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New market models for export of excess heat from buildings

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MASTER THESIS

for

Laura Perez Casado

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New market models for export of excess heat from buildings

Nye markeds modeller for å eksportere spillvarme fra bygninger

Background and objective

Office buildings, commercial buildings, IT centers, supermarkets, and hospitals may have excess heat from cooling equipment or some other processes. This heat may be used within the building or exported to the district heating network. Unfortunately, there are no clear requirements and rules to support export of the excess heat or cooling from the buildings. There are also some other practical issues related to the excess heat such as: reliability of heat delivery, heat measurements, and guaranty of desired parameters. However, few examples in Norway and Europa show that the excess heat will become an important player in the energy market. Therefore, it is important to identify technical and economical requirements that will enable a higher use of excess heat in the energy market.

In a previous master thesis a model for estimation of the excess heat from hospitals is developed. This model can be extend to test and give a critical view what market models may be suitable to enable cost-economical export of excess heat. Introducing simplified models for heat pump performance and waste heat estimation may be beneficial. Student can develop her work in Excel.

The objective is to analyse district heating models that can be suitable for export of the excess heat. Based on the heat estimation models and different energy price models, the student should show advantages and disadvantages of different district heating models for heat export. Due to practical issues such as reliability of heat delivery and temperature levels, unceratinity analysis should be perfomed.

The following tasks are to be considered:

1. Literature review on the following topic would be necessary: price mechanisms and models in district heating, price models for electricity, district heating and cooling substation configuration that enable import and export of heat, heat pump technologies to enable high temperature heat.
2. Develop heat pump models for waste heat generation. Analysis on the working fluid and heat pump construction that enable high enough condensation temperature may be beneficial.
3. Develop models for heat and cooling import and export based on the literature review.

4. Perform uncertainty analysis where most of the relevant parameters will be treated such as: temperature levels, heat delivered, cost level, etc.
5. Analysis on heat pump size and waste heat amount may be beneficial.
6. Analyse the results and define which price models may have advantages for heat and cooling export.
7. Prepare material for a draft article.

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

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

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

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

Risk assessment of the candidate's work shall be carried out according to the department's procedures. The risk assessment must be documented and included as part of the final report. Events related to the candidate's work adversely affecting the health, safety or security, must be documented and included as part of the final report. If the documentation on risk assessment represents a large number of pages, the full version is to be submitted electronically to the supervisor and an excerpt is included in the report.

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

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

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

Department of Energy and Process Engineering, 20. January 2016

Natasa Nord

Olav Bolland
Department Head

Natasa Nord
Academic Supervisor

Abstract

As time goes by, the shift from paper to digital information management is continuously increasing in our society. In 2011, it was reported that data centers consume 1.1-1.5% of worldwide electricity [1], and that around 2% of global CO_2 emissions can be accounted by the IT sector [2]. Due to this fact, data centers have become ubiquitous, as we can find them in nearly every sector such as business, communications, academic or even governmental systems.

Consequently, there has been a continuous increase in the power demand in data centers that has created an observable impact on the power grid. A study group formed by the American Society of Heating Refrigeration and Air-Conditioning (ASHRAE) states that rack powers will increase to nearly 70 kW within the next decade, this will be met due to the introduction of ultra-dense computing architectures [3]. Added to this increase, there is also an increase in energy costs, which together, have made the energy efficiency in data centers a topic of concern.

According to a number of recent studies made by the Uptime Institute [4], the cost of power to the server can exceed the cost of the server itself within approximately four years. If we take into account then, the cooling and infrastructure costs, the operational cost over its lifetime is five times the cost of the server. This theory suggests that improving the cooling efficiency is a step in the right direction.

Rising up the cooling efficiency is also important in thermal management aspects. Poor thermal management can have some implications such as increased downtime, poor reliability, premature failure of servers due to inefficient airflow distribution [5]… which consequently would end up in increasing operating costs.

On the other hand, generally talking, most data centers, by design, consume vast amounts of this energy in an incongruously wasteful manner. Online companies typically run their facilities at maximum capacity all the day long, whatever the demand. As a result, data centers can waste 90% or more of the electricity they pull off the grid. This waste can be reused i.e., into for example, the district heating system.

Recovering this waste energy from the cooling part means that there is a considerable improvement in the energy efficiency of the data center. Also, as we extract this excess heat, there is a reduction in the net power that we would need to cool down this heat. Consequently, there will be also a reduction in the operating costs and finally, it becomes much more environmentally friendly.

Acknowledgments

The completion of this thesis could not have been possible without the support of so many people whose names may not all be enumerated.

I wish to express my sincere gratitude to Jørn Stene, for answering thousands of doubts and rendering his help during all the period of my thesis work.

To all my family and close friends for such support and patience.

Oscar, Ambra and Carlos, for uncountable library hours that I would repeat over and over.

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Finally, thanks to my supervisor, Natasa Nord.

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Abbreviations

DHW: domestic hot water

ODP: ozone depletion potential

GWP: global warming potential

DHS: district heating system

PUE: power usage effectiveness

CRAC: computer room air conditioning

DCie: data center infrastructure efficiency

UPS: uninterruptible data center power

PDU: power distribution unit

COP: coefficient of performance

CFC: chlorofluorocarbons

HCFC: hydrochlorofluorocarbons

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1. Introduction

Although technology is exposed to a constant and fast development, there are some areas of study that still lack of some sort of incentive so as to introduce them in our daily life.

In this project, the possibility of reusing the excess heat that can be found in data centers has been analyzed. Since its demand is constantly increasing, it seems a good starting point to a possible new way of increasing energy efficiency and making the data center being more environmentally friendly.

The work has been based on a theoretical large data center placed in Trondheim, Norway. The data and the key points that have been assumed are from three companies that collaborated in this project: *Nivos Oy* from Finland, *Romonet* and *Servetech* both from London.

Some challenges were exposed regarding the quality of the waste recovery heat and in the different alternatives of heat pumps/chillers so as to reuse it. Three different alternatives are studied in this thesis, with its thermodynamic study and life cycle cost analysis so as to be able to choose the best option for our concrete system. A previous analysis on the different possible working fluids has been carried out, as well.

The option I proposed to reuse this heat is in the district heating system, which by means of different heat pumps and liquid chillers, will increase the temperature of the low temperature waste heat from the data centers. This way, it is affordable to be used in the district heating system. As a result of this procedure, individual buildings will be served what means that they will not need their own boilers or furnaces to cover its base load demand. As a summary, I replaced its local heating system with the district heating system based on waste heat from data centers.

In the overall, it seems a pretty interesting projects which deals with different key points in the improvement of the energy efficiency field nowadays. The use of heat pumps/chillers in this system will allow to decrease the cooling demand for the data center whereas, at the same time, I will cover some of the heating demand of the buildings via district heating system thanks to the excess heat in the same data center.

1.1. Purpose of this report

This report has been organized in a way so as the main purposes are as follow:

- Overview of the trends of growth of data centers and the analysis of its efficiency as possible ways to increase it.
- Analysis of the viability of using excess heat from large data centers and discussion about its quality so as to use it into the district heating system.
- Comparison study between different heat pumps used to satisfy our main aim, which is the use of the excess heat, by means of a life cycle cost analysis.
- Analysis of the potential savings to both, the data centers and the buildings that are supplied with the heat.
- Explanation of some possible future perspectives.

2. Literature study

Some background was needed so as to define each of the concepts that were used in this project. In this chapter, each of the parts that are going to be used in the system from a theoretical point of view are explained.

2.1. Efficient data centers

A data center is defined as a place that contains primarily electronic equipment for data processing, storage and communications of digital information. They are designed for computers and not for people, that’s why data centers typically have no windows and minimal circulation of fresh air. This fact justifies the fact that, i.e. the cooling demand in a data centers is more or less independent from the ambient temperature.

The trend is that data centers are growing larger in size, which has led to an increase in the electricity consumption and the dissipation of heat. According to a survey carried out by the Association for Computer Operation Managers (AFCOM) and InterUnity Group, the power requirements of data centers are increasing by 8% per year on average, and 20% per year in the largest centers [6]. Consequently, due to this increase in the power requirements, servers have become more power dense and more energy is needed to operate and cool down the equipment.

It is important to notice that the vast majority of servers usually operate at or even below the 20% of their maximum capacity most of the time. This fact is also met even when the system is idle, where 60 to 100% of the maximum power is still drawn from the grid [7]. Almost in its totality, all this electric power that is supplied to the servers is then dissipated into heat. This heat leads to a need of large scale cooling systems so the permissible temperature levels within the racks does not exceed, and to maintain these temperatures levels in a safe operational range.

Figure 1. Example of servers in a data center [8]

Many analyses have been done regarding how the power is distributed in a data center depending on its needs. An analysis of a typical 5000-square-foot data center carried out by the EMERSON Network Power shows that the cooling part is the most dominating need in a data center since it stands for approximately 38% of the total power demand.

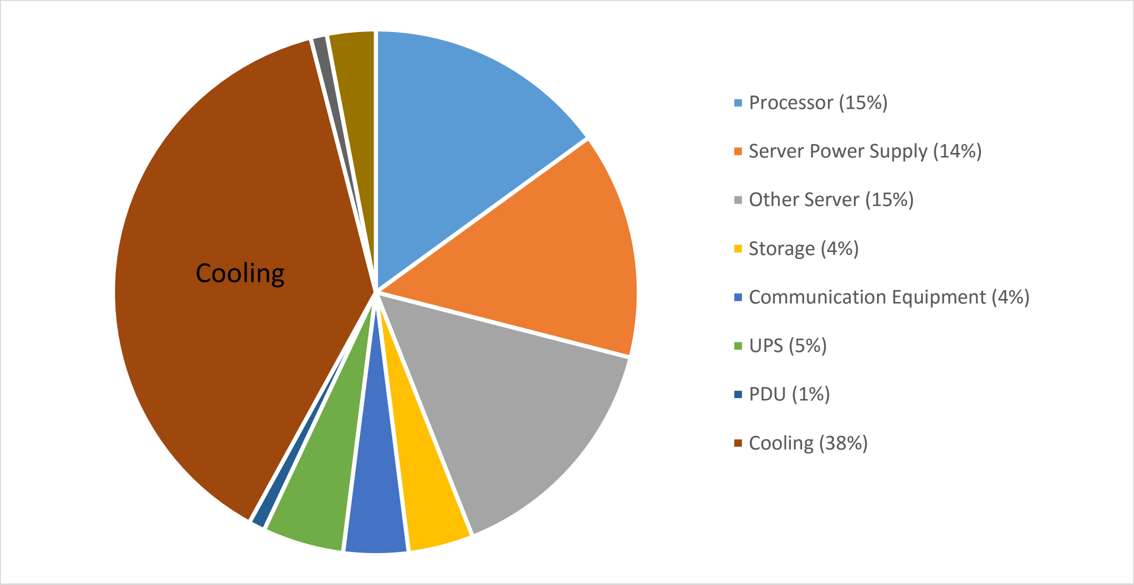

Figure 2. Analysis of the power usage in a typical data center of 5000 square-foot [9]

In Figure 2 it is important to highlight the fact that we can sum up all the categories into two main groups which would be:

Support system (48%) - Supply	***Computing Equipment (52%) - Demand***
Cooling	Processor
Building Switchgear/MV Transformer	Server Power Supply
UPS	Other Server
PDU	Storage
Lighting	Communication Equipment

Table 1. Differentiation between supply and demand in a data center

The reason why there is so much power destined to cooling is due to the standard for cooling data center are the CRACs (Computer Room Air Conditioners). These units are electrically-driven and, as explained above, since cooling is such an important portion of the data center grid power consumption, so as to increase the efficiency, the main aim is to reduce the grid power consumption of the cooling system for data centers.

Furthermore, as stated before, one of the most important benefits that re-using data center excess heat offers is the improvement of the efficiency in the data center. The most important metric to measure efficiency is called PUE (Power Usage Effectiveness). The PUE is the most recognized data center efficiency standard. It was introduced by The Green Grid in 2006, and since then it has been understood as the total amount of power consumed by IT equipment (W_{out}) relative to the total facility power ($W_{in)}$ [10].

$$PUE = \frac{Electrical\ utility\ power}{IT\ equipment\ electric\ power} = \frac{\dot{W_{TOT}}}{\dot{W}_{IN}}$$

Equation 1. PUE

According to a survey of more than 500 data centers conducted by The Uptime Institute, the average PUE rating for data centers is 1,8 [11]. That means for every 1 W of electric input to the IT equipment, an additional 0,8 W would be consumed to cool and distribute power to the IT equipment. A PUE of 1 requires the maximum theoretical efficiency. However, this is not possible as it will always take some form of energy consumption to support the IT equipment. It should be recorded over a representative period so that a realistic annual average can be obtained.

As a conclusion, the PUE intends to highlight the amount of electric power that is used to run the equipment compared to the needs of the auxiliary system in a data center, such as the cooling and the Power Distribution Units. Unfortunately, whilst there is theoretical literature available regarding methods of calculating the PUE, none look into the sensitivity of PUE to certain parameters or attempt to repeat a PUE calculation using open source information [12]. This makes it hard to transfer ideas in order to reduce the energy consumption within data centers.

Consequently, there is a requirement of establishing a metric of data center efficiency, which should be used to rank data centers [13]. Some of these efficiency targets would be:

- Ranking based on the DCie metric, which stands for the data center infrastructure effectiveness.
- Age of the facility, since the major difference in efficiency between facilities is related to age.
- The geographic weighting, as the humidity and the external ambient temperature can affect the efficiency of a data center.
- Humidity ranges and humidity control.

The ASHRAE recommends a humidity range of 5,5ºC dew point to 60% relative humidity and 15ºC dew point [14], which is not a problem in Norway since it is not a country with high levels of humid air.

2.2. Waste heat recovery

One of the main challenges to the implementation of the waste heat recovery system, although there are huge amounts of heat that could be utilized, it is the low temperature level. Also, the capture temperature is limited by the temperature limits of the electronic systems, which normally remains below 85ºC [7].

The available temperature on the waste heat is depending on the type of chiller/heat pump system. It makes it challenging to deal with this low temperature heat, although working with heat pumps seem to be a good idea to overcome this problem.
In order to capture this excess heat the principal cooling methods for data centers have been analyzed:

- Air cooling

The heat generated by the processors is redirected to a heat sink and then transferred to the chilled air blowing into the server. Normally, the cooling air enters through a second-floor plenum to the cold aisle in front of the racks and exits through the opposite side. The hot air goes to the upper part and is moved to the CRAC units where it is cooled by chilled water. It is important that this chilled water is at a sub-ambient temperature so as to produce enough heat transfer rate. Although, this system is not used to recover heat.

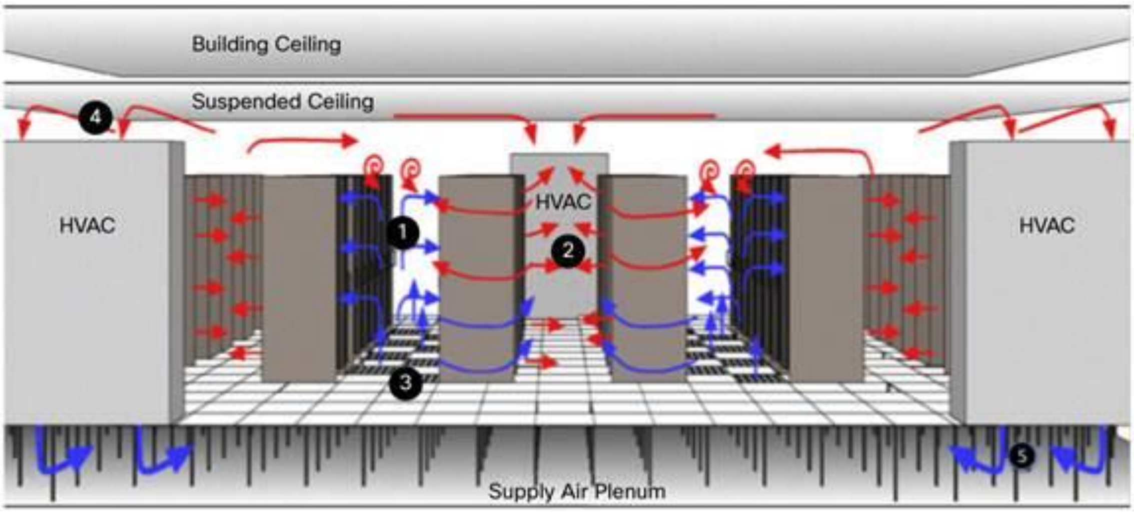

Figure 3. Air cooling functioning in a data center [15]

- Liquid cooling

Comparing to the air cooling, the liquid cooling has a higher heat transport capacity and therefore it can remove higher levels of heat during the heat exchange [16]. However, it might increase the overall costs and the complexity, but liquid cooling seems to be appropriate for high power density components such as CPUs.

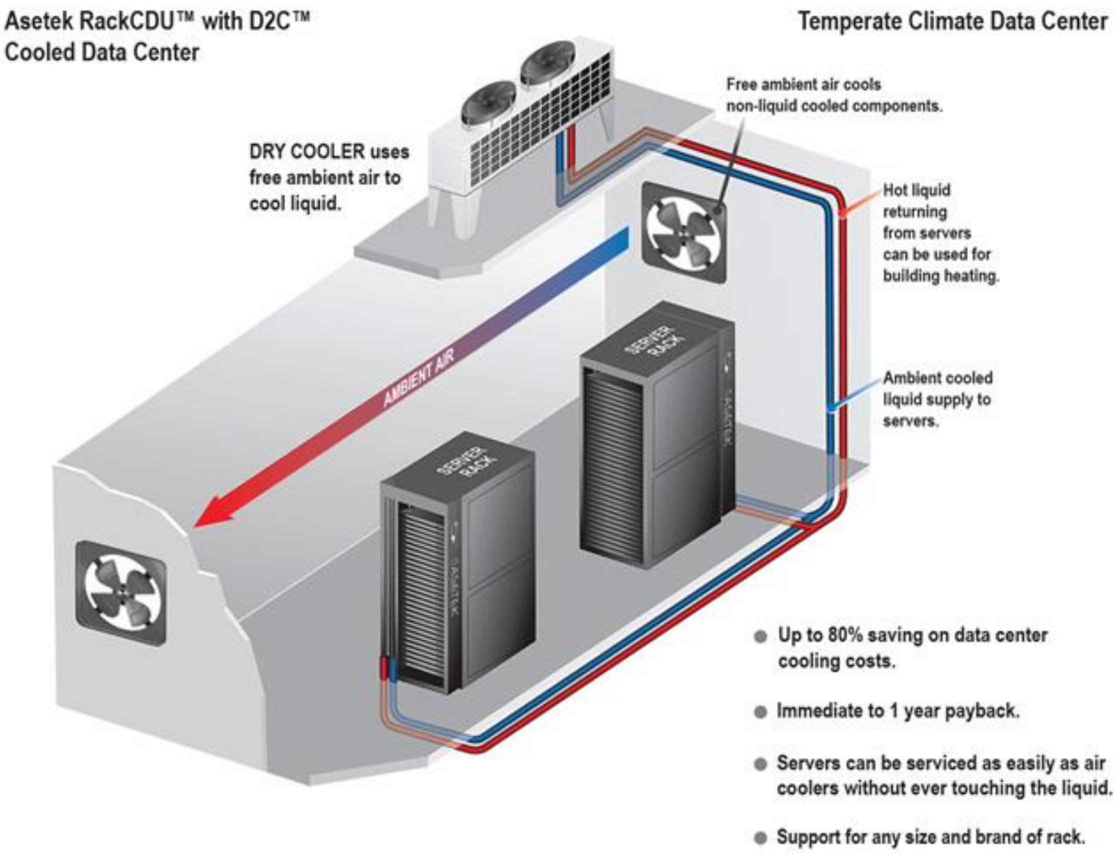

Figure 4. Liquid cooling in data centers [17]

- Liquid immersion cooling

Liquid immersion cooling is not as complex and expensive as liquid cooling. The servers are immersed in non-chemical and with very low boiling point liquids. These servers would create heat and consequently will create vapor. Vapor will rise to top, where it arrives to the condenser placed in the upper part of the structure and condenses.

