P=p[5];x=1)
x_14=(h[4]-h[6])/(h[3]-h[6])
T[14]=(T_DH_s-T_DH_r)*x_14+T_DH_r
```

```
p[4]=p[5]
T[4]=T[5]
pr=p[3]/p[1]
s[4]=entropy(R$;T=T[4];x=1)
s[5]=entropy(R$;T=T[5];x=0)
s[6]=entropy(R$;T=T[6];P=P[6])

"State 7 - After Isoenthalpic Valve"

T[7]=T[1]
h[7]=h[6]
p[7]=p[1]
x[7]=quality(R$;T=T[7];h=h[7])
s[7]=entropy(R$;T=T[7];h=h[7])

"State 8 - Equal to State 1"

p[8]=p[1]
T[8]=T[1]
h[8]=h[1]
s[8]=s[1]

"UA-value evaporator"

exp(((T_DC_r-T[1])-(T_DC_s-T[1]))/DeltaT_lm_e)=(T_DC_r-T[1])/(T_DC_s-T[1])
UAe=Q_dot_e/DeltaT_lm_e
Q_dot_e=m_dot*(h[1]-h[7])
Q_dot_e=1000                                        "Design capacity in the evaporator"

"UA-values condenser"

x_15=(h[5]-h[6])/(h[3]-h[5])
T[15]=(T_DH_s-T_DH_r)*x_15+T_DH_r

exp(((T[3]-T[13])-(T[4]-T[14]))/DeltaT_lm_dsh)=(T[3]-T[13])/(T[4]-T[14])
exp(((T[5]-T[15])-(T[6]-T[16]))/DeltaT_lm_sc)=(T[5]-T[15])/(T[6]-T[16])
exp(((T[5]-T[15])-(DeltaT_cond))/DeltaT_lm_c)=(T[5]-T[15])/(DeltaT_cond)


UAdsh=(Q[3]-Q[4])/DeltaT_lm_dsh
UAc=(Q[4]-Q[5])/DeltaT_lm_c
UAsc=(Q[5]-Q[6])/DeltaT_lm_sc

UAcond=UAdsh+UAc+UAsc

COP_H=(h[3]-h[6])/(h[3]-h[2])
COP_C=(h[1]-h[7])/(h[3]-h[2])

Q_dot_DH=m_dot*(h[3]-h[6])

"Plots T-Q"

Q[3]=(h[3]-h[6])*m_dot
Q[4]=(h[4]-h[6])*m_dot
Q[5]=(h[5]-h[6])*m_dot
Q[6]=0

T[13]=T_DH_s
T[16]=T_DH_r
Q[13]=(h[3]-h[6])*m_dot
Q[14]=Q[4]
Q[15]=Q[5]
Q[16]=0

T[21]=T_DC_r
T[27]=T_DC_s
Q[21]=Q_dot_e
Q[27]=0
```

Q[1]=Q[21]
Q[7]=Q[27]
Q[1]=Q[8]

```
R$='Ammonia'

"Supply and return temperatures for DH and DC"
T_DH_s=343
T_DH_r=323
T_DC_s=285
"T_DC_r=291"


DeltaT_sh=3                                         "Superheating after evaporator"
DeltaT_sc=3                                         "Subcooling condenser"

"Pinch temperatures for evaporator and condenser"
"DeltaT_evap=5
DeltaT_cond=5"


k=1,3
vi=3
pii_bi=vi^k
n_is_opt=0,8                                        "Isentropic efficiency"
n_is=n_is_opt*( pr^((k-1)/k)-1)/(pii_bi ^((k-1)/k)-((k-1)/k)*pii_bi^(-1/k )*(pii_bi-pr)-1)
n=n_is/n_is_opt

"-----CYCLE-----"

"State 1 - After Evaporator"

T[1]=T_DC_s-DeltaT_evap
p[1]=p_sat(R$;T=T[1])
x[1]=1
h[1]=enthalpy(R$;P=p[1];x=x[1])
s[1]=entropy(R$;P=p[1];x=x[1])

"Stage 2 - After Superheating"

T[2]=T[1]+DeltaT_sh
p[2]=p[1]
h[2]=enthalpy(R$;T=T[2];P=p[2])
s[2]=entropy(R$;T=T[2];P=p[2])

"State 3 - After Compressor"

s[30]=s[2]
T[30]=temperature(R$;P=p[3];s=s[30])
h[30]=enthalpy(R$;s=s[30];P=p[3])
h[3]=h[2]+(h[30]-h[2])/(n_is)
T[3]=temperature(R$;P=p[3];h=h[3])
p[3]=p[5]
s[3]=entropy(R$;T=T[3];P=P[3])

"State 4, 5 & 6 - Condenser"

T[5]=T[14]+DeltaT_cond
h[5]=enthalpy(R$;T=T[5];x=0)

p[6]=p[5]
"h[6]=enthalpy(R$;T=T[6];P=p[5])"

{T[6]=T_DH_r+DeltaT_cond                            "T[6] fixed by pinch temperature (DeltaT_cond) in condenser"
p[6]=pressure(R$;T=T[6]+T_subcool;x=0)
T_subcool=T[5]-T[6]}

T[6]=T_DH_r+DeltaT_pinch_sc                         "T[6] fixed by subcooling difference temperature (DeltaT_sc)"
p[6]=pressure(R$;T=T[6]+DeltaT_sc;x=0)
DeltaT_sc=T[5]-T[6]

h[4]=enthalpy(R$;P=p[5];x=1)
```

```
x_14=(h[4]-h[6])/(h[3]-h[6])
T[14]=(T_DH_s-T_DH_r)*x_14+T_DH_r
p[4]=p[5]
T[4]=T[5]
pr=p[3]/p[1]
s[4]=entropy(R$;T=T[4];x=1)
s[5]=entropy(R$;T=T[5];x=0)
s[6]=entropy(R$;T=T[6];P=P[6])

"State 7 - After Isoenthalpic Valve"

T[7]=T[1]
h[7]=h[6]
p[7]=p[1]
x[7]=quality(R$;T=T[7];h=h[7])
s[7]=entropy(R$;T=T[7];h=h[7])

"State 8 - Equal to State 1"

p[8]=p[1]
T[8]=T[1]
h[8]=h[1]
s[8]=s[1]

"UA-value evaporator"

exp(((T_DC_r-T[1])-(T_DC_s-T[1]))/DeltaT_lm_e)=(T_DC_r-T[1])/(T_DC_s-T[1])
UAe=Q_dot_e/DeltaT_lm_e
Q_dot_e=m_dot*(h[1]-h[7])
Q_dot_e=1000
Qd=1000                                     "Design capacity in the evaporator"
part=Q_dot_e/Qd

UAe=UAed
UAed=131,4


"UA-values condenser"

x_15=(h[5]-h[6])/(h[3]-h[5])
T[15]=(T_DH_s-T_DH_r)*x_15+T_DH_r

exp(((T[3]-T[13])-(T[4]-T[14]))/DeltaT_lm_dsh)=(T[3]-T[13])/(T[4]-T[14])
exp(((T[5]-T[15])-(T[6]-T[16]))/DeltaT_lm_sc)=(T[5]-T[15])/(T[6]-T[16])
exp(((T[5]-T[15])-(DeltaT_cond))/DeltaT_lm_c)=(T[5]-T[15])/(DeltaT_cond)


UAdsh=(Q[3]-Q[4])/DeltaT_lm_dsh
"UAc=(Q[4]-Q[5])/DeltaT_lm_c"
UAsc=(Q[5]-Q[6])/DeltaT_lm_sc

UAdsh=UAdshd*(part)^0,8
UAc=UAcd*(part)^0,8
UAsc=UAscd*(part)^0,8


UAdshd=11
UAcd=94,13
UAscd=0,9699

UAcond=UAdsh+UAc+UAsc

COP_H=(h[3]-h[6])/(h[3]-h[2])
COP_C=(h[1]-h[7])/(h[3]-h[2])

Q_dot_DH=m_dot*(h[3]-h[6])

"Plots T-Q"
```

```
Q[3]=(h[3]-h[6])*m_dot
Q[4]=(h[4]-h[6])*m_dot
Q[5]=(h[5]-h[6])*m_dot
Q[6]=0

T[13]=T_DH_s
T[16]=T_DH_r
Q[13]=(h[3]-h[6])*m_dot
Q[14]=Q[4]
Q[15]=Q[5]
Q[16]=0

T[21]=T_DC_r
T[27]=T_DC_s
Q[21]=Q_dot_e
Q[27]=0
Q[1]=Q[21]
Q[7]=Q[27]
Q[1]=Q[8]
```

```
R$='Ammonia'

"Supply and return temperatures in DH and DC"
T_DH_s=343
T_DH_r=323
T_DC_s=285
T_DC_r=291

DeltaT_sh=3                                        "Superheating"
DeltaT_sc=3                                        "subcooling"