Figure 5. Immersion liquid cooling in data centers [18]

- Tower free cooling

Normally used when it is not practical to create large floor openings in facilities. Due to the risk of legionella this cooling method is not recommended.

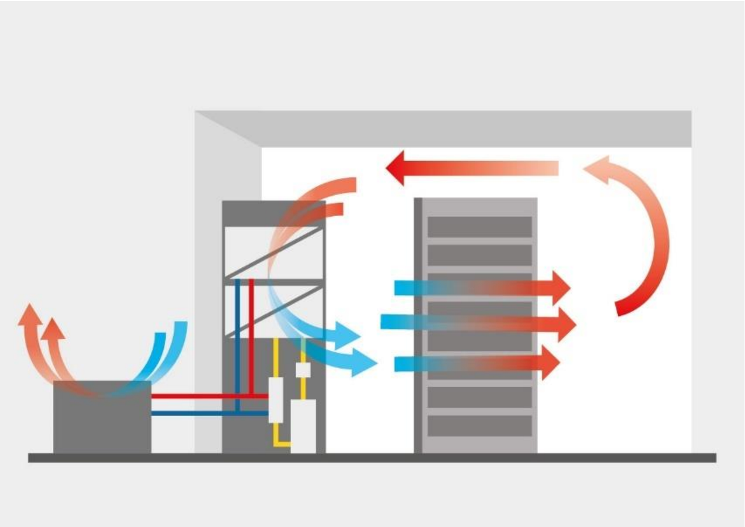

Figure 6. Free cooling in data centers [19]

To sum up, in the state-of-art cooling methods, the energy efficiency is only moderate, which means it may be increased during next years. IBM states that liquid cooling is very efficient for high power density subsystems due to its high heat transfer coefficients, but Intel doubts its efficiency for the whole system [20].

	Air cooling	**Liquid cooling**	**Tower free cooling**
Energy efficiency	Medium	High for high power density subsystem but medium for whole data centers	High
Retrofit cost	Medium	High	Medium
Weather dependence	Low1	Low	High

Table 2. Overall comparison between different cooling methods [21]

Table 2 that shows a summary of the characteristics for each of the cooling methods, where heat recovery is possible, it can be concluded that the best cooling method in our case is the liquid cooling method, because the annual heat recovery can be really high and may reach a high efficiency, if the temperature level in the district heating system is low enough and the heating capacity of the system is much lower than the heating load in the district heating system. We avoided using tower free cooling or methods of cooling of the kind due to risk of legionella. [22]

So as to make our system run, the only energy source we used was the waste heat dissipated from the IT equipment in the data center. By using this energy source we can find two different advantages:

- Use of excess heat that otherwise would have been rejected to the ambient, what it means it would be lost energy.
- Reduction of the power needed to make the CRACs run, since the cooling demand would be inferior, as we are using part of the heat that would have been needed to be cooled down.

What became a point of discussion was to figure out how could we transport it in an efficient manner so as not to create significant heat losses.

Most data centers operate in a temperature range between 68 and 72 ºC [23]. So in case the racks would exceed significantly this range, failure would occur. In this case, the whole system should be switched off. This radical decision has to be made quickly because in case the cooling system fails, the temperatures would increase very fast and it would keep rising if the equipment would be still switched on. Besides, all equipment in racks is different between each other, but their technical specifications would explain what their suitable operating temperatures are.

Regarding the use of the excess heat from data centers, it is logic to think about the geographical area of the buildings we want to supply heat. For the building to meet the requirements it must be within a relatively small geographical area, which means that relocating the data center to another country so as to take advantage of a higher efficiency or even a lower carbon utility electrical supply is quite practical.

2.3. Heat pump/chiller technology

In the future, sustainable energy systems will have two basic principles which are efficient end-use and efficient use of renewables, as well. In this aspect, heat pumps are a good technology to use since the energy source used by a heat pump is renewable energy from ground, air, water and waste heat sources.

As an example, according to the International Energy Agency, IEA, the buildings sector needs to reduce its CO_2 emissions by over 70% in comparison with 2010 levels to limit the possible increase in global temperature. The main challenge here appears when we have to reduce the emissions while at the same time the energy demand is rising. Heat pumps can contribute to meet this goal, due to its efficient end-use and renewably supply.

In our concrete case of study, heat pumps are used as chillers so as to remove the excess heat from a data center in order to use it into the district heating system. As it can be understood, at the same time this excess heat is removed, we are achieving a decrease in the cooling demand from the data center, which ends up in a saving of energy. So this cooling demand is reduced thanks to the rejection of the heat via heat pump/chiller, as they are using as heat source its heat, this means this heat will not need to be cooled down.

In the following chapters, a life cycle cost analysis was carried out to prove which kind of heat pump technology is the best to use in this case. Possible working fluids have also been discussed.

COP is a value that is calculated based on the input depending upon the output desired, not as the efficiency, where the output depends upon the input that is given.

There is a theoretical maximum value for the COP, which is 1. This value is met when the input energy is exactly the same as the output energy. It means the energy is completely transferred, without any losses. In the practice this fact cannot be possible, due to the need of some energy to make the system run, and of course losses are met during the process.

Referring to the thermodynamic topic, the higher value of COP is met when:

- The condensation temperature is much lower than the critical temperature.
- We have a high pressure level with high compressor efficiencies and low relative pressure loss.
- Low vapor density and low liquid density (low pressure loss in components, valves and tubing).
- Large U-value and moderate heat exchanger surfaces.

A heat pump utilizes at least one compressor which transports the refrigerant around in a closed pipe circuit. A heat exchanger at the first stage is the responsible of getting the temperature from different heat sources, that are commented in the following paragraph, so as to transfer it to the refrigerant. Once this refrigerant is warmed up, it goes through another heat exchanger where it will release the high temperature and then the cycle would start again.

Different heat sources can be found nowadays.

Heat source	**Availability**	**Typical limitations**
Ambient air	Everywhere	Temperature, noise from fans
Bedrock	Almost everywhere	Rock condensation, available area, uncompacted material, temperature.
Soil	Lawn, ground, moor	Soil type, available area, temperature
Seawater	Along the coast	Temperature, distance to the sea. Depth profile, pollution
Ground water	Gravel, sand, rock	Flow rate, temperature, water quality, setting
Lake water	Lakes	Temperature, distance to lake, depth profile, pollution
River water	Rivers	Flow rate, temperature, distance to the river, depth profile
Ventilation air	In buildings	Air flow rate, temperature, pollution
Sewage	Main pipelines	Flow rate, temperature, treated/untreated source
Grey water	In buildings	Variations in flow rate, temperature, pollution

Cooling water	From industry	Variations in flow rate, temperature, pollution

Figure 7. Different heat sources for heat pumps [22]

The current trend in heat pumps is to design them closer to full coverage of the heating demand, compared with previous designs [24]. Due to both economic and legislative reasons, this development has been met. Consequently, heat pumps will operate at part-load almost all the time and so the power utilization will be low. It is important then to use adapted electric motors, drives and to make control of getting a good COP also in part load.

In this project variations with part load were done, theoretically talking, the COP may increase with decreasing part load [22], but we could see in the following chapters that it may not be accomplished in all the cases.

The maximum outlet water temperature from the condenser that each working fluid could achieve:

HFO – 1234zy		**R717 - ammonia**		**R744 – CO_2**
2-stage / 25 bar	2-stage / 40 bar	1-stage / 50 bar	2-stage / 60 bar	1 stage / 150 bar
70ºC	90ºC	80ºC	90ºC	90 ºC

Table 3. Temperatures depending on pressure difference

This project was mainly focused in heat pumps designed for high-temperature heat recovery. Different types of heat pumps that accomplish this temperature level could be possible:

- Single-stage (advanced) systems
- Two-stage systems
- Cascade systems
- Hybrid systems
- Combined systems

Three alternatives will be studied in this thesis, first an analysis of the single-stage system based on the existing example in Gløshaugen will be explained. Followed by a two-stage systems in order to see the differences both systems can show. Finally, a combined system using a hybrid cascade system will be studied, based on the analysis made by Hanna Risnes [25].

2.4. District heating systems

The main function of a district heating system is to supply costumers with thermal energy for space heating and production of domestic hot water. DHW is also able to cover low temperature industrial heat demand, which means a supply temperature of 50ºC at the costumer. It has many advantages such as reliability, low investment, it is environmentally friendly, its energy efficiency… [25].

A district heating system comprises different parts such as heating plants, primary heat distribution system for hot water and heat exchanger substations in the different buildings [22].

In a district heating system, we can find two different types of energy input depending on the different level of energy supply we are talking about:

- Primary Energy Supply: direct use of renewables or fossil fuels, with no previous users for this energy input.
- Secondary Energy Supply: heat recycled from combined heat and power, waste incineration and industrial processes.

In Figure 8 the fundamental idea of district heating based on the different types of energy inputs is displayed:

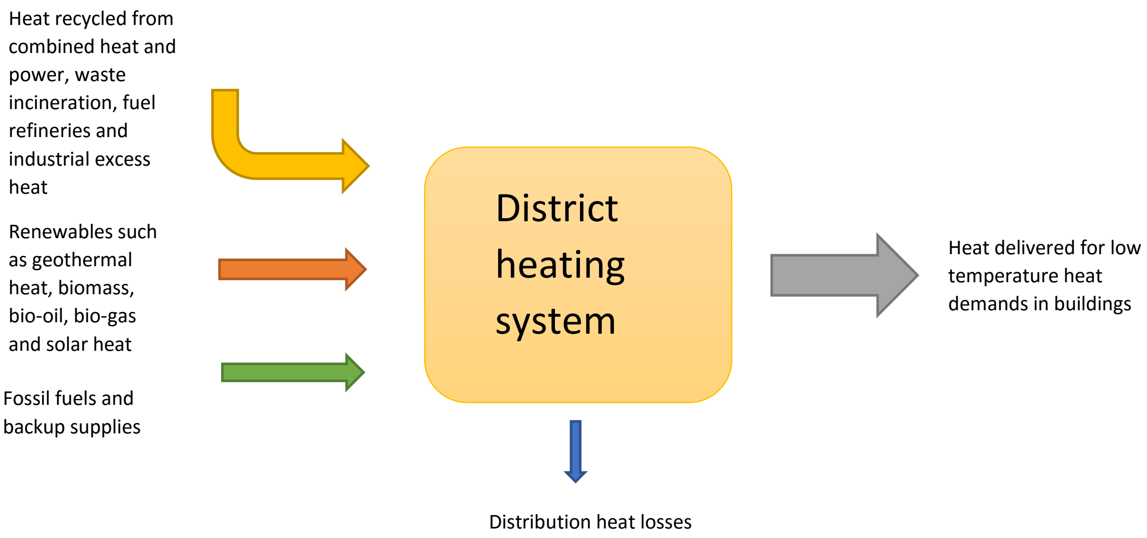

Figure 8. Principal idea of district heating systems

A district heating system comprise a heat source, a closed loop transmission network of pipes and local substations in which the district heating water is transferred to the heat consumer circuit and cooled down [25].

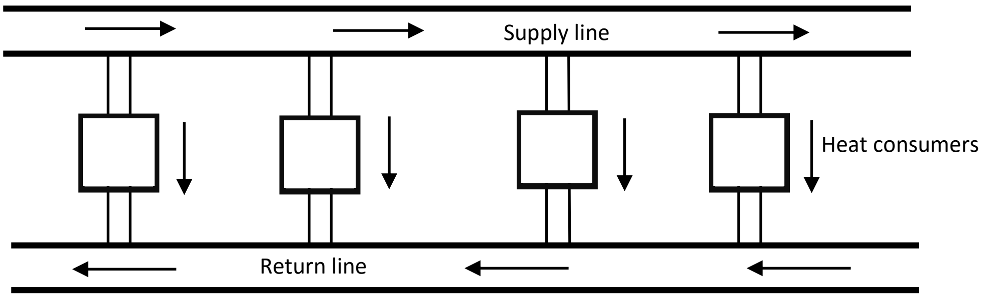

Figure 9. Sketch of the primary side of a district heating network

As we can see in Figure 9, distribution pipeline system would be called the primary side of a district heating network, while the secondary part would be the consumer circuit such as space heating and DHW circuits. On the primary side, every substation is connected both to the supply and the return pipelines.

There are different supply temperature levels for each different district heating system. In case of low-temperature supply, the main advantage is that the heat losses from the pipes to the ground will be smaller. On the other hand, if the supply temperature is high, it means that the flow will be smaller and then consequently that the dimensions of the pipes will be smaller.

Country	**Supply Temperature (°C)**	**Return Temperature (°C)**	**Hot water (°C) (DHW)**
Denmark	70	40	<60
Finland	70	40	55
Korea	70	50	55
Romania	95	75	
Russia	95	75	50
United Kingdom	82	70	65
Poland	85	71	55
Germany	80	60	55
Norway	120	70	

Table 4. Examples of temperatures used for district heating design temperatures at DOT

In Table 4 we can see some examples of both supply and return temperature for the different countries and the temperature for the hot water in each of them, as well.

2.4.1. Heat sources

There are different energy sources such as industrial waste heat, geothermal, solar systems and heat pumps. Also, there are the conventional boilers and cogeneration systems.

In order to increase the low-grade energy sources efficiencies, it is important to have a low district heating return temperature. This is because low temperature return water is able to absorb more thermal energy from these sources. For this reason, the temperature level for consumer's installations should be as low as possible so as to increase the efficiency of the system.

2.4.2. Distribution system

The district heating water is distributed from the heat source through supply pipes to the customer's heat exchanger stations and it is returned after heat has been extracted. It is important to remark that as there is a limited high pressure in order to avoid the possible damage in the pipelines, there is also a limited low pressure so as to avoid cavitation.

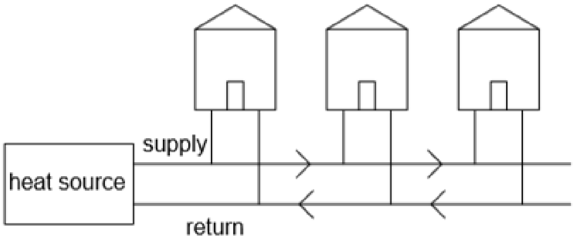

Figure 10. Sketch of the distribution system in district heating systems

2.4.3. Pricing mechanism

District heating has been considered as an efficient, environmentally friendly and cost-effective method for heating in buildings, and is playing an important role in the mitigation of climate change.

There are two basic pricing principles which are:

- Cost-based pricing that gives a price level for all annual costs together with a certain return on investment capital.
- Market-based pricing which is either direct or indirect It involves a concrete formula or the price is set once a year in accordance with a market alternative.

Besides, the district heating systems must be competitive to the other ways of generating heat. This competitiveness depends on three different factors:

a) The cost of heat distribution
b) The cost for heating up a dwelling

- *Price components of district heating*

The district heating price is composed by three main factors:

a) Connection fee: it is referred to the price in order to be able to connect a dwelling to the district heating network.
b) Standing cost: fixed costs such as meter reading, maintenance…
c) Unit cost: is the price per unit of supplied heat

Here in Norway, there is a regulated market for district heating pricing, which means that the price is regulated by the government and this price will dictate the profit that district heating companies can make. Normally, the kWh-price is 10% lower than the electricity price.

The main advantages of the regulated market are [25]:

- It is simple, flexible and ease of administration
- District heating companies have incentives to inflate costs
- Undermining suppliers incentives to reduce costs and upgrade technologies

2.5. Excess heat for district heating

Apart from the above commented different heat sources we can take advantage of, we can also use excess heat as a heat source the excess heat. In order for excess heat to be used for district heating it has to have a high sufficiently temperature. Besides that, taking into the account that for each country there is a different supply temperature, we can also play with the fact that depending on the final temperature we can get, we can deliver the heat to either the supply or the return pipeline depending on the available temperature level. The main disadvantage of delivering heat in the return pipeline is that it may be more difficult to get profit from it as we would get approximately 30% less benefit than delivering in the supply pipeline.

In this project, our aim is to work with this excess heat from a concrete load in order to deliver it into the district heating system and after that, make a study to see the availability, since as I was dealing with computer cooling, the load was about to be more or less constant.

3. Possible scenarios

This project deals with different scenarios in order to choose the best option to our system.

3.1. Heat pumps

Nowadays heat pumps are a good option since although they have high investment costs, its operational costs are low, which ultimately we can take advantage of. So as to analyze how profitable a heat pump system is, we need to carry out some calculations [22]:

- Specific investment
- Annual cooling / heating supply
- Plant's annual COP (SPF)
- Energy prices

Apart from that, it is also important to make a comparison with the different efficiencies of each system. Taking into account the efficiencies and then the life cycle cost analysis will make easier to do the right choice depending on your specific requirements.

In this project, three different types of heat pumps have been analyzed:

- One-stage ammonia chiller/heat pump
- Two-stage ammonia chiller/heat pump
- Hybrid two-stage chiller/heat pump

It was needed to carry out the calculations for each of the heat pumps in order to find the average COP value and the condenser heating capacity. The study has been complemented with a life cycle cost analysis (LCC).