"Pinch difference temperatures in evaporator and condenser"
DeltaT_evap=5
DeltaT_cond=5


"Isentropic efficiency"
k=1,3
vi=2
pii_bi=vi^k
n_is_opt=0,8
n_is_low=n_is_opt*( pr_low^((k-1)/k)-1)/(pii_bi ^((k-1)/k)-((k-1)/k)*pii_bi^(-1/k )*(pii_bi-pr_low)-1)
n_is_high=n_is_opt*( pr_high^((k-1)/k)-1)/(pii_bi ^((k-1)/k)-((k-1)/k)*pii_bi^(-1/k )*(pii_bi-pr_high)-1)


"-----CYCLE-----"

"State 1 - After Evaporator"

T[1]=T_DC_s-DeltaT_evap
p[1]=p_sat(R$;T=T[1])
x[1]=1
h[1]=enthalpy(R$;P=p[1];x=x[1])
s[1]=entropy(R$;P=p[1];x=x[1])

"Stage 2 - After Superheating"

T[2]=T[1]+DeltaT_sh
p[2]=p[1]
h[2]=enthalpy(R$;T=T[2];P=p[2])
s[2]=entropy(R$;T=T[2];P=p[2])

"State 3 - After Compressor"

s[30]=s[2]
T[30]=temperature(R$;P=p[3];s=s[30])
h[30]=enthalpy(R$;s=s[30];P=p[3])
h[3]=h[2]+(h[30]-h[2])/(n_is_low)
T[3]=temperature(R$;P=p[3];h=h[3])
s[3]=entropy(R$;T=T[3];h=h[3])

pr_low=p[3]/p[1]

"State 4 - After Intercooler / Before Compressor (High Stage)"

p[3]=p[9]
p[4]=p[3]
p[5]=p[7]
p[9]=sqrt(p[1]*p[7])
"p[9]=1140"

x[4]=1
h[4]=enthalpy(R$;P=p[4];x=x[4])
T[4]=temperature(R$;P=P[4];x=x[4])
s[4]=entropy(R$;x=x[4];P=P[4])

"State 5 - After Compressor (High Stage)"
```

```
s[50]=s[4]
T[50]=temperature(R$;P=p[5];s=s[50])
h[50]=enthalpy(R$;s=s[50];P=p[5])
h[5]=h[4]+(h[50]-h[4])/(n_is_high)
T[5]=temperature(R$;P=p[5];h=h[5])
s[5]=entropy(R$;T=T[5];h=h[5])

pr_high=p[5]/p[4]

"State 6, 7 & 8 - After Condenser"

T[7]=T[16]+DeltaT_cond

"p[7]=p_sat(R$;T=T[7])"
x[7]=0
h[7]=enthalpy(R$;T=T[7];x=0)
p[8]=p[7]
h[8]=enthalpy(R$;T=T[8];P=p[7])

T[8]=T_DH_r+DeltaT_pinch_sc                  "T[8] fixed by pinch difference temperature in condenser (
DeltaT_cond)"
p[8]=pressure(R$;T=T[8]+DeltaT_sc;x=0)
DeltaT_sc=T[7]-T[8]

h[6]=enthalpy(R$;P=p[8];x=1)
x_16=(h[6]-h[8])/(h[5]-h[8])
T[16]=(T_DH_s-T_DH_r)*x_16+T_DH_r
p[6]=p[7]
T[6]=T[7]

s[6]=entropy(R$;x=1;P=P[6])
s[7]=entropy(R$;x=0;P=P[7])
s[8]=entropy(R$;T=T[8];P=p[8])

"Stage 9 - After 1st Valve"

h[8]=h[9]
x[9]=quality(R$;P=p[9];h=h[9])
T[9]=temperature(R$;P=p[9];h=h[9])
s[9]=entropy(R$;T=T[9];h=h[9])

"Stage 10 - After Intercooler - Before 2nd Valve"
p[10]=p[9]
x[10]=0
T[10]=temperature(R$;x=x[10];P=p[10])
h[10]=enthalpy(R$;P=p[10];x=x[10])
s[10]=entropy(R$;T=T[10];x=x[10])

"Stage 11- After 2nd Valve"

h[10]=h[11]
p[11]=p[1]
T[11]=T[1]
x[11]=quality(R$;P=P[11];h=h[11])
s[11]=entropy(R$;T=T[11];h=h[11])

"Stage 12 - Equal to Stage 1"
T[12]=T[1]
h[12]=h[1]
p[12]=p[1]
s[12]=s[1]

Q_dot_DH=m_dot_H*(h[5]-h[8])
h[10]*m_dot_L=(h[3]*m_dot_L)-(h[4]*m_dot_H)+(h[9]*m_dot_H)

COP_H=(h[5]-h[8])/((h[5]-h[4])+(h[3]-h[2]))
COP_C=(h[1]-h[11])/((h[5]-h[4])+(h[3]-h[2]))
```

```
"UA-value evaporator"
exp(((T_DC_r-T[1])-(T_DC_s-T[1]))/DeltaT_lm_e)=(T_DC_r-T[1])/(T_DC_s-T[1])
UAe=Q_dot_e/DeltaT_lm_e
Q_dot_e=m_dot_L*(h[1]-h[11])
Q_dot_e=1000


"UA-values condenser"
x_17=(h[7]-h[8])/(h[5]-h[8])
T[17]=(T_DH_s-T_DH_r)*x_17+T_DH_r

exp(((T[5]-T[15])-(DeltaT_cond))/DeltaT_lm_dsh)=(T[5]-T[15])/(DeltaT_cond)
exp(((T[7]-T[17])-(T[8]-T[18]))/DeltaT_lm_sc)=(T[7]-T[17])/(T[8]-T[18])
exp(((T[7]-T[17])-(DeltaT_cond))/DeltaT_lm_c)=(T[7]-T[17])/(DeltaT_cond)

UAdsh=(Q[5]-Q[6])/DeltaT_lm_dsh
UAc=(Q[6]-Q[7])/DeltaT_lm_c
UAsc=(Q[7]-Q[8])/DeltaT_lm_sc

UAcond=UAdsh+UAc+UAsc

"Plots"
Q[5]=(h[5]-h[8])*m_dot_H
Q[6]=(h[6]-h[8])*m_dot_H
Q[7]=(h[7]-h[8])*m_dot_H
Q[8]=0

T[15]=T_DH_s
T[18]=T_DH_r
Q[15]=(h[5]-h[8])*m_dot_H
Q[18]=0
Q[16]=Q[6]
Q[17]=Q[7]


T[21]=T_DC_r
T[27]=T_DC_s
Q[21]=Q_dot_e
Q[27]=0
Q[12]=Q[21]
Q[11]=0
```

```
R$='Ammonia'

"Supply and return temperatures in DH and DC"
"T_DH_s=343"
T_DH_r=323
T_DC_s=285
T_DC_r=291

DeltaT_sh=3     "Superheating"
DeltaT_sc=3     "subcooling"

"Pinch difference temperatures in evaporator and condenser"
{DeltaT_evap=5
DeltaT_cond=5}

"Isentropic efficiency"
k=1,3
vi=2
pii_bi=vi^k
n_is_opt=0,8
n_is_low=n_is_opt*( pr_low^((k-1)/k)-1)/(pii_bi ^((k-1)/k)-((k-1)/k)*pii_bi^(-1/k )*(pii_bi-pr_low)-1)
n_is_high=n_is_opt*( pr_high^((k-1)/k)-1)/(pii_bi ^((k-1)/k)-((k-1)/k)*pii_bi^(-1/k )*(pii_bi-pr_high)-1)
n_low=n_is_low/n_is_opt
n_high=n_is_high/n_is_opt

"-----CYCLE-----"

"State 1 - After Evaporator"

T[1]=T_DC_s-DeltaT_evap
p[1]=p_sat(R$;T=T[1])
x[1]=1
h[1]=enthalpy(R$;P=p[1];x=x[1])
s[1]=entropy(R$;P=p[1];x=x[1])

"Stage 2 - After Superheating"

T[2]=T[1]+DeltaT_sh
p[2]=p[1]
h[2]=enthalpy(R$;T=T[2];P=p[2])
s[2]=entropy(R$;T=T[2];P=p[2])

"State 3 - After Compressor"

s[30]=s[2]
T[30]=temperature(R$;P=p[3];s=s[30])
h[30]=enthalpy(R$;s=s[30];P=p[3])
h[3]=h[2]+(h[30]-h[2])/(n_is_low)
T[3]=temperature(R$;P=p[3];h=h[3])
s[3]=entropy(R$;T=T[3];h=h[3])

pr_low=p[3]/p[1]

"State 4 - After Intercooler / Before Compressor (High Stage)"

p[3]=p[9]
p[4]=p[3]
p[5]=p[7]
"p[9]=sqrt(p[1]*p[7])"
p[9]=1140

x[4]=1
h[4]=enthalpy(R$;P=p[4];x=x[4])
T[4]=temperature(R$;P=P[4];x=x[4])
s[4]=entropy(R$;x=x[4];P=P[4])