3.1.1. Analysis of the COP

- One-stage ammonia heat pump/chiller

This system follows the next scheme:

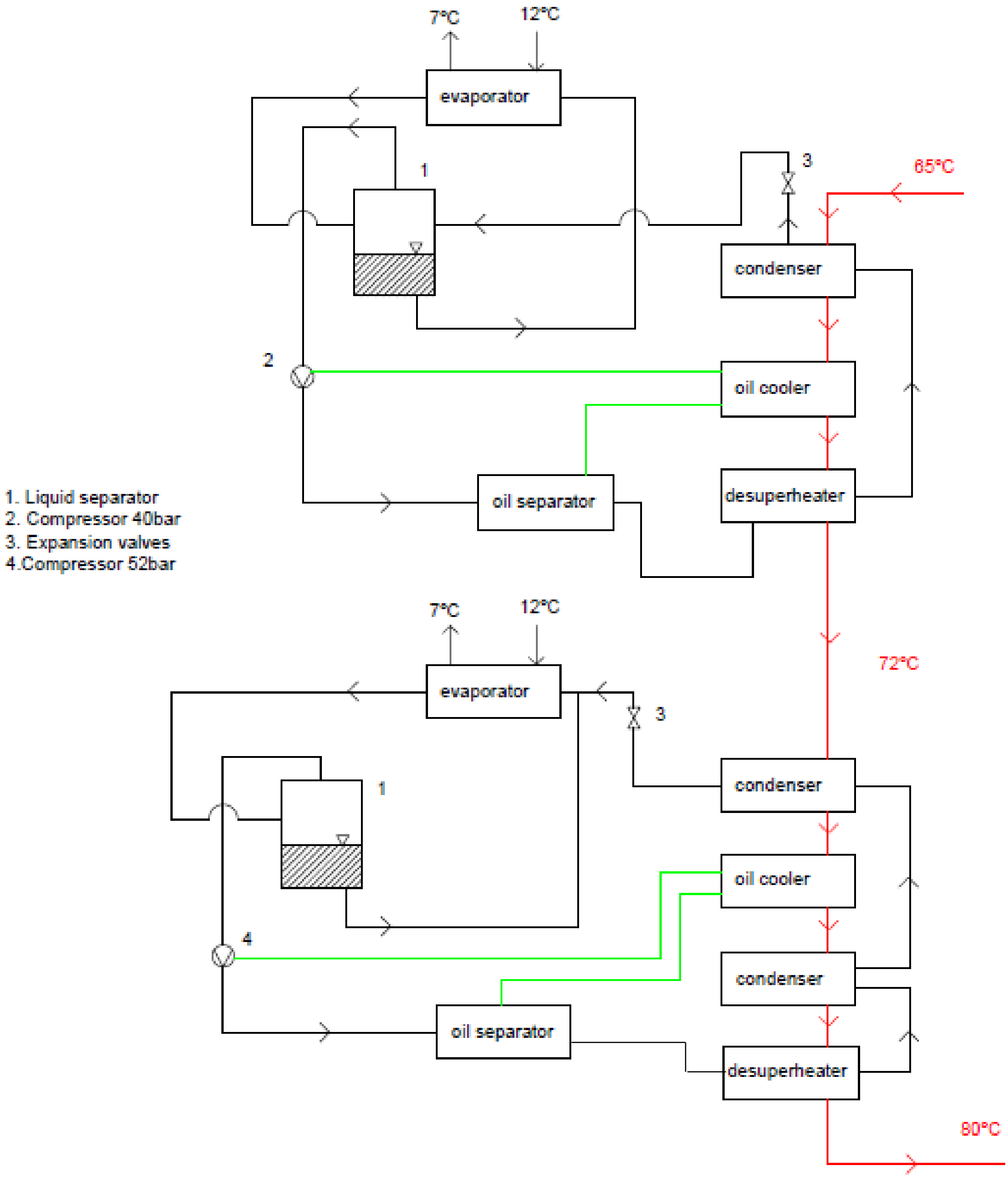

Figure 11. Scheme for the one-stage heat pump/chiller from NTNU [26]

Talking about the heat pump, which will be the part oriented to the district heating network, all the system starts when the excess heat leaves the data center at 65ºC so as to go through a condenser that acts like a heat exchanger with the heat that comes from the desuperheater. After that, the warm water passes through the oil cooler which will increase even more the temperature thanks to the heat that the oil cooler provides. This oil cooler, as its name indicates cools down the oil so as to redirect it again to the compressor, this rejected heat is the one used to increase the temperature of the water. Up to now, the water has passed through the Unit 1, which is marked in Figure 11. At this point, the water is at 72ºC. It is important to highlight the order of the components in this heating system. It is logic to think that the water has to go from the lower temperature up to the highest temperature, that is why they are placed this way. Entering through the Unit 2 of the heating system, we find a subcooler so as to increase the efficiency and avoiding expansion losses. This subcooler is fed by some heat coming from the condenser. The water then goes through a second oil cooler which makes it warm up in order to be ready to go through the condenser, fed by the desuperheater, as in Unit 1 and finally through the desuperheater so as to reach a temperature of 80ºC and be redirected to the return pipeline of the district heating system.

Some interesting points in the heating system are the use of desuperheaters, oil coolers and subcoolers. So as to be clear in these points, we defined them by means of a graph with Coolpack.

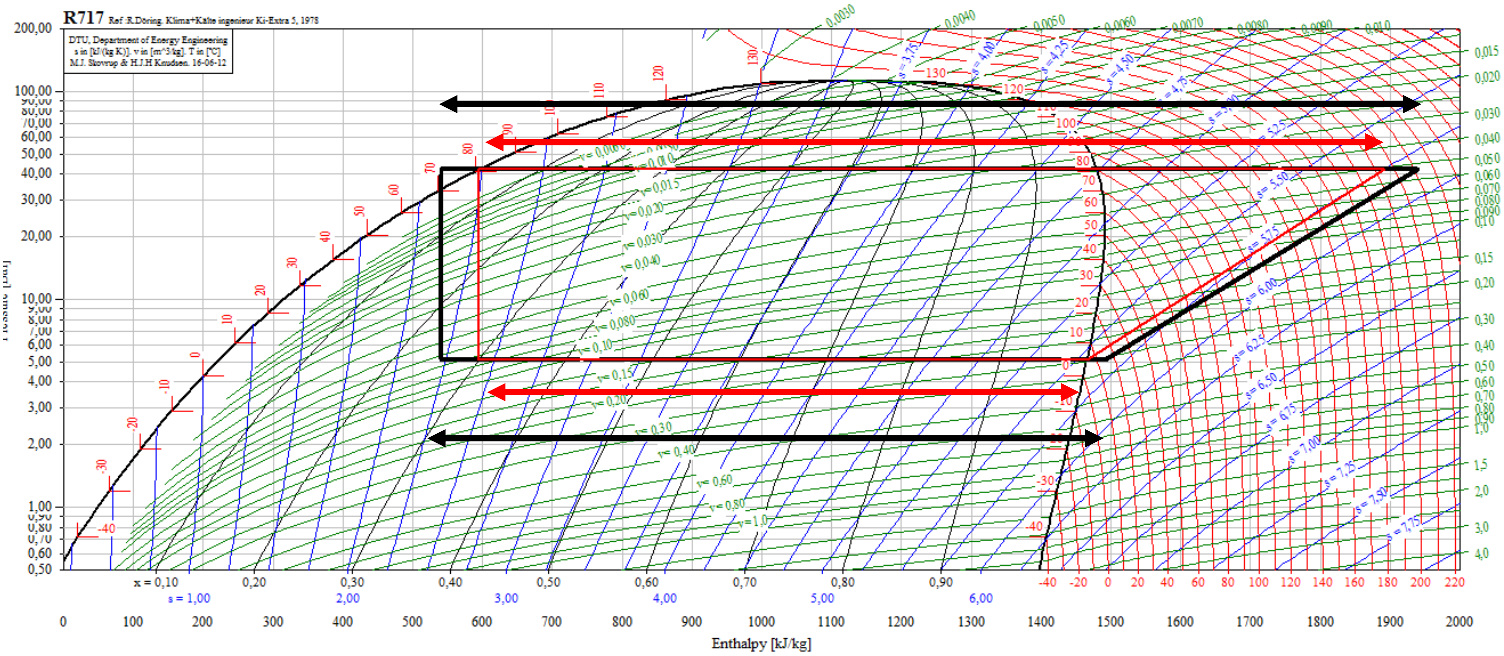

Figure 12. Thermodynamic cycle using Coolpack

As we can see in Figure 12, there are two different cycles shown. As an example case we assumed:

- Isentropic efficiency of 75%
- A typical value of LTMD of 4,33 for the evaporator and a LMTD of 5 for the condenser. Using then the formula:

$$LMTD = \frac{\Delta t_{a-}\Delta t_b}{\ln\left(\frac{\Delta t_a}{\Delta t_b}\right)}$$

Equation 2. LMTD

For the red cycle, neither subcooler nor evaporator is used, which means as we can see in Figure 12 a smaller value of both cooling and heating capacity can be achieved, which of course leds to a smaller value of COP.

$$COP_h = \frac{Heating\ capacity}{Work\ done\ by\ the\ compressor}$$

Equation 3. COP for the heating

Also, as we can see in the Figure 12, there is no big difference between the work done by the compressor since we are using a desuperheater that will reheat the water after the oil cooler and the condenser, so it will reduce the required condensation temperature. So, in the black cycle we used a subcooler and a desuperheater. As commented above, both the heating and cooling capacities in this case are higher, this is due to, on one hand, the desuperheater that takes profit of the rejected heat in the compressor and on the other hand, the subcooler that decreases the temperature of the water so as to increase its efficiency. Both avoid expansion losses.

Regarding the cooling system, we have two evaporators that are theoretically connected in parallel as shown in Figure 13. Parallel connection of the evaporatorsFigure 13:

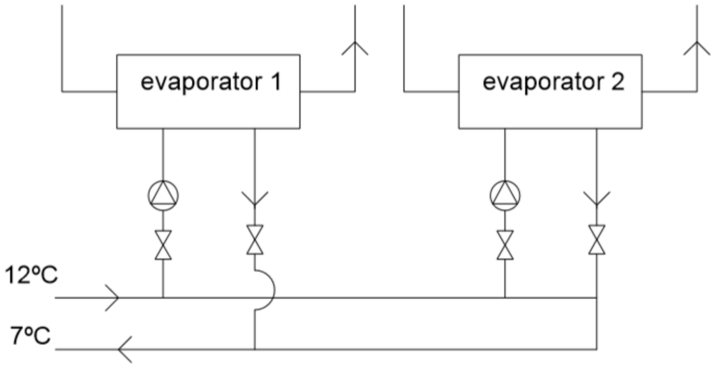

Figure 13. Parallel connection of the evaporators

After the evaporator, we can find a liquid separator that is the responsible of separating the liquid from the vapor, this way we avoid the presence of drops in the compressor, which could damage it. In this system, there are two compressors, the one in Unit 1 of 40 bar, and the one in Unit 2 of 52 bar. This fact makes sense as our aim is to warm up the water so, the higher the pressure in the compressor is, the higher the temperature of the working fluid, and so consequently, the higher the temperature in water.

Since we had the evaporator connected in parallel and the five other heat exchangers connected in series, this is the looking of the temperature glide in these heat exchangers.

Once we analyzed the general behavior of this heat pump, the COP values for different outlet water temperatures has been calculated, and also for different part loads.

Qcooling (kW)	Qheating (kW)	W (kW)	Part load heating(%)	Tsupply (ºC)	COPh	COPc	Relative enery saving(%)
800	1314	514	100	80	2.56	3.56	60.88
548	919,8	371.8	70	80	2.47	3.47	59.58
382	657	275	50	80	2.39	3.39	58.14
800	1102	302	100	60	3.65	4.65	72.60
548	771.4	223.4	70	60	3.45	4.45	71.04
388	551	163	50	60	3.38	4.38	70.42
800	965	165	100	40	5.85	6.85	82.90
553	675,5	122.5	70	40	5.51	6.51	81.87
393	482,5	89.5	50	40	5.39	6.39	81.45

Table 5. Results for the one-stage ammonia heat pump/chiller

In Table 5 some results are displayed. First of all, regarding the temperature supply variation, an increase in COP with decreasing supply temperature is met. Consequently, the lover the supply temperature is, larger is the relative energy saving value. Regarding the part load variation, Figure 14 shows the variation in COP at part load operation, and an explanation about its variation is proposed after it.

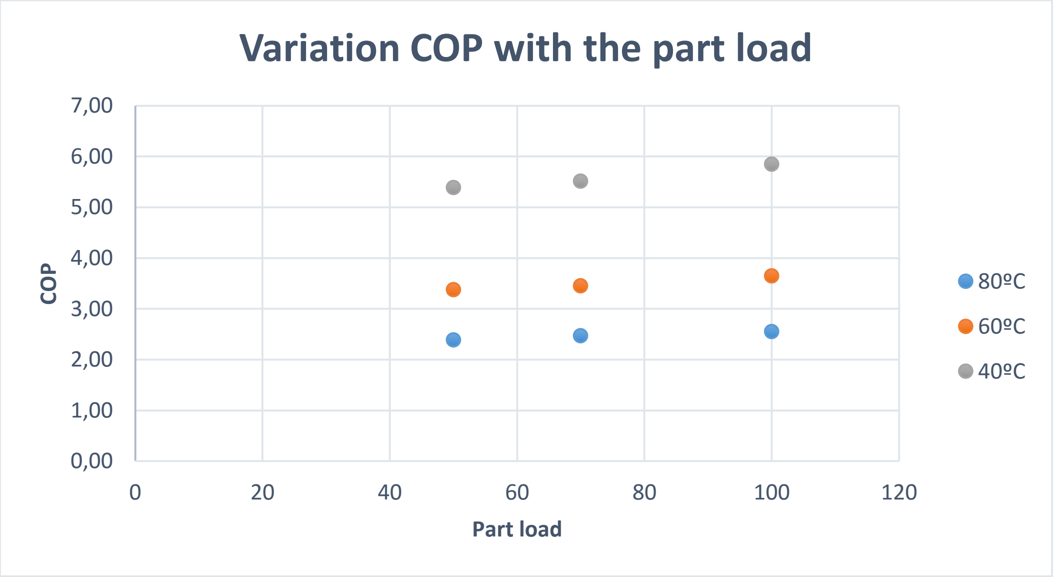

Figure 14. Variation of the COP vs part load, variation in 50-100% ranging from 40 to 80ºC

The COP drops slightly when the part increases, i.e. from 100 to 50%, but we have to take into the account that as there is an increase in the part load, the compressor efficiency may increase when using the variable speed drive motor since the pressure losses in the compressor will be lower due to a lower mass flow rate.

The part load has been reduced up to 50%, this is due to the fact that from 100% to 50%-part load the compressors are using the variable speed drive, VSD. This mode offers a relatively high and very stable COP. Below 50%-part load, approximately 50% of COP drops rapidly at decreasing capacity. In this range the compressors are using the slide valve control. That is why the analysis was not performed at below 50%-part load. [22]

What can be also be inferred from Figure 14, is that as the outlet water temperature is lower, the COP is higher. This fact can be explained by noticing that COP varies depending on the inlet and outlet temperature. Remembering that the equation for the COP is as follows:

$$COP_H = \frac{Q_{heating}}{W}$$

Equation 4. COP for the heating

The less work the compressor has to reach the required outlet temperature, the smaller will the W value be (as it is referred to the work done by the compressor), and consequently, the smaller the W value is, the higher the COP will be.

- Two-stage ammonia heat pump/chiller

The scheme of this system is as follows:

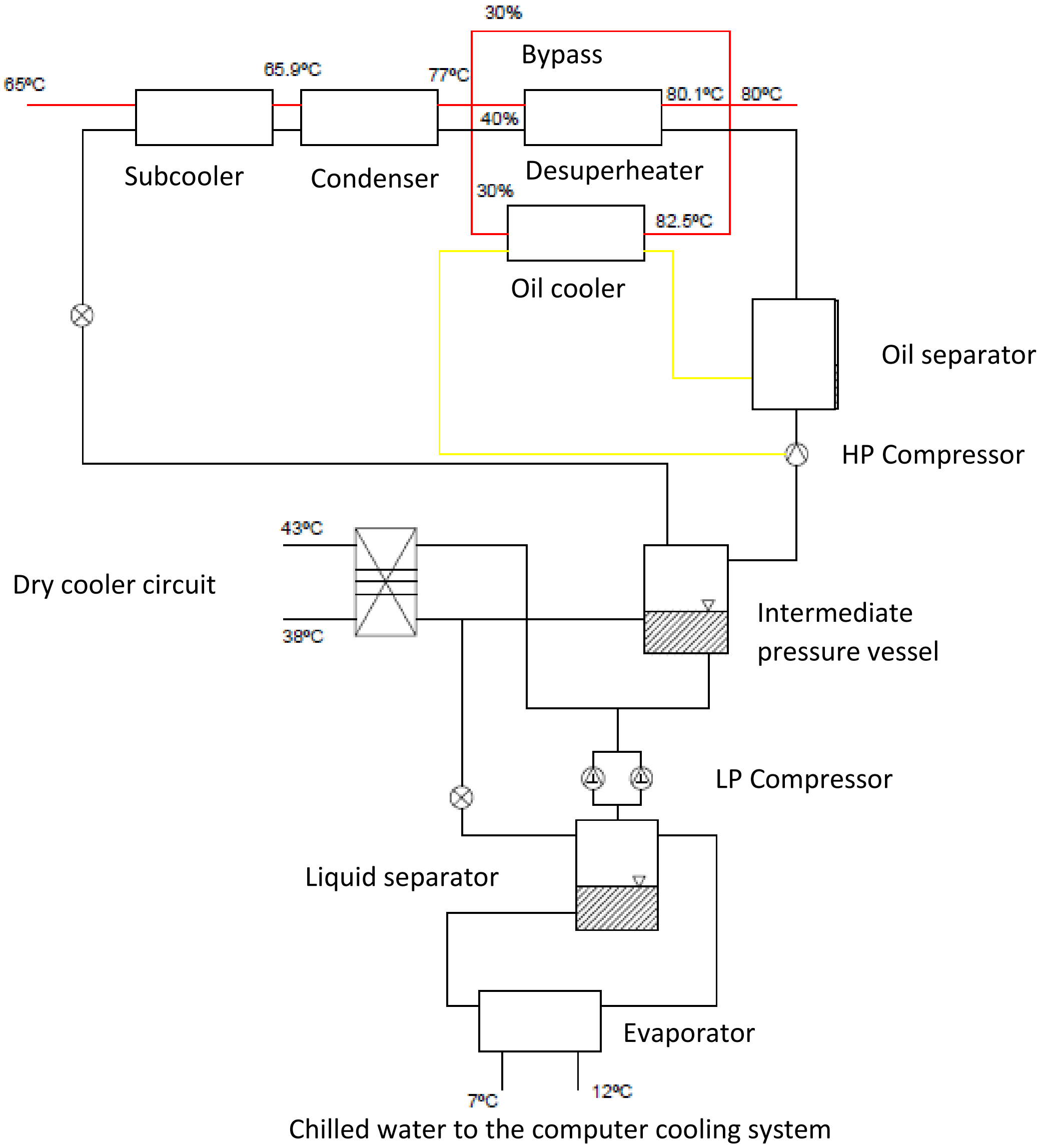

Figure 15. Scheme for the two-stage ammonia heat pump/chiller [27]

The working fluid used for this scheme was ammonia. The principal scheme for the two-stage ammonia heat pump/chiller is powered by two reciprocating compressors for the first stage. For the high pressure stage, a single screw compressor is used. So as to be able to meet the maximum possible heat recovery, the high pressure stage has been equipped with a subcooler, a condenser, an oil cooler and a desuperheater.