"State 5 - After Compressor (High Stage)"
```

s[50]=s[4]
T[50]=**temperature**(R$;***P***=p[5];***s***=s[50])
h[50]=**enthalpy**(R$;***s***=s[50];***P***=p[5])
h[5]=h[4]+(h[50]-h[4])/(n_is_high)
T[5]=**temperature**(R$;***P***=p[5];***h***=h[5])
s[5]=**entropy**(R$;***T***=T[5];***h***=h[5])

pr_high=p[5]/p[4]

"State 6, 7 & 8 - After Condenser"

T[7]=T[16]+DeltaT_cond

"p[7]=p_sat(R$;T=T[7])"
x[7]=0
h[7]=**enthalpy**(R$;***T***=T[7];***x***=0)
p[8]=p[7]
"h[8]=enthalpy(R$;T=T[8];P=p[7])"

T[8]=T_DH_r+DeltaT_pinch_sc "T[8] fixed by subcooling difference temperature (DeltaT_sc)"
p[8]=**pressure**(R$;***T***=T[8]+DeltaT_sc;***x***=0)
DeltaT_sc=T[7]-T[8]

h[6]=**enthalpy**(R$;***P***=p[8];***x***=1)
x_16=(h[6]-h[8])/(h[5]-h[8])
T[16]=(T_DH_s-T_DH_r)*x_16+T_DH_r
p[6]=p[7]
T[6]=T[7]

s[6]=**entropy**(R$;***x***=1;***P***=P[6])
s[7]=**entropy**(R$;***x***=0;***P***=P[7])
s[8]=**entropy**(R$;***T***=T[8];***P***=p[8])

"Stage 9 - After 1st Valve"

h[8]=h[9]
x[9]=**quality**(R$;***P***=p[9];***h***=h[9])
T[9]=**temperature**(R$;***P***=p[9];***h***=h[9])
s[9]=**entropy**(R$;***T***=T[9];***h***=h[9])

"Stage 10 - After Intercooler - Before 2nd Valve"
p[10]=p[9]
x[10]=0
T[10]=**temperature**(R$;***x***=x[10];***P***=p[10])
h[10]=**enthalpy**(R$;***P***=p[10];***x***=x[10])
s[10]=**entropy**(R$;***T***=T[10];***x***=x[10])

"Stage 11- After 2nd Valve"

h[10]=h[11]
p[11]=p[1]
T[11]=T[1]
x[11]=**quality**(R$;***P***=P[11];***h***=h[11])
s[11]=**entropy**(R$;***T***=T[11];***h***=h[11])

"Stage 12 - Equal to Stage 1"
T[12]=T[1]
h[12]=h[1]
p[12]=p[1]
s[12]=s[1]

Q_dot_DH=m_dot_H*(h[5]-h[8])
h[10]*m_dot_L=(h[3]*m_dot_L)-(h[4]*m_dot_H)+(h[9]*m_dot_H)

COP_H=(h[5]-h[8])/((h[5]-h[4])+(h[3]-h[2]))
COP_C=(h[1]-h[11])/((h[5]-h[4])+(h[3]-h[2]))

"UA-value evaporator"

```
exp(((T_DC_r-T[1])-(T_DC_s-T[1]))/DeltaT_lm_e)=(T_DC_r-T[1])/(T_DC_s-T[1])
UAe=Q_dot_e/DeltaT_lm_e
Q_dot_e=m_dot_L*(h[1]-h[11])
Q_dot_e=1000
Qd=1000
part=Q_dot_e/Qd

UAe=UAed
UAed=131,4


"UA-values condenser"
x_17=(h[7]-h[8])/(h[5]-h[8])
T[17]=(T_DH_s-T_DH_r)*x_17+T_DH_r

exp(((T[5]-T[15])-(DeltaT_cond))/DeltaT_lm_dsh)=(T[5]-T[15])/(DeltaT_cond)
exp(((T[7]-T[17])-(T[8]-T[18]))/DeltaT_lm_sc)=(T[7]-T[17])/(T[8]-T[18])
exp(((T[7]-T[17])-(DeltaT_cond))/DeltaT_lm_c)=(T[7]-T[17])/(DeltaT_cond)

UAdsh=(Q[5]-Q[6])/DeltaT_lm_dsh
"UAc=(Q[6]-Q[7])/DeltaT_lm_c"
UAsc=(Q[7]-Q[8])/DeltaT_lm_sc