Excess heat is rejected from the low pressure stage via dry cooler.

The results Hannah Risnes obtained are as follows:

Qcooling (MW)	**Qheating (MW)**	**Part load (%)**	**Tsupply (ºC)**	**COP heating**	**COP cooling**	**Relative enery saving (%)**
860	1263	100	80	3,91	4,91	74,42
799	-	100	60	4,66	5,66	78,54
812	974	100	40	5,30	4,30	81,13
812	974	100	40	3,91	4,91	74,42
575	693	70	40	4,66	5,66	78,54
426	505	50	40	5,15	6,15	80,58
265	316	30	40	5,51	6,51	81,85

Table 6. Specifications for the two-stage ammonia heat pump/chiller

In Table 6, first of all a comparison of COP using different outlet temperature is achieved. As commented in the previous chapter, once again it is proved the fact that as lower is the outlet temperature, the higher is the COP due to the work that is needed to be done by the compressor to achieve the required outlet temperature.

As a second term, a comparison of the COP was made maintaining the same low outlet temperature of 40ºC and varying the part load. At this point, I have to comment that since this data has been taken from a previous project [27], and it follows the theoretical background I stated in the previous chapter. The higher the part load, the higher the COP. In the case of 100%, there is no part load and she got a heating COP of 3.91, while having 50%-part load, the heating COP was 5.15.

- Hybrid two-stage heat pump/chiller

Two technologies are combined in hybrid heat pumps: absorption and compression. To carry out these two processes, this kind of heat pumps use a mixture of media, for instance, ammonia and water. Due to changes in the composition of the mixture caused by absorption and desorption, heat is extracted and emitted at a non-constant temperature [28]. An advantage with hybrid heat pumps is an equal lift in temperature with a lower compression ratio, compared to conventional heat pumps. This leads to a higher COP. Also, the condensation temperature is higher compared to conventional compression machines, due to the lower saturation pressure for a mixture than for pure gaseous refrigerants.

A sketch of the functioning of the hybrid chiller/heat pump that was considered as an alternative is displayed.

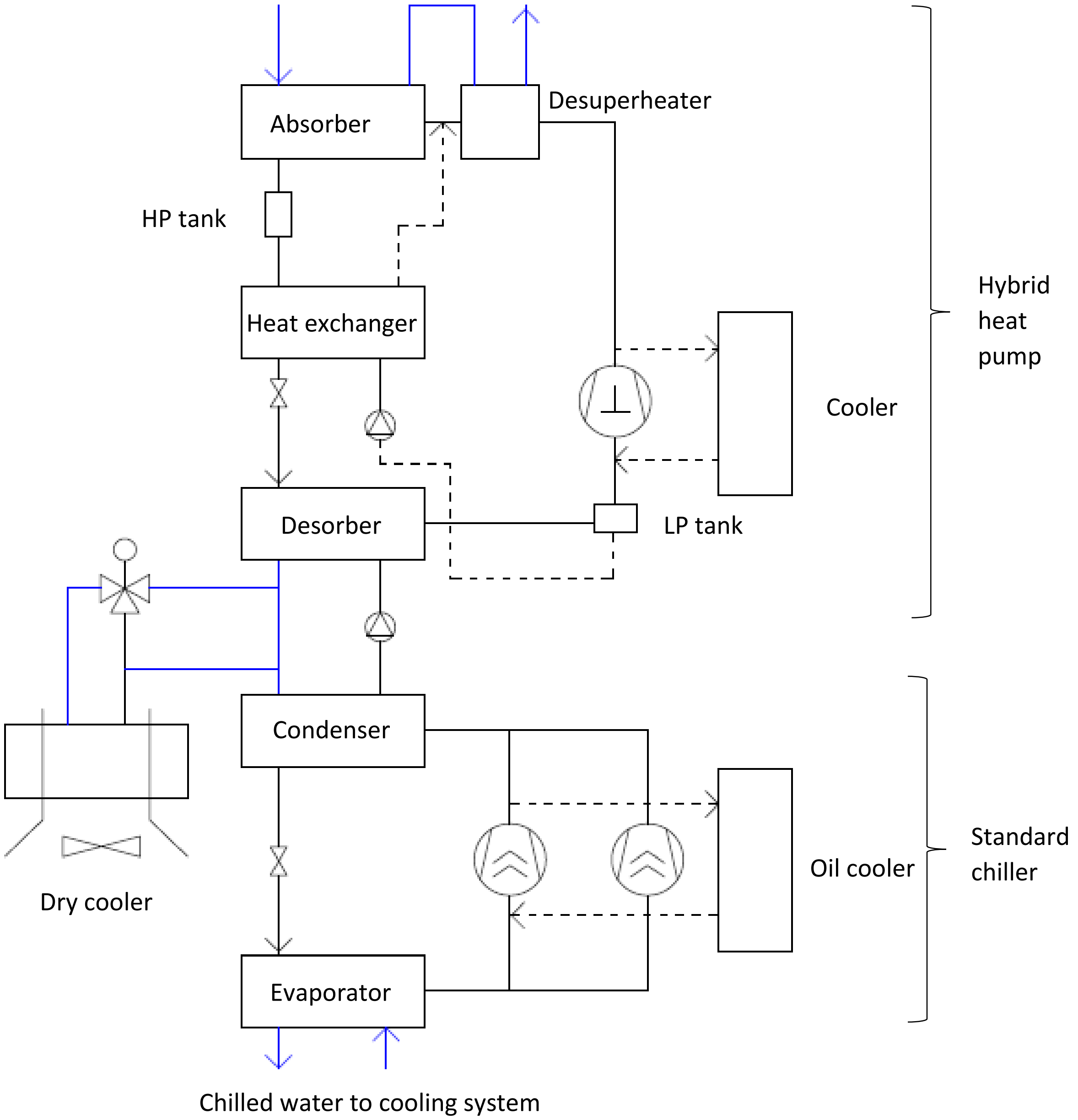

Figure 16. Scheme for the two-stage hybrid ammonia chiller/heat pump [27]

This scheme is equipped with two mono-screw compressors, that are controlled between 2600 and 1500 rpm.

Part load (%)	Tsupply (ºC)	COPh	COPc	Relative enery saving (%)
100	65/80	2.67	3.67	62.55
70	65/80	2.58	3.58	61.24
50	65/80	2.67	3.67	62.55
30	65/80	2.58	3.58	61.24
100	45/60	3.28	4.28	69.51
70	45/60	3.17	4.17	68.45
50	45/60	3.28	4.28	69.51
30	45/60	3.17	4.17	68.45
100	34/40	5.59	6.59	82.11
70	34/40	5.40	6.40	81.48
50	34/40	5.59	6.59	82.11
30	34/40	5.35	6.35	81.31

Table 7. Results for the hybrid two-stage ammonia heat pump/chiller [27]

3.1.2. Life Cycle Cost analysis

Regarding the economic issue, the same structure for the three different alternatives were followed, which is based on an analysis of the capital, operating and maintenance costs.

- Capital costs

First of all, I calculated the annuity factor, which is a calculation that helps in the determination of the amount of eligible withdrawals without possible incurring penalties [29].

$$a = \left(\frac{r}{1-(1+r)^{-n}}\right)$$

Equation 5. Annuity factor

Being r the real interest rate, which I assumed to be 4%, and n the lifetime. As I was dealing with a data center with large capacity, we assumed n to be 20 and 25 years of lifetime so as to see the difference in the annuity factor.

- Operating costs

To calculate the operating costs, I needed to calculate both the annual energy use and the electricity price.

For the electricity price, by using as a resource the "Statistisk sentralbyrå", I got the electricity prices for households. I based our study in the latest possible updated values, although there has been an 8.3% of decrease of the electricity price compared to the previous year in the same period. Anyway, the overall electricity price has remained unchanged since while there was a decrease in the electricity price, there was an increase in the tax on the consumption of electricity.

Electricity prices in the end-user market for the first quarter of 2016

	Øre/kWh
Households. Total price of electricity, grid rent and taxes	87.4
Electricity price	28.7
Grid rent	26.6
Taxes	32.1

Table 8. Electricity prices for households in Norway

In Table 8. Electricity prices for householdsTable 8, we can see the price in øre/kWh for the households, separated by each part of the bill.

The annual operating costs were calculated as follows:

$$Annual\ operating\ costs\ (ACC) = Annual\ energy\ use\ \left[\frac{kWh}{year}\right] * Electricity\ price[\frac{€}{kWh}]$$

Equation 6. Annual operating costs

Where the annual energy use depends on the average COP during one year of operation.

- Maintenance costs

Finally, for the maintenance costs we will assume its value as a 3% of the investment for the heat pump/chiller [22].

3.1.3. Comparison analysis

In this chapter different alternatives of the heat pumps have been compared. The results for each chiller/heat pump were displayed in Annex 2.

- COP comparison

In the following graph it can be seen how the COP varies, in heating and cooling mode at the different outlet water temperatures, and also how it varies in the different systems. It is important to take into the account that this results are at 100%-part load.

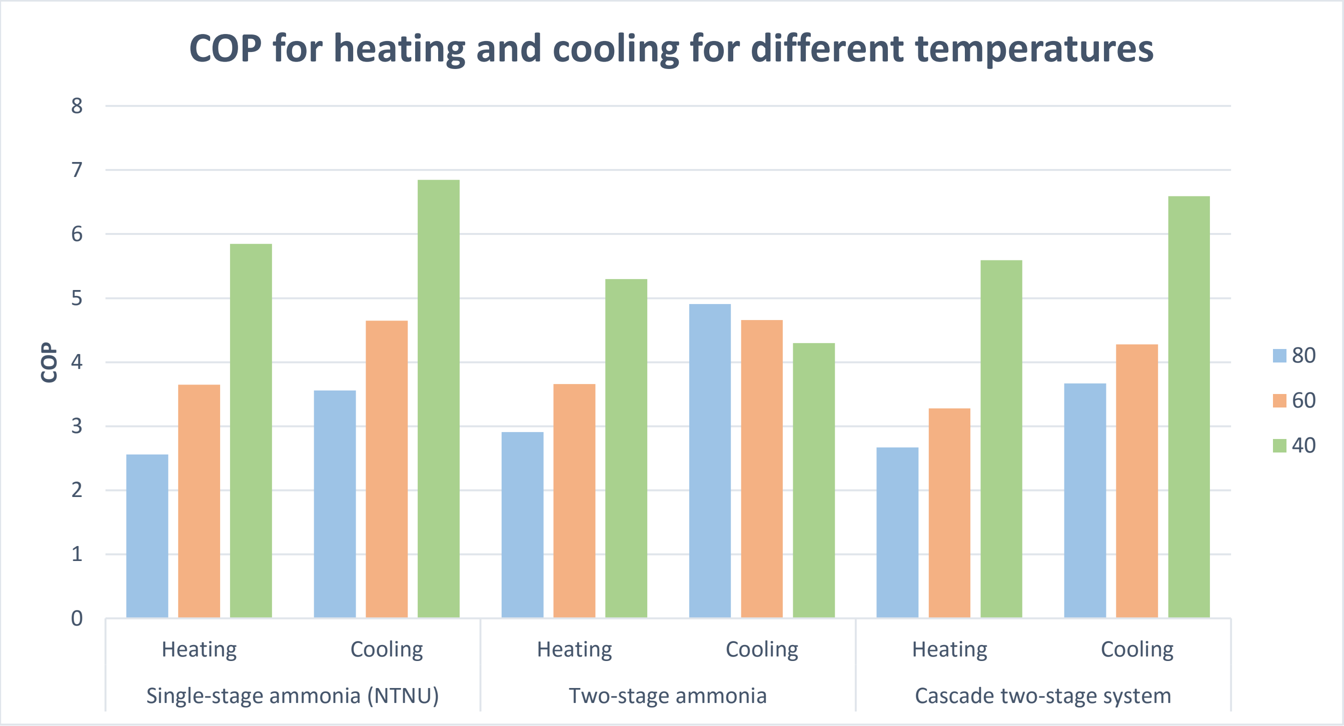

Figure 17. COP comparison for the different alternatives at different outlet water temperatures

As it can be seen in Figure 17, the COP in cooling mode is always higher than the heating mode since:

$$COP_C \approx COP_H + 1$$

Equation 7. COP for the cooling

Making an analysis for each of the alternatives, it can be seen that for the single-stage ammonia heat pump/chiller, the COP is higher as the outlet water temperature is lower. This fact makes sense since, as explained before in

Figure 12, the lower the outlet water temperature is, the higher is the COP value, since the required work from the compressor to achieve the desire temperature is reduced.

Regarding the two-stage ammonia plant, in the heating process the same phenomena happened. The lower the outlet temperature is, the higher the COP. In the cooling part we can see that as the outlet temperature is higher, the COP is also higher, but without much difference, compared to the heating system. This fact can be explained easier by using a graph.

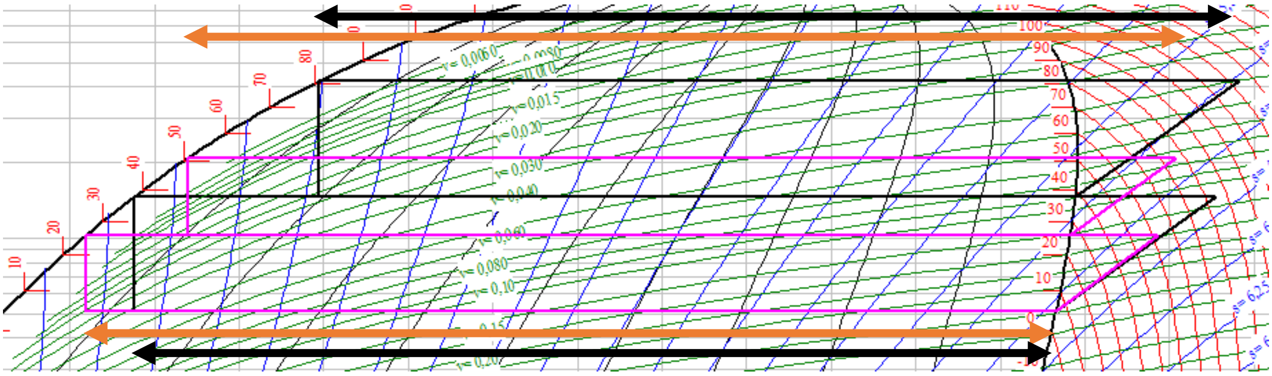

Figure 18. Comparison thermodynamic cycle of a two-stage ammonia for different outlet water temperatures

As it can be inferred from Figure 18 the work that the compressor should do in order to achieve the desired temperature is lower if the required outlet temperature is also lower. But, at the same time, the pressure we are working with is less, which leds to a smaller saturated temperature value. As we can see, as the outlet temperature is lower, the enthalpy value will be even less since it is more placed to the left side, which means that the cooling capacity is higher. To sum up, we understood that it is true that the work done by the compressor is less as the outlet temperature is lower, but there is even more variation in the liquid saturated temperature, which leds us to the conclusion that there is an important increase in the heating capacity and a little increase in the cooling capacity.

Finally, in the cascade two-stage system we can see the same tend as in the one-stage system, which means that the lower the outlet water temperature is, the higher the COP will be.

In our specific case, since we were planning to deliver the heat to the district heating system, the best option was to use the 80ºC outlet water temperature.

- Life Cycle Cost Analysis:

Following the theoretical background, we stated above, we got the results for the capital costs for the different alternatives of heat pumps. As we were working with large data centers, the lifetime could be from 20 to 25. In this project we analyzed both options so as to see the difference in the capital cost that they mean.

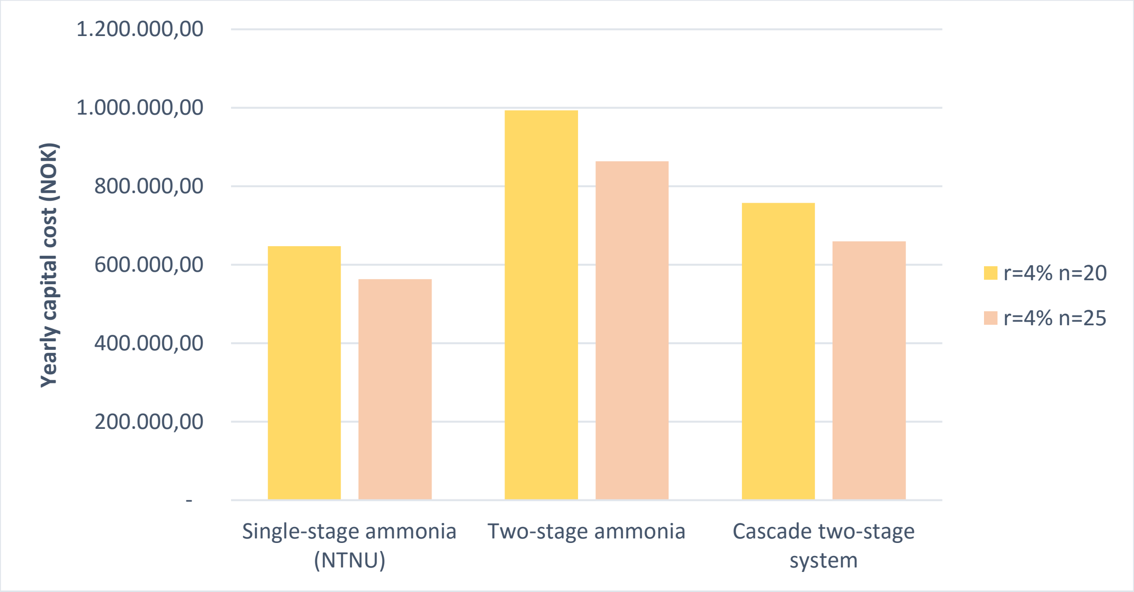

Figure 19. Comparison of the annual capital cost for the different alternatives

The two-stage ammonia is the design with highest capital costs, while the single-stage ammonia is the cheapest option. It is obvious to think that as longer is the lifetime, the capital costs are less, and this difference can be inferred in the Figure 19. This point is really important to take into the account since we are trying to optimize our resources, and economical feasibility has always an important role in all the investments.

Regarding the operating costs, by multiplying the annual energy use times the electricity price in Norway. The results we got are as follows:

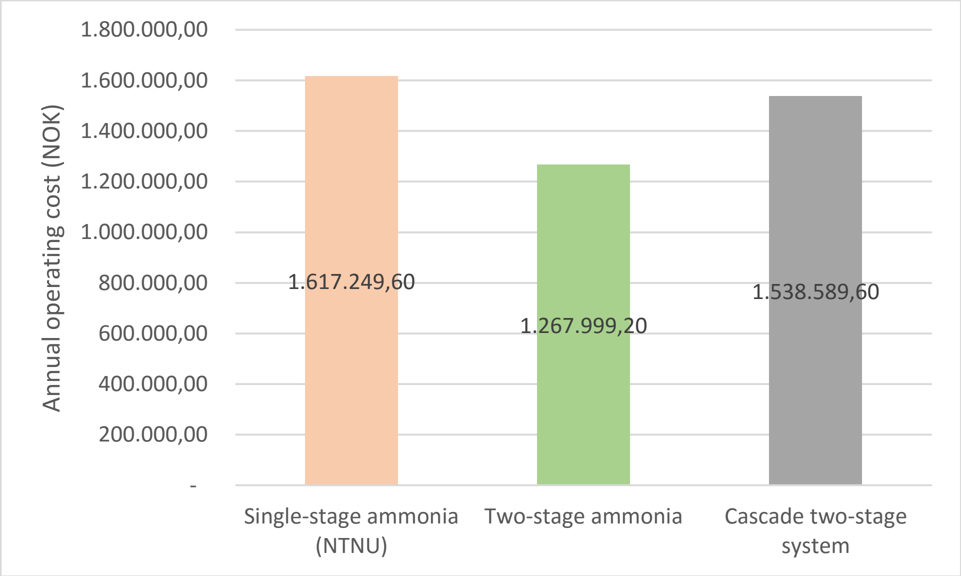

Figure 20. Comparison of the annual operating costs for each alternative

As Figure 20 shows, the single-stage ammonia heat pump has the highest annual operating cost due to the lowest average COP, followed by the cascade two-stage system but with no much difference. Annual operating costs for the two-stage ammonia chiller/heat pump are 20% less compared to the single-stage ammonia.

Finally, regarding the maintenance costs, as stated in the previous chapter, we assumed them to be a 3% of the investment cost.

Figure 21. Comparison of the annual maintenance costs for the different alternatives

Although the comparison between them is similar compared to the investment costs for each of the alternatives, it is important to take into the account that the maintenance costs are way lower compared to the capital and operational costs.

Table 9 summarizes the capital costs, operating costs and maintenance costs for the different alternatives.

	Investment costs (NOK)	Annuity factor	Yearly capital cost (NOK)	Annual operating costs (NOK)	Maintenance costs (NOK)	TOTAL (NOK)
Single-stage	8.800.000,00	0,0735	646.800	1.617.249,60	176.000,00	2.550.299,6

ammonia (NTNU)						
Two-stage ammonia	13.500.000,00	0,0735	992.250	1.267.999,20	270.000,00	2.530.249,2
Cascade two-stage system	10.300.000,00	0,0735	757.050	1.538.589,60	206.000,00	2.501.639,6

Table 9. Summary of the Life Cycle Cost analysis for the different alternatives

As a conclusion we could state from these values that the cheapest option would be the two-stage ammonia, while the most expensive one would be the cascade two-stage system, but without much difference between them.

4. Methodology

4.1. Load definition

- Cooling load

This project deals with the issue of using the excess heat from large data centers. So as to be able to do the different calculations to study the viability of different technologies, cooling demand of data centers was required. Since there is not much information available, an approximate curve has been designed by means of a deterministic and a stochastic part.

- Deterministic part:

It is important to highlight that since we were working with data centers, it is a fact that the servers are working during the whole year, so the cooling demand over the year is more or less constant. To justify this constancy, we analyzed the cooling demand that was met in one of the buildings in Gløshaugen at the Norwegian University of Science and Technology (NTNU) in Trondheim, Norway.

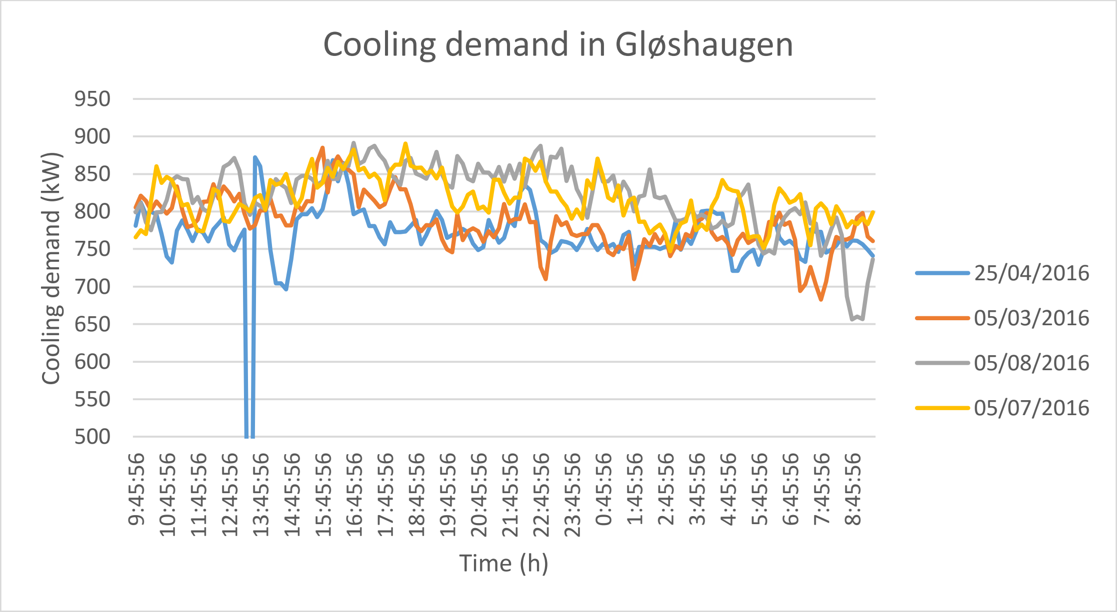

Figure 22. Cooling demand variation in Gløshaugen

This graph refers to the cooling demand of a building in Gløshaugen. There is a little variation during the year due to, for instance, computers usage, i.e. people use more computers during the day rather than during night. Also, since the cooling demand is more or less independent of the ambient temperature, we cannot see clear differences between winter and summer seasons in the cooling demand. This graph only displays four examples, all the measurements can be found in Annex I.

This cooling load is expressed by hours during the day. Unfortunately, the system in Gløshaugen does not record the data. So, as it just shows the daily demand we could just express the graph on a daily basis, but still we can see with the data we took from different days during two months, that there is no huge variation between them. It can be seen one point that goes straight to zero and that has been deleted since it was not our interest of study. This variation may have been due to some failure in the system or even a momentary turn off in the system.

Nonetheless, our main aim was to find a curve that had to be somehow similar to the trend line that follows the above graph.

Then, summing up both conclusions we understood that we should get a graph that yearly must accomplish a curve as it appears in Figure 23, but at the same time it should be smoother, since, as we justified before, there is no huge variation of the use of the servers in a data center.

Seeing this curve, we could get to the conclusion of a very similar function that accomplishes this curve: the cosinus.

The cosinus function follows a sinusoidal curve, and with this curve we could then vary the smoothness and then get the function we need.

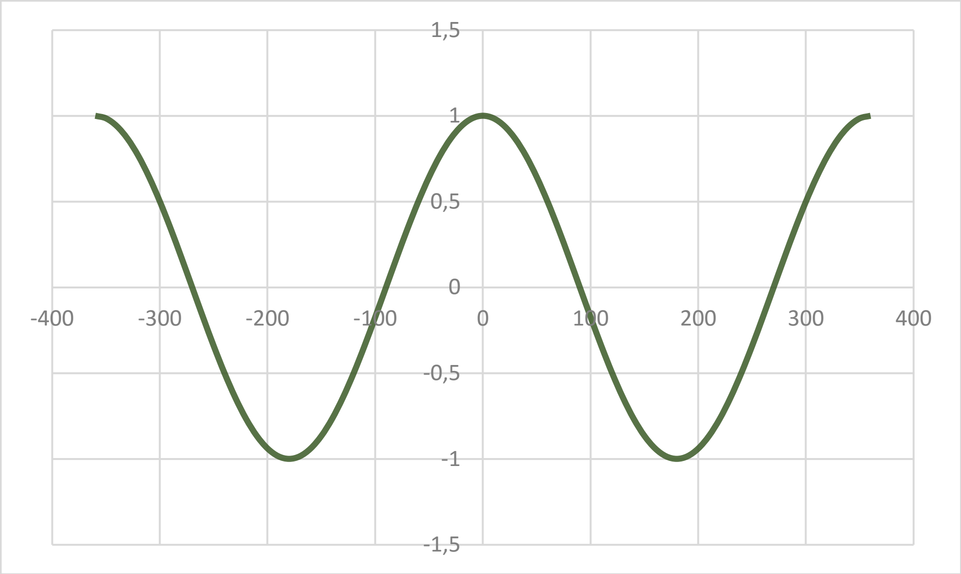

Figure 23. Sinusoidal curve from cosinus function

This cosinus curve is expressed in degrees, from -360º to 360º, this way we can understand how this function is represented.

We define then, the following function which will give us the deterministic part of this approach:

$$Deterministic\ approach = \cos(B - M)$$

Equation 8. Deterministic approach

Where:

- B: the time, expressed in hours.
- M: a constant that will be defined by us as the day of maximum demand over the year.

This way we will have a curve that will follow the cosinus one but placing the highest point in the maximum day of demand. In our case we placed the maximum demand in the day 200 out of 365, which will be around June-July.

- Stochastic part:

Since this is an approach, some random variables were needed due to the fact that there is no linear curve that follows the cooling demand. This means we had to deal with some way of defining the variations we could get in a real case. This part is done by using the random function in Excel limited to numbers from 0 to 1 and assuming a standard deviation of 0.45.

Also, these random values will follow a normal curve in each of the random point we will define (one for each hour of the year), as follows:

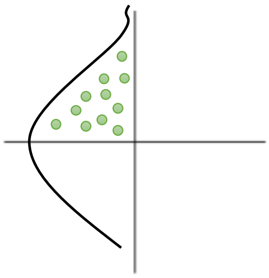

Figure 24. Placing of random variables following the absolute value of a normal curve

As it can be seen in Figure 24, the random values are just placed in the upper part of the normal curve. As justified above, the cooling demand does not have huge variations, by using the random variables taking into the account the possible negative points, the variation that was created was too big compared to the one a real data center has in reality. The solution we proposed to this problem was then limiting these random variables to its absolute values, this way the variation is one half of the one that could have been.

Finally, we summed up the parts so as to get our final approach by following the function:

$$Total = Yearly\ average\ value + Deterministic\ part + Stochastic\ part$$

Equation 9. Total theoretical statistical approach

As this project studied the viability of exporting the excess heat from large data centers, we used as the yearly average value of the cooling demand information we got from a data center situated in Finland from a company called “Nivos Oy”, which collaborated in this project. Their maximum cooling demand was 2,6 MW, so we assumed as the yearly average value as one third of this value.

Then the remaining formula will be as follows:

$$Total = 0{,}87 + \cos(B - M) + Random\ variable$$

Equation 10. Statistical theoretical approach

And the resulting graph will then be:

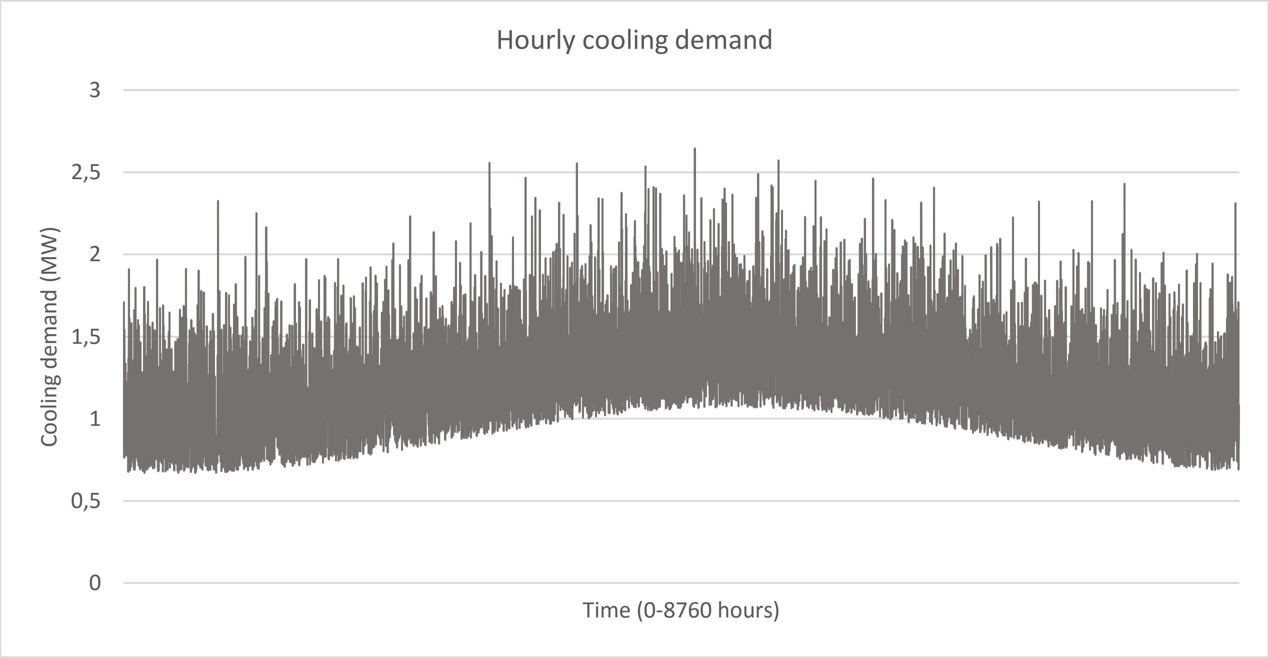

Figure 25. Defined computer cooling demand over a year

Notice in Figure 25 that we limited the random variation to an absolute value. This assumption is taken due to the avoiding of huge variations in cooling demand, since we already justified that the variation is not supposed to be big.

- Heading load:

So as to define the heating load we pursued the same procedure we followed for the cooling load. We determine both deterministic and stochastic part, but in this case placing the maximum demand day in a different one than we used for the cooling demand. Also, in this case we assumed a yearly average demand of 1,1267 MW which corresponds to an approximately 75% more than the cooling demand previously analyzed. This assumption is based on studying the differences between heating and cooling demand in different systems such as the system found in Gløshaugen at the NTNU. In Gløshaugen, the cooling demand represented an approximately 60-75% of the heating demand (depending on the part load) [26]. Also, we stablished the smooth parameter up to 1,2 since there is an important variation of the heating demand over the year, and we were working with a standard deviation of 0,5.

Since our heating load was referred to a group of buildings, it was easy to understand that in cold seasons, such as winter and autumn, the heating load would be higher than in summer periods. Moreover, by analyzing different load curves [30], we found out that in the hottest season the heating load was zero, since there was no heating demand from the buildings.

In Trondheim, we found a seasonal temperature variation as follows:

	TEMPERATURE (ºC)
Winter	-2,67
Autumn	0,17
Summer	11,67
Spring	9,67

Table 10. Average seasonal temperatures in Trondheim

Using this data that was found in the yr.no website [31], we could find the following graph which shows the trend line the temperature displaces over the year. It is important to take into account that when constructing the heating load curve, I used the temperature data since the space heating demand is proportional to the variation of temperature between the ambient and the room.

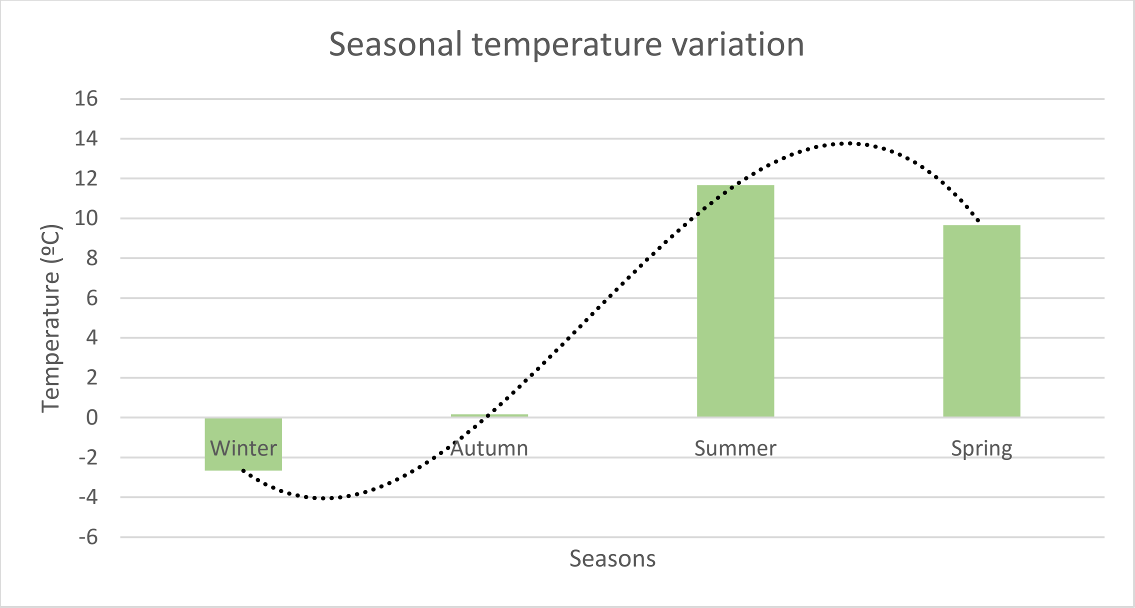

Figure 26. Variation of the seasonal temperature

Figure 26 shows the average seasonal temperature variation over the year in Trondheim. So, it is easy to think that, as the lower the ambient temperature is the higher the heating demand is, the curve of the heating demand should somehow follow a curve on the right opposite way as in Figure 26. So this means that the higher point will be found in winter months while the lowest heating demand will be placed in summer months. In our statistical approach we placed the maximum demand day in the day 35, which would mean first week of February.