UAdsh=UAdshd*(part)^0,8
UAc=UAcd*(part)^0,8
UAsc=UAscd*(part)^0,8

UAdshd=10,89
UAcd=99,92
UAscd=0,996

UAcond=UAdsh+UAc+UAsc

"Plots"
Q[5]=(h[5]-h[8])*m_dot_H
Q[6]=(h[6]-h[8])*m_dot_H
Q[7]=(h[7]-h[8])*m_dot_H
Q[8]=0

T[15]=T_DH_s
T[18]=T_DH_r
Q[15]=(h[5]-h[8])*m_dot_H
Q[18]=0
Q[16]=Q[6]
Q[17]=Q[7]


T[21]=T_DC_r
T[27]=T_DC_s
Q[21]=Q_dot_e
Q[27]=0
Q[12]=Q[21]
Q[11]=0
```

 Appendix - Thermodynamic Models in EES

## Appendix B

# Appendix - Optimization Model in GAMS

```
sets
t                     time periods    /t1*t8760/
;

scalars
T_DH_r                DH return temperature /45/
T_DC_s                DC supply temperature /12/
T_DC_r                DC return temperature /18/

n                     number of year the simulation is carried out over /25/

i_initial             initial level 0 or 1 of the HP /1/
in_initial            initial level 0 or 1 of the cooling plants /0/
stor_DC_initial       initial level of the DC storage /0/
stor_DH_initial       initial level of the DH storage /0/

Q_DC_prod_min         min load of HP/0.5/
Q_DC_prod_max         max load of HP/100/

Q_DCn_prod_min         min load of HP/0.1/
Q_DCn_prod_max         max load of HP/100/

c_fixed               fixed investment cost for co-production capacity (DKK) /15952685/
c_size                size dependent investment cost for co-production capacity (DKK per MW) /4031493/

c_gas_heating         cost of producing MWh of heating in the current system (DKK) /800/
COP_C_Comp            COP cooling of the current system /4.25/
c_OM_Comp             cost of O&M of the current cooling plants (DKK per hour of operating) /2/
c_OM_HP_fixed         cost of O&M of HP fixed per year (DKK)/270/
c_OM_HP_prod          cost of O&M of HP heating produced (DKK per MWh) /0.29/
c_elect               cost of MWh of electricity (DKK) /1200/

dT_DH                 difference DH /20/
dT_DC                 difference DC /6/

cost_tanksize_DC      cost dependent on size (DKK per m^3) /1500/
cost_tanksize_DH      cost dependent on size (DKK per m^3) /1500/
rho                   density /998/

cp                    specific heat kJ per kg and K /4.180/
convert               conversion factor kJ to MWh /0.000000277777778/
cost_stor_DC          cost dependent DC on size (DKK per MWh)
cost_stor_DH          cost dependent DH on size (DKK per MWh)

k                     discounting factor /0.04/
a                     annuity present value factor
;

a = (1-(1+k)**(-n))/k;

cost_stor_DC = cost_tanksize_DC/(rho*cp*convert*(dT_DC));
cost_stor_DH = cost_tanksize_DH/(rho*cp*convert*(dT_DH));

parameters
d_DC(t)               MWh per hour : hourly cooling demand
/
$ondelim
$include cooling demand.csv
$offdelim
/

d_DH(t)               MWh per hour : hourly heating demand
/
$ondelim
$include heating demand.csv
$offdelim
/

t_outside(t)          temperature outside in Celsius degrees
/
$ondelim
```

```
$include temp outside.csv
$offdelim
/

T_DH_s(t) - :         supply temperature for DH
COP_H_HP(t)           COP for heating in HP
COP_C_HP(t)           COP for cooling in HP
;
;
T_DH_s(t)$(t_outside(t) >= 4.5) = 343;
T_DH_s(t)$((-10 < t_outside(t)) and (t_outside(t) < 4.5)) = -1.4584*t_outside(t)+349.612;
T_DH_s(t)$(t_outside(t)<= -10) = 363;

COP_H_HP(t)=20.9174-0.0510138*T_DH_s(t);
COP_C_HP(t)=19.932-0.0482713*T_DH_s(t);

variables

Q_DC_HP(t)            MWh: DC production in HP
Q_DH_HP(t)            MWh: DH production in HP

total_DH_prod         MWh: total production of DH in the HP

Q_DC_n(t)             MWh: DC production in actual system
Q_DH_n(t)             MWh: DH production in actual system

stor_DC(t)            MWh: level of DC storage
in_stor_DC(t)         MWh: level of DC storage going in
out_stor_DC(t)        MWh: level of DC storage going out
stor_DH(t)            MWh: level of DH storage
in_stor_DH(t)         MWh: level of DH storage going in
out_stor_DH(t)        MWh: level of DH storage going out

direct_DC_HP(t)       MWh:direct production of the HP for cooling purposes
direct_DH_HP(t)       MWh:direct production of the HP for heating purposes