Carrying out all the statistical approach that was explained in the above headland, the resulting heating demand graph is:

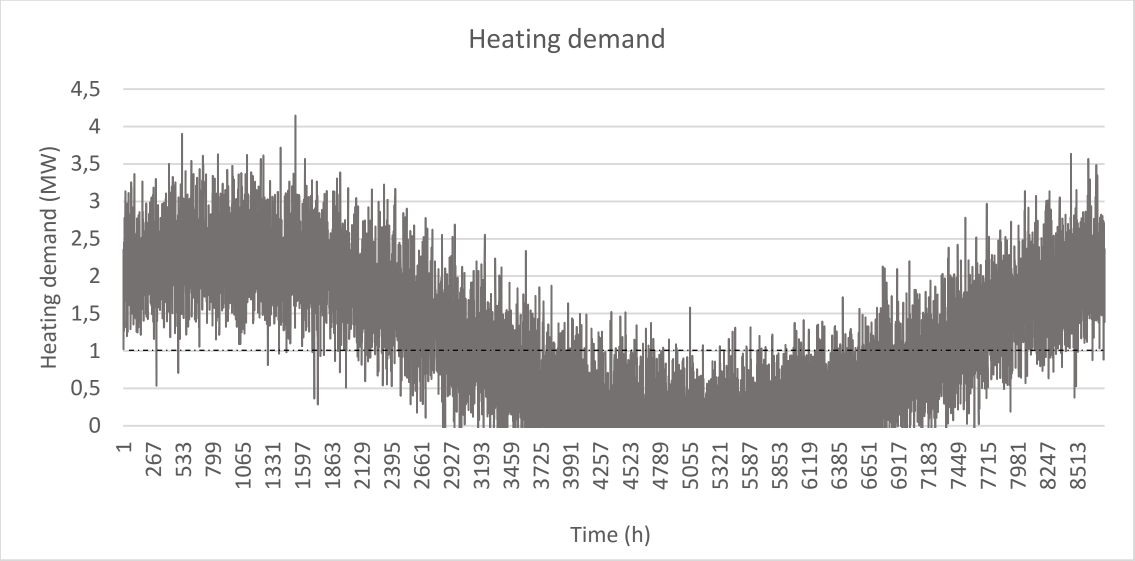

Figure 27. Defined heating demand for a group of dwellings over a year

As it can be seen in Figure 27, the curve follows a trend that is higher un winter seasons and lower in summer seasons. This is due to the fact that people living in buildings do not need such heating demand to satisfy their needs since in summer space heating for instance is not needed, and the hot water demand is lowered drastically.

The area under the black line is the heating demand referred to the DHW, while the rest is referred to the space heating.

The graph monthly based has also been analyzed.

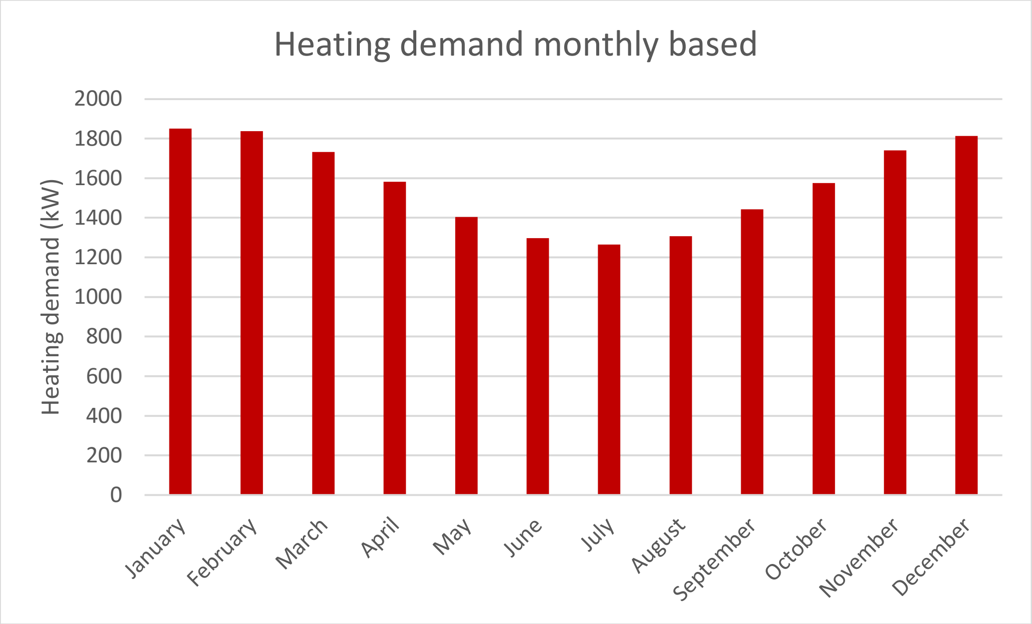

Figure 28. Monthly based heating demand of a group of buildings

The heating demand is less during the warmer months. This fact makes sense due to the dependence of the ambient temperature the heating demand may vary.

4.2. Chiller/Heat pump design

As it has been stated before, this project main aim was to find the best possible solution to use the excess heat from a large data center so as to use it afterwards into the district heating system in Trondheim, Norway.

A heat pump/chiller seemed to be the best option as at the same time we were using the excess heat from the data center, so we are using heat that otherwise would have been rejected to the ambient, and as we were removing it from the data center, it caused a decrease in the cooling demand of the data center. This process can be seen in Figure 29.

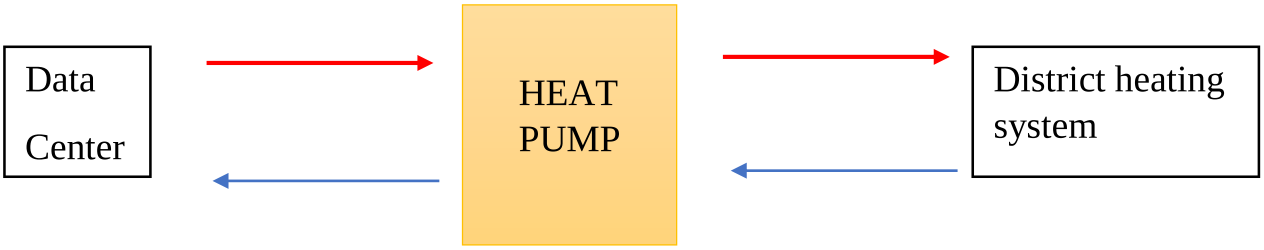

Figure 29. Scheme of the main functioning of the system

The red lines can be understood as the heating system while the blue ones would be the cooling system.

Once we calculated the COP and the Life Cycle Cost analysis for the three alternatives that were proposed in the previous chapter, we could decide which one would be the best option for our system.

Based on the results we got, although the higher COP (both in heating and cooling) is achieved by the two-stage ammonia heat pump, it is also the most expensive one. Whereas for the single-stage ammonia heat pump, the COP achieved is a little bit lower than the one with the hybrid cascade two-stage system. Between these two options, we had then to take a look to the life cycle cost analysis, where we easily found out that the best option would be the one-stage heat pump system since, even though the COP is slightly lower, the total costs for buying it, make it run and maintain it are less.

Since this chiller/heat pump is designed to achieve an outlet water temperature of 80ºC, the heat will be delivered to the return pipeline in the district heating system.

4.2.1. Working fluid selection

An analysis in order to choose the best working fluid option has been carried out in this project. Basing the first approach on the ozone depletion potential (ODP) factor and the global warming potential (GWP) factor some group of working fluids can be discarded. Since chlorofluorocarbons (CFC) and hydrochlorofluorocarbons (HCFC) have a large value of ODP and GWP, it is not recommended to use them.

Regarding the hydrofluorocarbons (HFC), although they have a large GWP value, their ODP is equal to 0. An interesting working fluids of this branch could be R134a, but there are some studies that are planning to replace it with an hydrofluoroolefin (HFO). Anyway, HFCs are normally used for low capacity installations besides R134a, and in this case I needed to find an option for high a capacity installation.

Finally, natural working fluids were taken into account. They seemed to be a really interesting option since its ODP is zero and its GWP is more or less 0. The best options in this branch would be R744 (CO_2), R290 and R717 (NH_3). An analysis of each of them has been studied.

- R744 (carbon dioxide)

CO_2 is non-flammable and non-toxic but its main disadvantage for this study was the need of relatively high operating pressures.

In order to make it clear, a pressure enthalpy diagram is displayed in Figure 30.

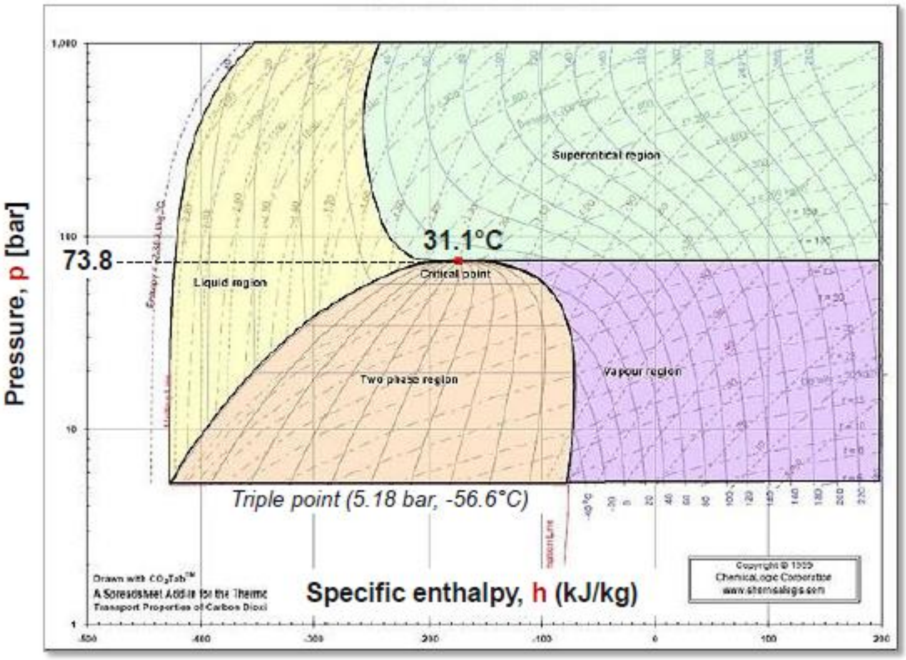

Figure 30. Pressure enthalpy diagram for R744 [22]

The critical temperature is 31,1ºC and the critical pressure is 73,8 bar. This high operating pressure ends up in a high vapor density and a relatively low pressure ratio. Due to low critical temperature and high critical pressure as can be seen in Figure 30, heat rejection is met at supercritical pressure in most applications, which consequently brings the system to a large expansion loss, but still a low superheating loss [22].

In order to see a comparison between the different pressures met in different working fluids, Figure 31 has been added:

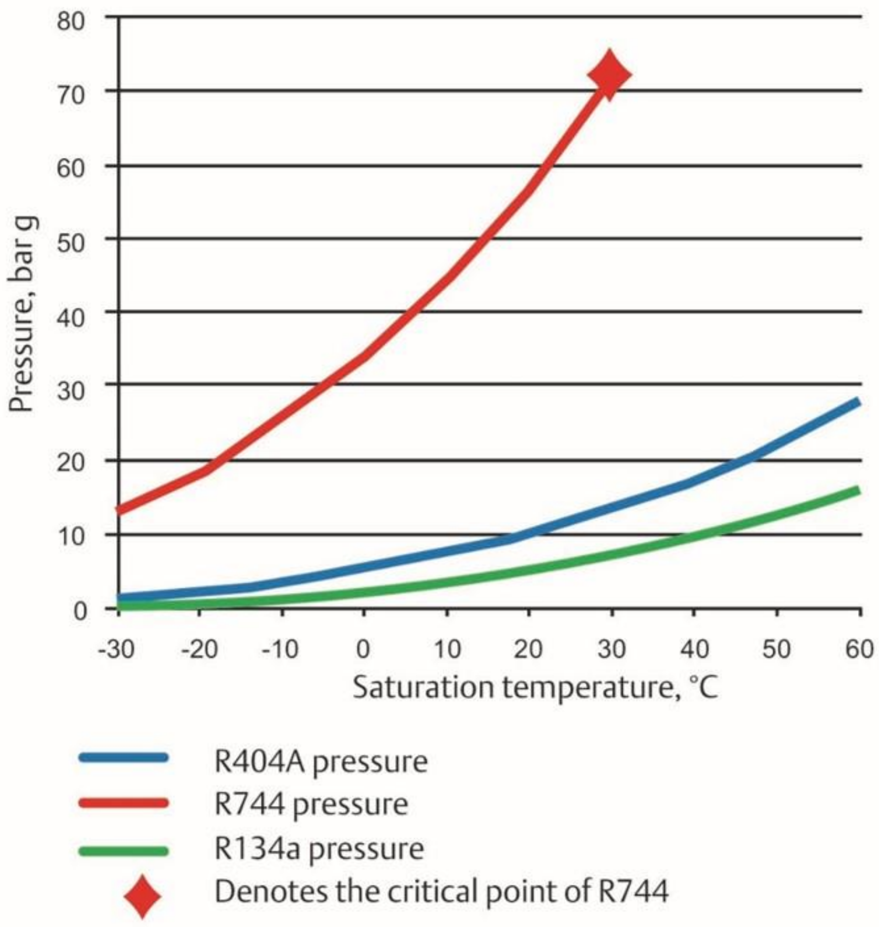

Figure 31. Comparison between different pressures with different working fluids [32]

It can be seen the huge difference between the different pressure for the different working fluids. R744 is then known as a working fluid that needs to work with high operating pressures.

Also, the temperature fit with R744 as a working fluid is really affected by the average temperature during heat rejection, so it means the higher variation of temperature we have, more affection to the COP we would find, since in this system, there is no much difference between the inlet and the outlet cold temperature, but there is a huge difference between the inlet and the outlet hot temperature.

To sum up, since we are dealing with an important change of temperature, R744 does not seem to be the best option to our system since its operating pressure is too high, and the temperature it is able to achieve is not needed to our system since we are trying to supply heat to the DHS.

- R290

Although being a natural working fluid, and having OPD and GWP close to zero, the maximum outlet water temperature from the condenser with this working fluid is 50ºC. This fact makes it impossible to work with it, because of the risk of legionella, and also because since my goal is to deliver heat into the DHS, the outlet water temperature must be higher so as to respond to the DHS temperature levels in Norway.

- R717 (ammonia)

The main disadvantage of ammonia is its high level of toxicity, which makes it mandatory to be careful while working with this fluid, in case of failure.

Ammonia has a very high critical temperature and a high critical pressure. This high critical temperature makes it suitable to be used in high temperature heat pump applications. Due to its high specific enthalpy of evaporation, ammonia can achieve a high cooling and heating capacity, which make it ideal to work with.

$$Q_e = q_e * m_R$$

Equation 11. Evaporator capacity

Following the equation, the higher the enthalpy of evaporation, the lower the mass flow rate, and so the vapor and the liquid density are also much lower, compared to other working fluids.

Regarding the dimensions of the pipelines and the valves, the suction pipelines are between 30 and 60% smaller than HFCs at -15ºC and equal ${}^{\Delta t}/_{\Delta p}$. In the liquid pipelines, it is between 60 and 80% smaller than that of HFCs at +0ºC and equal ${}^{\Delta t}/_{\Delta p}$ [22].

Based on what is explained above, and compared to the other working fluids I previously analyzed, ammonia seemed to be the best option.

4.2.2. Components

Are we are basing our case to an example of heat pump/chiller similar to the one that is actually installed in Gløshaugen, we assumed the components we were using would be the same so as to be able to do the comparison with our concrete heating load.

- Compressors:

When choosing the most suitable compressor for each system, some requirements must be taken into account:

- Operating limits (temperature and pressure)
- Reliability and safety
- Energy efficiency at full load and part load
- Noise and vibration
- Specific investment costs per kWh

In this chiller/heat pump two compressors were used. Since I was dealing with large loads, I first had to discard to type of compressors which volume capacity is too large or too low to our system. This means scroll and turbo compressors are not suitable since they have low and high compressor volume which we do not need to our system [22].

It is true that piston compressors would be good to work under short-term repeated stress, but using them between a low and a medium compressor volume. On the other hand, screw compressors work from a medium to a high compressor volume which would be the best option to our load. Since, as we said above, I was dealing with large heating demand, screw compressors seemed to be the best option.

I used two twin screw compressors, with a 40 and 52 bar compression (it gets higher as the temperature gets higher), for the unit 1 and the unit 2, respectively [26].

Since the main purpose is to control the heating capacity in order to be able to meet the heating demand in the buildings, the mass flow rate for the working fluid must be controlled by changing the suction volume of the compressor.

$$\dot{Q}_c = \dot{m}_R * \Delta h_c = \dot{V}_s * \rho_1 * (h_2 - h_3)$$

Equation 12. Condenser capacity

The option so as to control this volume was using a slide valve control, since it is meant to be used with screw compressors.

- Heat exchangers

We used plate heat exchangers with frame as they have a high heat-transfer coefficients, compact size and their cost effectiveness [33]. Our concrete plate heat exchanger had a high U-value which we assumed to be around 2500 W/m.

Concretely, in the evaporator a semi-welded plate heat exchanger with frame of 16 bar of pressure was used. Its plates were made out of stainless steel and it was insulated so as to prevent condensation [26].

Regarding the condenser, the oil cooler and the sub-cooler, we used a plate and shell heat exchanger of 16 bar.

- Oil system

In our system, an oil separator, an oil pump and an oil cooler were used. The oil pump had an oil flow rate of 85 l/min. The oil separator was used so as to, as its own name indicates, to separate the oil of the working fluid, the oil pump was used to impulse the oil and to have the proper working pressure. Finally, the oil cooler was used to cool the oil, and at the same time, this rejection of heat was used to warm the water up [26].

- Dry-cooler

A 1200 Kw dry-cooler was installed. As it has low revolutions per minute, the noise created is really low. The dry-cooler was installed in order to reject the excess heat that cannot be re-used in the district heating system to the environment.

4.2.3. Interface with supply

In this thesis, an hourly based model was carried out. The initial data from where I had to start my analysis was defined myself based on some data collected from the laboratory in Gløshaugen. The measurements were made in a daily base, since the data in Gløshaugen was not saved in the system itself. By an analysis of the constancy the cooling demand followed in the data center placed in Gløshaugen, at the NTNU in Trondheim, a cooling demand was designed made out of a statistical approach. The same way, the heating demand was also designed, but this time based on some examples of typical heating demand of groups of buildings [30].

Due to this theoretical approach, graphs appearance is supposed to be more idealistic than the reality. This fact is difficult to be seen in the hourly cooling and heating demands due to some random iterations done so as to offer the more possible realistic graph. Even though, this ideality can be seen in the load duration curves since they are quite linear compared with a real one with measured values.

By using the data explained above, some analytical study for different alternatives of chiller/heat pumps was carried out.

Some comparisons between the heating capacity of the resulting chiller/heat pump and the heating demand of a group of buildings were made in order to figure out the amount of energy that can be sold. The most interesting key point in this procedure is too see when is it worth it to export excess heat. A discussion regarding whether is better to supply the excess heat (supply or return pipeline) was also exposed.

5. Results

In this chapter some results are displayed. Based the work on the initial theoretical and statistical approach I proposed for the cooling and the heating demand, some graphs were displayed in order to see clearly the result of this analysis.

5.1. Load duration curve

The load duration curve is determined from the cooling and heating demand approach that I previously explained. In this specific case, the superposition of both demands is as shown in Figure 32.

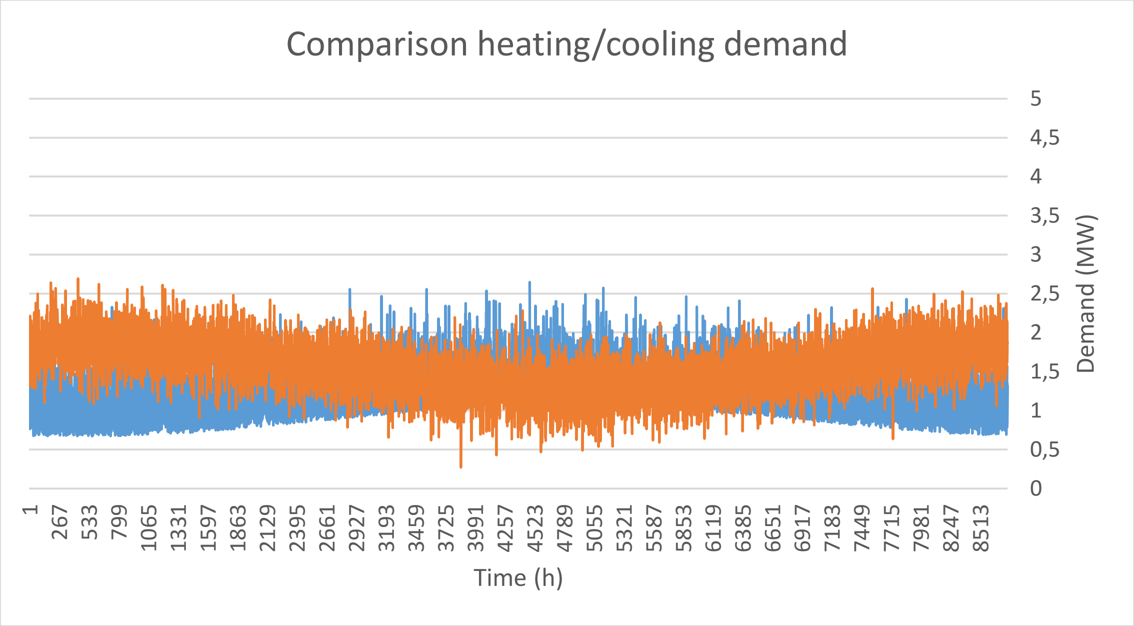

Figure 32. Superposition heating and cooling demand

In this case, the heating and the cooling demand met quite good, which makes think that it could be a good system, since a pretty big part of the heating demand is achieved with the cooling demand.

In this theoretical approach some assumptions were made as commented in the previous chapter “Load definition”. On one hand there is the cooling demand from the data center, and on the other hand the heating demand from a group of buildings.

First, the graph for the load duration curve of the cooling demand of the data center is displayed:

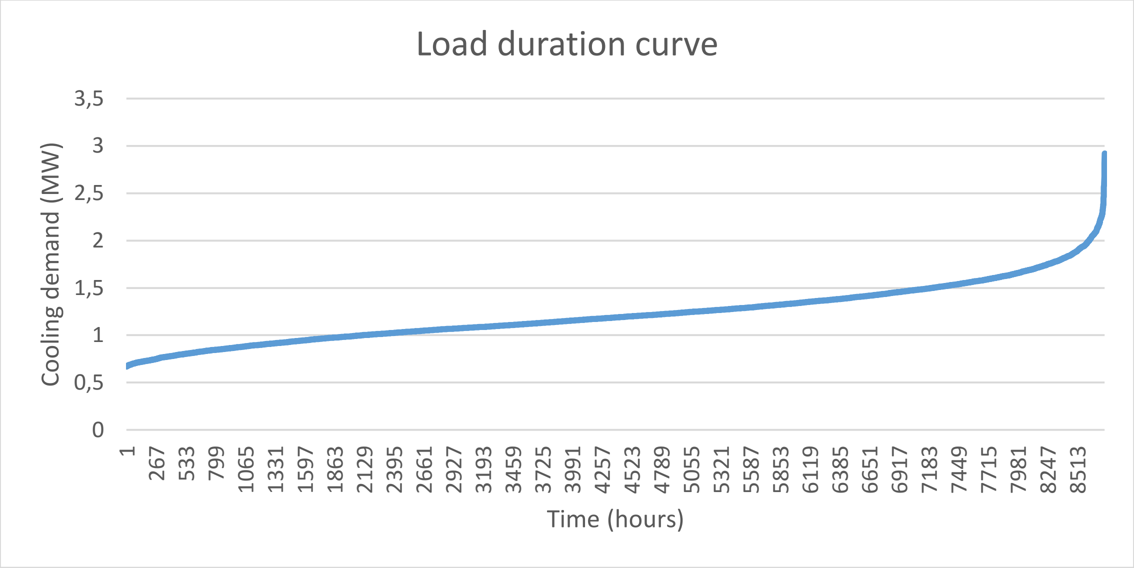

Figure 33. Load duration curve for the cooling demand of the data center

In Figure 33, the amount of cooling demand can be seen through a whole year. The cooling demand will never be zero, since I was working with computer cooling, and computers and electronic devices are used all over the year. An interesting aspect of this graph is its constancy. The curve is almost flat most of the year, but during a very short period it is very steep, what means the cooling demand is more or less constant over the year, but in reality it still has some peaks, that is what mainly makes the difference.

The result of my statistical approach for the heating demand left a load duration curve as showed in Figure 34.

Figure 34.

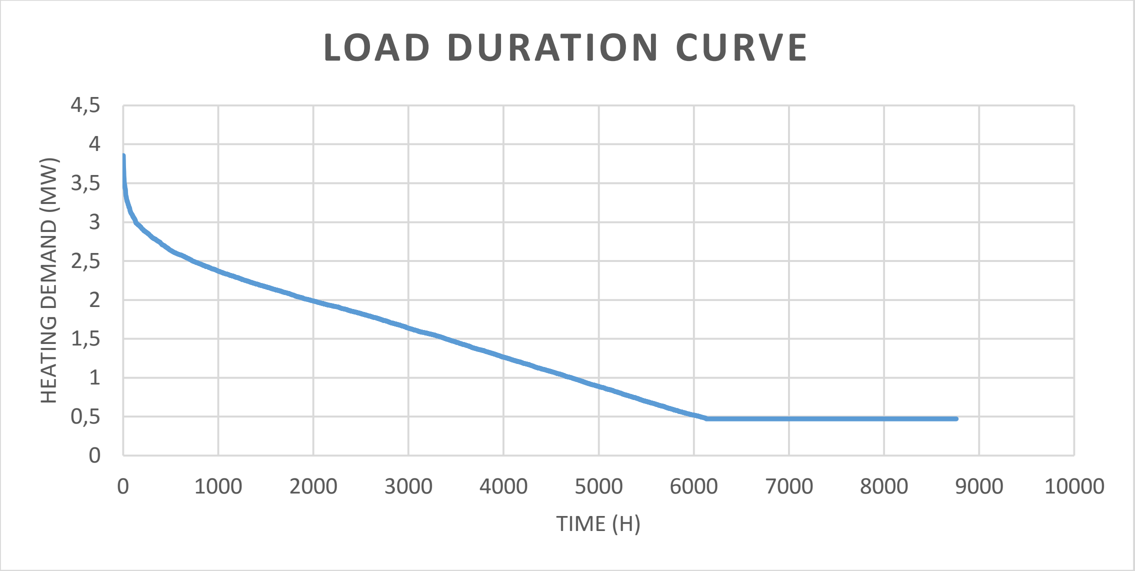

Figure 34. Load duration curve for the heating demand of a group of buildings.

From Figure 34 it is important to comment that its linearity is caused by the calculations that were carried out from a theoretical approach due to lack of data. It may normally have a similar curve but not as linear as it is in Figure 34.

A maximum heating demand of approximately 3,8 MW is met in this graph and a space heating demand is found about 6200 hours/year, since it is not supposed to be any heating demand during the warmer months in summer, for instance. The remaining demand is understood as the DHW needed from the buildings. Some sensitivity analysis regarding Figure 34 are analyzed in the following chapter.

By using the results from the one-stage ammonia chiller/heat pump of the heating capacity of the condenser, I got the following graph:

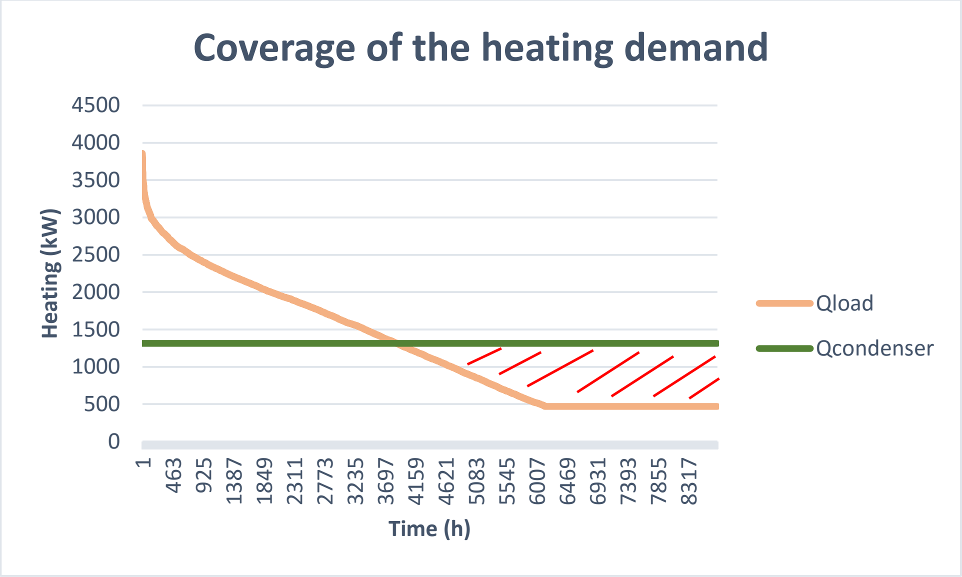

Figure 35. Coverage of the heating demand with the condenser capacity

In Figure 35, the area under the green line is what the condenser is able to achieve. That means that this is the heating demand that this system is able to meet.

The DHW is completely covered with the heat pump and to know how much amount can it be covered from the space heating; some calculations were carried out by following Equation 13.

$$A = \frac{y_{i+1}}{2} * (x_{i+1} - x_i)$$

Equation 13. Area under the curve

By using the Equation 13, the area that is found under the condenser heating capacity is 7583828,55 kWh. This is a 71,53% of the total heating demand.

Analyzing deeply Figure 35, I highlighted the red zone which would be excess heat. This excess heat would be rejected to the ambient. This graph show that still there is an important amount of heat that would be rejected. A sensitivity analysis will be done in order to show how the graph would vary in case of variation of the heating capacity of the condenser.

Here it is important to comment that the cooling load I defined was thought to be for a large data center, and that the heat pump data I used comes from the example of chiller/heat pump installed at the NTNU, while the heating load was defined for a group of buildings. That is why it is possible to cover that much amount of demand.

5.2. Sensitivity analysis

Some sensitivity analysis is carried out in order to analyze possible variations in the above statements.

The main goal of this project is to identify when is it worth it to export heat. To answer this statement, some variations in the heating capacity of the chosen chiller/heat pump has been analyzed in order to see the differences that would be found in case of variation.

An approximate 45%-decrease of the heating capacity is studied.

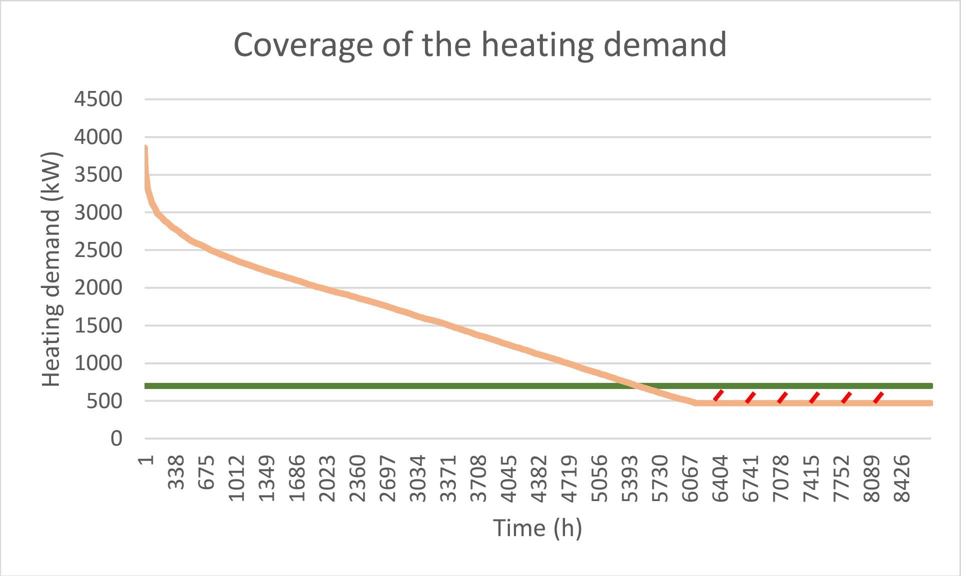

Figure 36. 45%-decrease of the heating capacity of the condenser

In Figure 36, the heating capacity of the condenser has diminished by 45% compared to the initial heating capacity. This means that with this chiller/heat pump would still be possible to cover the entire DHW demand, but a less percentage of space heating. At the same time, the excess heat that is rejected to the ambient is also less (see the red zone), i.e. that the utilization of the excess heat is much better than in the original example. This will increase the profitability and lead to a lower LCC and the annual supply of useful heat also increases. The larger the difference between the available heat from cooling and heating demand in the district heating system, the less profitable the investment in heat recovery is, i.e. if the available excess heat from heat recovery is smaller than the average heating demand in the district heating network, the heat recovery potential is at its maximum.

A second iteration was made. This time, a 70%-reduction was met.

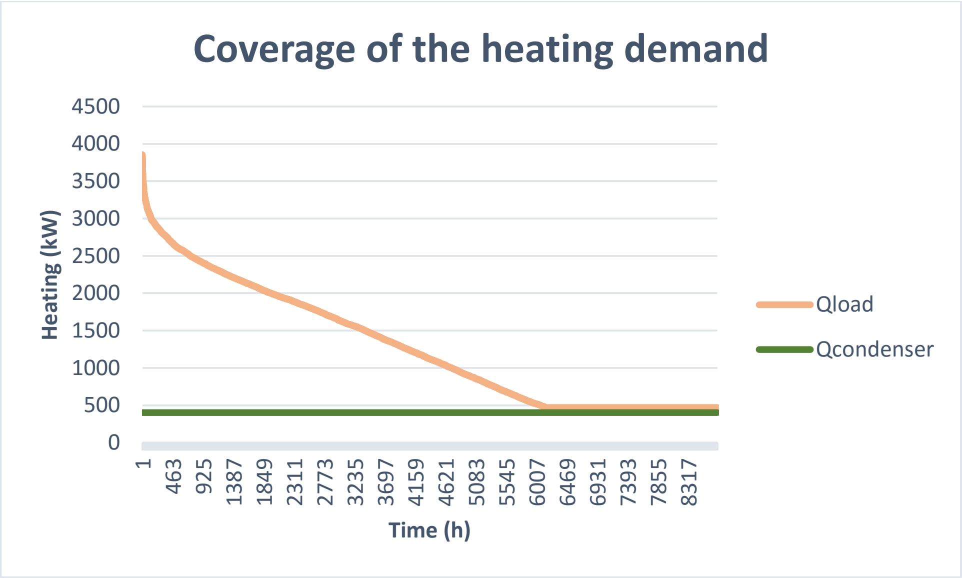

Figure 37. 70%-reduction of the heating capacity of the condenser

In Figure 37, no red zone is found. This means there is no excess heat that needs to be rejected to the ambient. As it can be seen in this case 100% of the excess heat is recovered and the profitability is at its maximum (i.e. lowest possible LCC).

6. Future prediction

It is a fact that data centers throw away huge amounts of waste heat into the environment. Many studies show that it is worth it to transform this waste heat into useful temperatures and though reuse them into i.e. district heating system. This process is ecologically and economically interesting, since we reduce the amount of waste heat rejected to the environment at the same time that we cover a high percentage of heating demand of different establishments. Economically interesting also because district heating system is cheaper than the normal electricity price. Also, when using the waste heat in i.e. space heating, CO_2 emissions can be reduced.

As a conclusion, this type of process seems to be in a starting point, and in views of future, it seems to be a key point to data centers to reduce its costs, and at the same time, a good way to save energy waste.

7. Conclusion

In this thesis, the possibility of using the excess heat from a large data center has been studied. The constant increase of the demand in data centers, made it interesting, as it may be a new way of transforming waste energy into useful energy.

Both the cooling demand for the data center and the heating demand for a group of buildings was estimated by using some statistical approach. From these demands, different load duration curves were displayed.

Regarding the cooling duration load, since I was dealing with computer cooling from data centers, cooling demand is present all over the year. Whereas the heating duration load is differentiated into two different heating modes: space heating and DHW. Space heating was limited to approximately 6200 hours of demand over the year, and DHW was thought to be needed during the whole year.

Coming up next, an analysis of different alternatives of chiller/heat pump and working fluids was done. From the efficiencies and a life cycle cost analysis of each of the alternatives, I came up with the solution that using one-stage ammonia chiller/heat pump was the best option.

A comparison between the heating capacity that the chiller/heat pump is able to achieve and the heating demand of the group of buildings was exposed in order to see if it was worth it to export excess heat. Based on the results I got, individual buildings were served during the whole year with DHW and partly space heating. The heating capacity of the chiller/heat pump covered approximately the 70% of the heating demand.

Some sensitivity analysis was also carried out. It showed whether it was a good option to export excess heat or not. Thanks to the different iterations exposed, a clear view of how much can the heat pump achieve of the heating demand is displayed in the different cases.

To sum up, exporting excess heat is a key point due to the saving of the heat that otherwise would have been rejected, but also for the decrease in the cooling demand for data centers. At the same time, with this method district heating network is used, which is likely to become more and more present in next years.

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ANNEX I

In this annex, all the measurements of the cooling demand of Gløshaugen are displayed. The main aim of this chapter is to justify the constancy of an example of a cooling demand from a data center. Some variations in the limits of the y-axis were made in order to see clearly the variations.