Q_DC_max              MWh: maximum cooling production capacity rating (the maximum attained)
stor_DC_max           MWh: max tank size DC
stor_DH_max           MWh: max tank size DH

i(t)                  -: indicates whether the HP is running at hour t
u(t)                  -: indicates if the HP is turning on at time t
total_hoursDCn        h: operating total hours of DC n

in(t)                  -: indicates whether the cooling plants are running at hour t
un(t)                  -: indicates if the cooling plants are turning on at time t

c_prod_DC(t)          DKK: production cost of DC
c_prod_DH(t)          DKK: production cost of DH

c_OM                  DKK: total cost of O&M for the production systems


inv_prod              DKK : production capacity investment and maintenance cost
inv_stor_DC           DKK: investment DC tank
inv_stor_DH           DKK: investment DH tank

z                     DKK : total cost
;

free variables
z;
positive variables
Q_DC_HP(t)
Q_DH_HP(t)
Q_DC_n(t)
Q_DH_n(t)
total_DH_prod

stor_DC(t)
in_stor_DC(t)
```

```
out_stor_DC(t)
stor_DH(t)
in_stor_DH(t)
out_stor_DH(t)

u(t)
un(t)
total_hoursDCn

direct_DC_HP(t)
direct_DH_HP(t)

Q_DC_max
stor_DC_max
stor_DH_max

c_OM
inv_prod
inv_stor_DC
inv_stor_DH
;
binary variables
i(t)
in(t)
;

Equations

cost_total          total plant cost
prod_cost_DC        cost of DC production
prod_cost_DH        cost of DH production
prod_inv            investment cost for production capacity
cost_OM             yearly costs of O&M
DH_prod_HP          DH production equation in the HP
DH_demand           total demand of DH
DC_demand           total demand of DC

DC_HP_prod          total production of DC in HP
DH_HP_prod          total production of DH in HP
totalDH_prod        total production of DH in the year by the HP

startup             startups of the HP
startupn            startups of the cooling plants
hoursDCn            total operating hours DC cooling plants
min_prod_DC         min production of DC in the HP
max_prod_DC         max production of DC in the HP
min_prod_DCn        min production of DC
max_prod_DCn        max production of DC

Q_DC_max_cap        capacity of the evaporator in the HP

DC_stor_level(t)    level of the DC tank
DH_stor_level(t)    level of the DH tank

stor_DC_max_cap     max capacity DC tank
stor_DH_max_cap     max capacity DH tank

stor_inv_DC         investment cost tank DC
stor_inv_DH         investment cost tank DH
;


cost_total..
z=e=sum(t,c_prod_DC(t)+c_prod_DH(t))+c_OM+(inv_stor_DC+inv_stor_DH+inv_prod)/a;
prod_cost_DC(t)..
c_prod_DC(t)=e=((Q_DC_n(t)/COP_C_Comp)+(Q_DC_HP(t)/COP_C_HP(t)))*c_elect;
prod_cost_DH(t)..        c_prod_DH(t)=e=(Q_DH_n(t)*c_gas_heating);
cost_OM..
c_OM=e=c_OM_Comp*total_hoursDCn+(c_OM_HP_fixed+c_OM_HP_prod*total_DH_prod);
prod_inv..               inv_prod=e=c_fixed+c_size*Q_DC_max;
DC_demand(t)..           d_DC(t)=e=direct_DC_HP(t)+out_stor_DC(t)+Q_DC_n(t);
DH_prod_HP(t)..          Q_DH_HP(t)=e= -0.0254+1.37975*Q_DC_HP(t);
DH_demand(t)..           d_DH(t)=e=direct_DH_HP(t)+out_stor_DH(t)+Q_DH_n(t);
```

```
DC_HP_prod(t)..           Q_DC_HP(t)=e=direct_DC_HP(t)+in_stor_DC(t);
DH_HP_prod(t)..           Q_DH_HP(t)=e=direct_DH_HP(t)+in_stor_DH(t);

min_prod_DC(t)..          Q_DC_HP(t)=g=Q_DC_prod_min*i(t);
max_prod_DC(t)..          Q_DC_HP(t)=l=Q_DC_prod_max*i(t);

min_prod_DCn(t)..         Q_DC_n(t)=g=Q_DCn_prod_min*in(t);
max_prod_DCn(t)..         Q_DC_n(t)=l=Q_DCn_prod_max*in(t);
totalDH_prod..            total_DH_prod=e=sum(t,Q_DH_HP(t));

startup(t)..              (i(t)-i_initial)$(ord(t)=1)+(i(t)-i(t-1))$(ord(t)>1)=l=u(t);
startupn(t)   ..
(in(t)-in_initial)$(ord(t)=1)+(in(t)-in(t-1))$(ord(t)>1)=l=un(t);
hoursDCn..                total_hoursDCn=e=sum(t,in(t));

DC_stor_level(t)..
stor_DC(t)=e=(stor_DC_initial+in_stor_DC(t)-out_stor_DC(t))$(ord(t)=1)+(stor_DC(t-1)+i
n_stor_DC(t)-out_stor_DC(t))$(ord(t)>1);
DH_stor_level(t)..
stor_DH(t)=e=(stor_DH_initial+in_stor_DH(t)-out_stor_DH(t))$(ord(t)=1)+(stor_DH(t-1)+i
n_stor_DH(t)-out_stor_DH(t))$(ord(t)>1);

Q_DC_max_cap(t)..         Q_DC_max =g= Q_DC_HP(t);
stor_DC_max_cap(t)..      stor_DC_max =g= stor_DC(t);
stor_DH_max_cap(t)..      stor_DH_max =g= stor_DH(t);

stor_inv_DC..             inv_stor_DC=e=cost_stor_DC*stor_DC_max;
stor_inv_DH..             inv_stor_DH=e=cost_stor_DH*stor_DH_max;


model basemodel /all/;

solve basemodel using mip minimizing z;

file results /results.csv/;
put results;
loop (t, put t.tl d_DC(t) d_DH(t)
         put Q_DC_HP.l(t) Q_DH_HP.l(t) Q_DC_n.l(t) Q_DH_n.l(t) direct_DC_HP.l(t)
         direct_DH_HP.l(t) in_stor_DC.l(t) in_stor_DH.l(t) out_stor_DC.l(t)
         out_stor_DH.l(t) stor_DC.l(t) stor_DH.l(t)/);

parameter
totalDC_demand            MWh: total demand of DC
totalDH_demand            MWh: total demand of DH
totalDC_HP_prod           MWh: total production of DC in HP
totalDH_HP_prod           MWh: total production of DH in HP
totalDC_n_prod            MWh: total production of DC in current system
totalDH_n_prod            MWh: total production of DH in current system
totalDC_direct            MWh: total direct supply of DC by HP
totalDH_direct            MWh: total direct supply of DH by HP
totalDC_instor            MWh: total input flow to the DC tank
totalDH_instor            MWh: total input flow to the DH tank
totalDC_outstor           MWh: total output flow from the DC tank
totalDH_outstor           MWh: total output flow from the DC tank
totalDC_cprod             DKK: total cost of cooling production
totalDH_cprod             DKK: total cost of heating production

total_hoursHP             hours: total running hours for the HP

storDCm3                  m3: capacity in m3 of DC storage tank
storDHm3                  m3: capacity in m3 of DH storage tank


;
totalDC_demand = sum(t, d_DC(t));
totalDH_demand = sum(t, d_DH(t));
totalDC_HP_prod = sum(t, Q_DC_HP.l(t));
totalDH_HP_prod = sum(t, Q_DH_HP.l(t));
totalDC_n_prod = sum(t, Q_DC_n.l(t));
totalDH_n_prod = sum(t, Q_DH_n.l(t));
totalDC_direct = sum(t, direct_DC_HP.l(t));
totalDH_direct = sum(t, direct_DH_HP.l(t));
```

```
totalDC_instor = sum(t, in_stor_DC.l(t));
totalDH_instor = sum(t, in_stor_DH.l(t));
totalDC_outstor = sum(t, out_stor_DC.l(t));
totalDH_outstor = sum(t, out_stor_DH.l(t));
totalDC_cprod = sum(t, c_prod_DC.l(t));
totalDH_cprod = sum(t, c_prod_DH.l(t));

total_hoursHP = sum(t,i.l(t));

storDCm3 = stor_DC_max.l/(rho*cp*convert*(dT_DC));
storDHm3 = stor_DH_max.l/(rho*cp*convert*(dT_DH));

Display totalDC_demand;
```

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