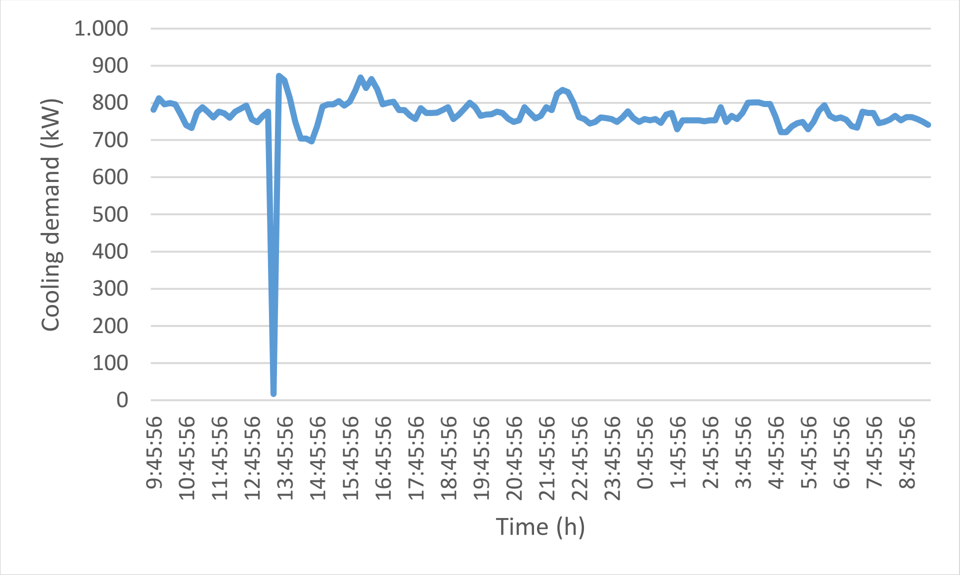

Figure 38. Cooling demand in Gløshaugen (25/04/2016)

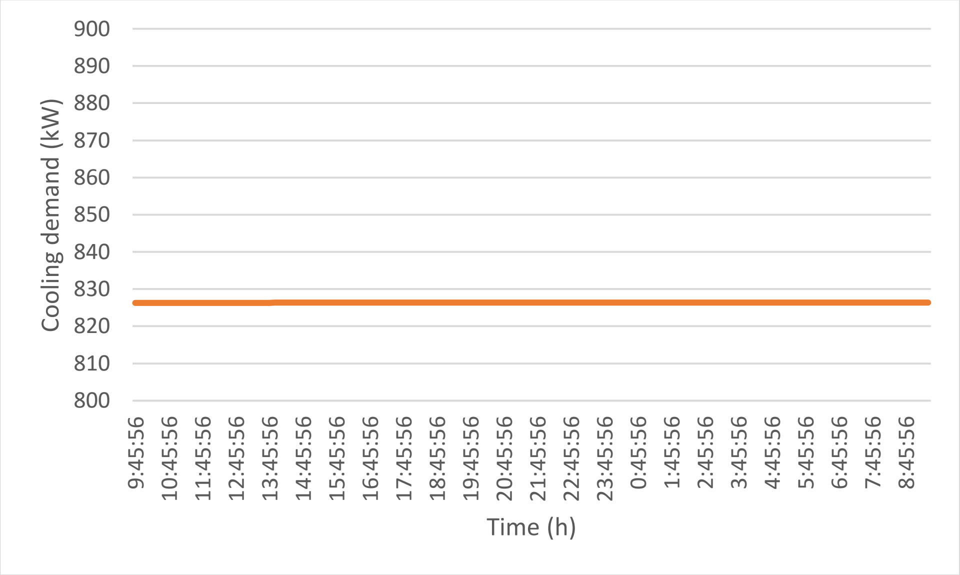

Figure 39. Cooling demand in Gløshaugen (27/04/2016)

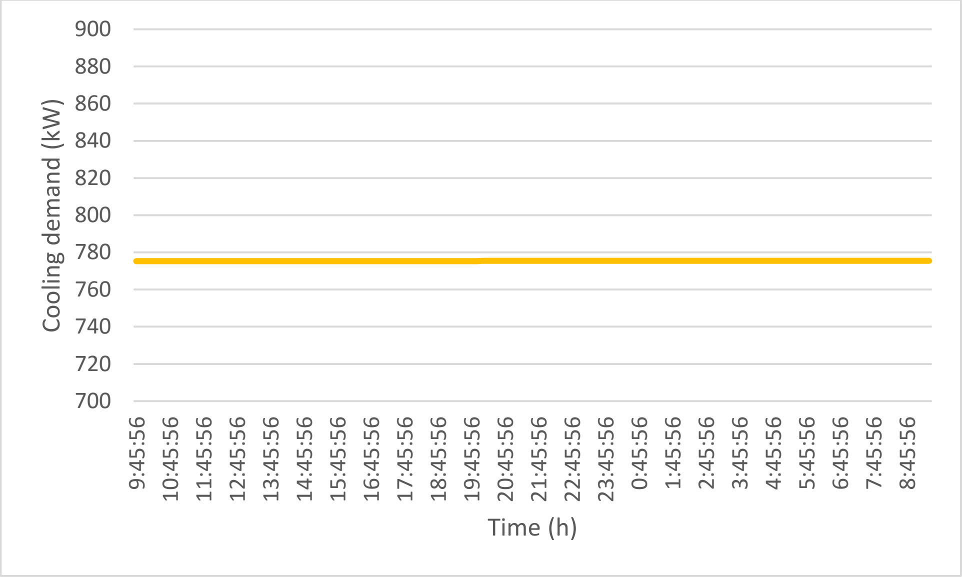

Figure 40. Cooling demand in Gløshaugen (28/04/2016)

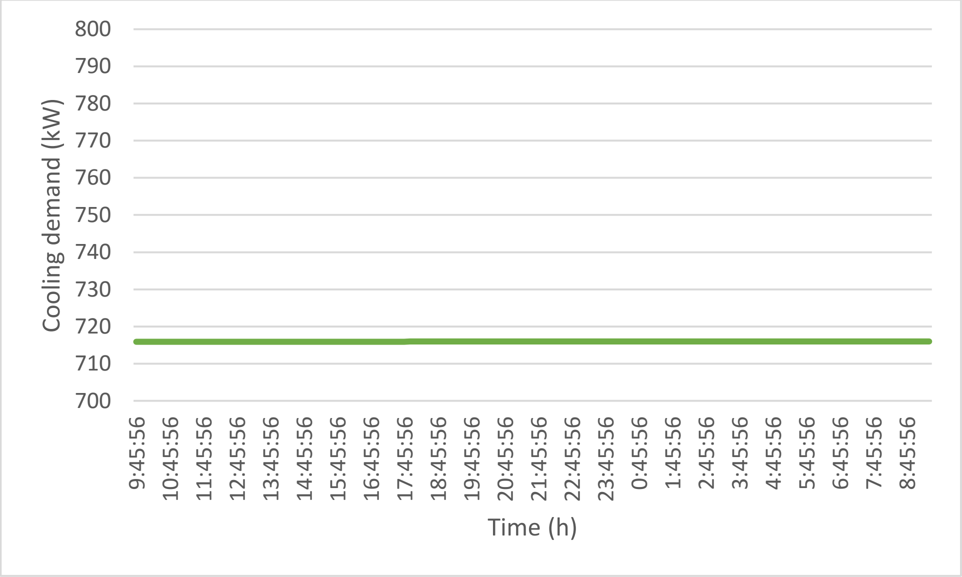

Figure 41. Cooling demand in Gløshaugen (02/05/2016)

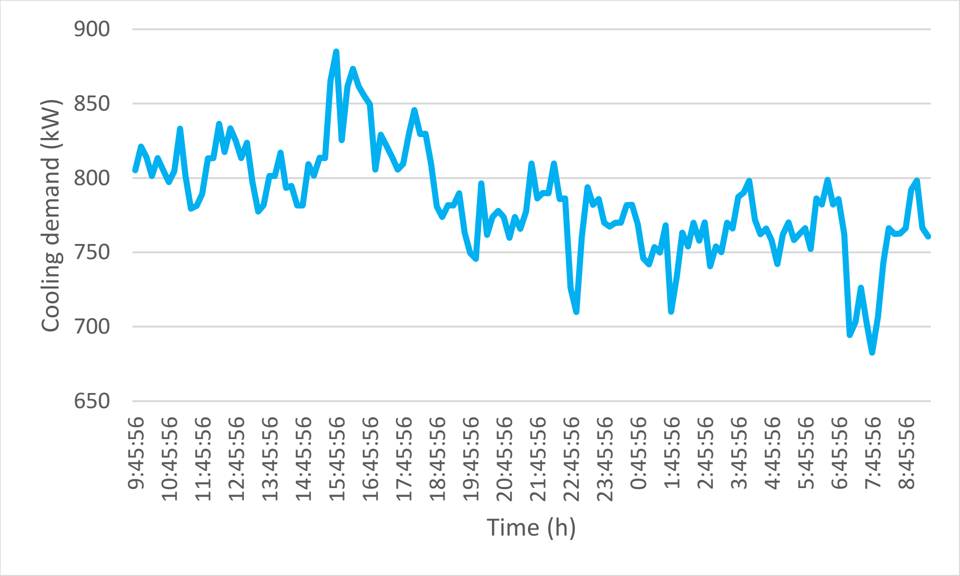

Figure 42. Cooling demand in Gløshaugen (03/05/2016)

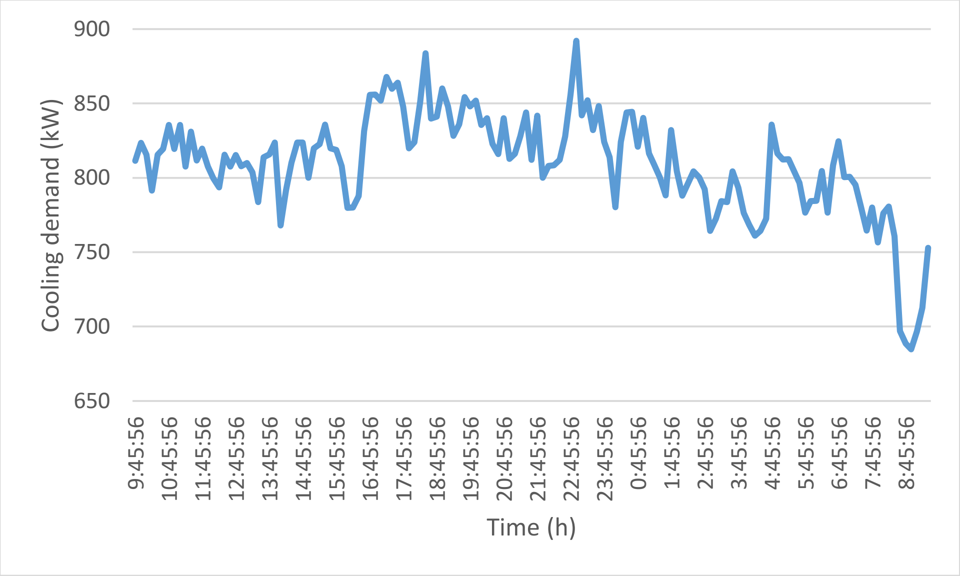

Figure 43. Cooling demand in Gløshaugen (05/05/2016)

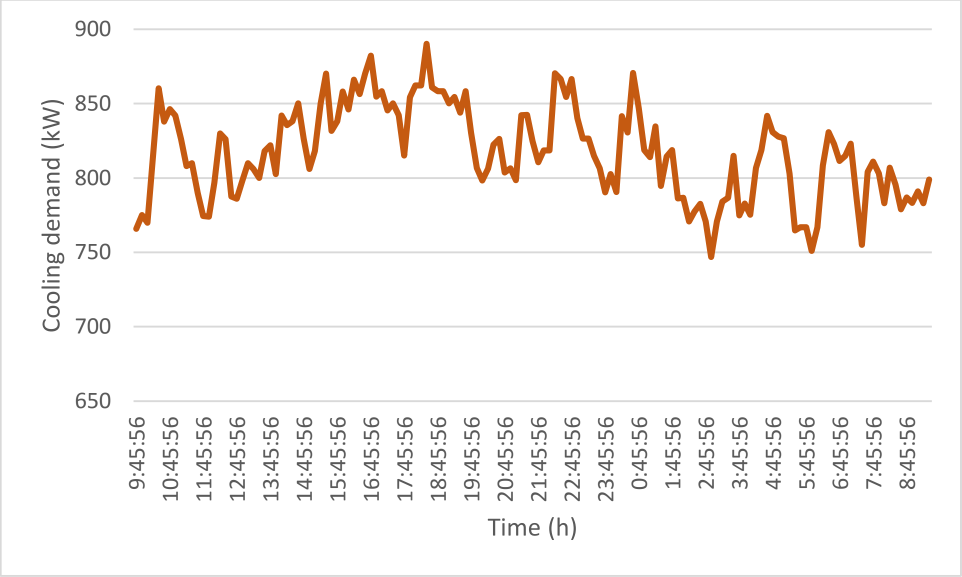

Figure 44. Cooling demand in Gløshaugen (07/05/2016)

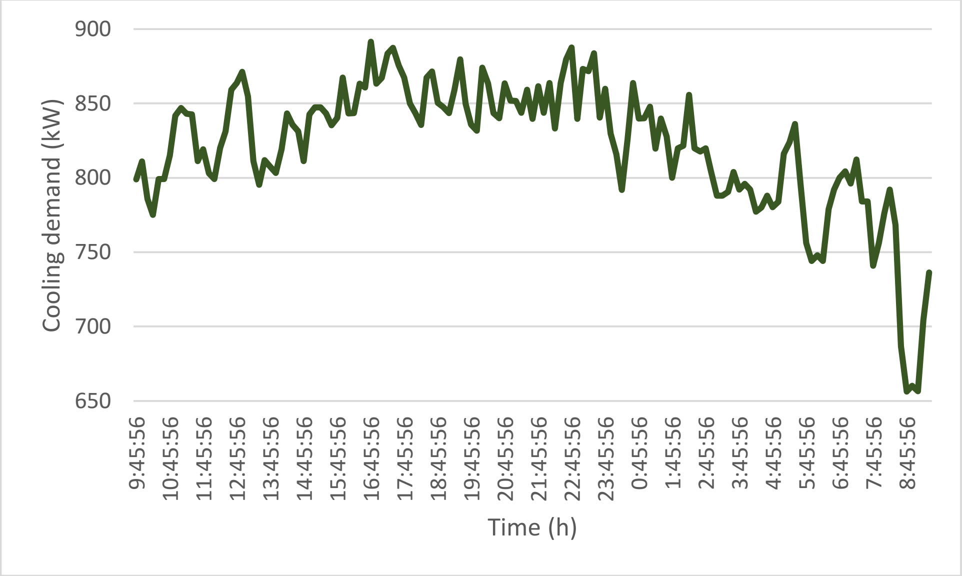

Figure 45. Cooling demand in Gløshaugen (08/05/2016)

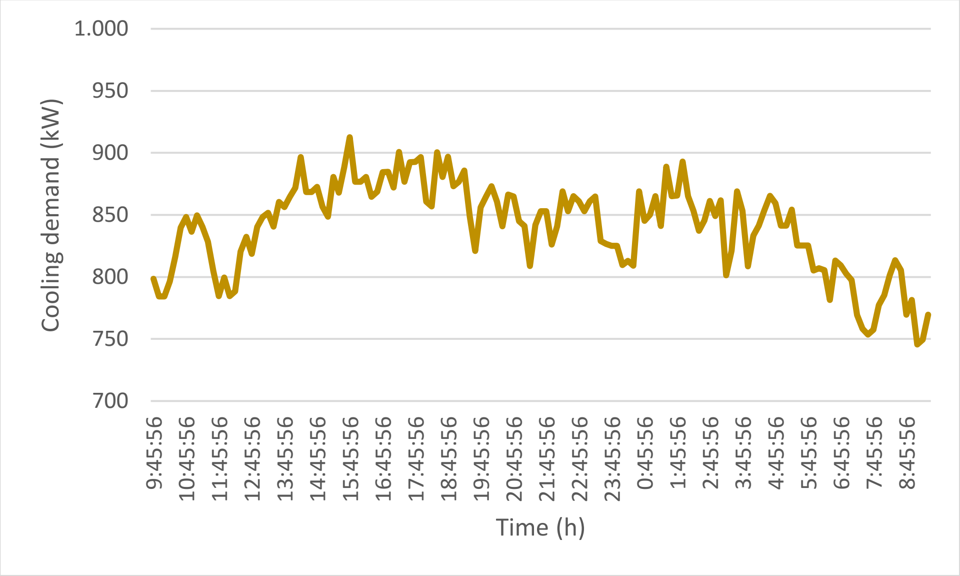

Figure 46. Cooling demand in Gløshaugen (09/05/2016)

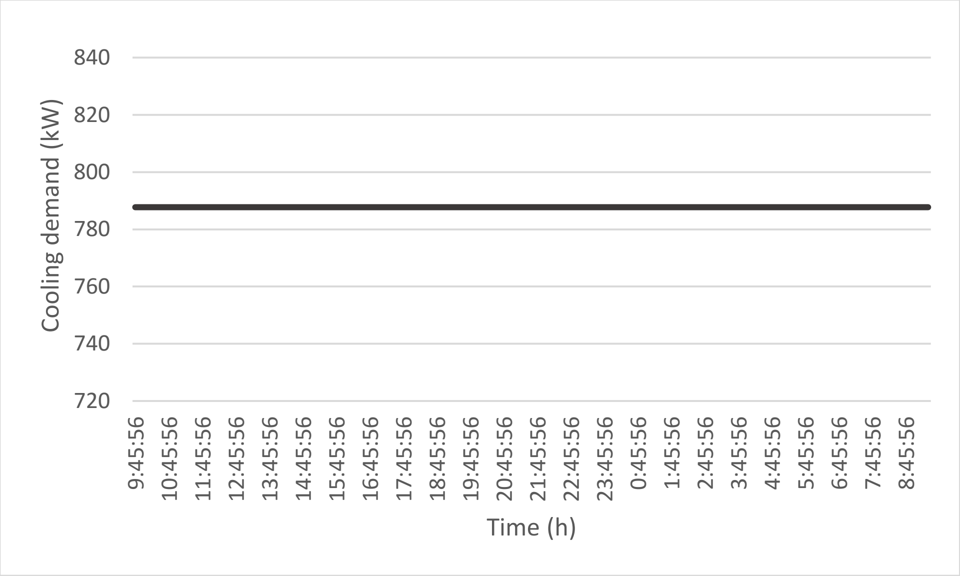

Figure 47. Cooling demand in Gløshaugen (15/05/2016)

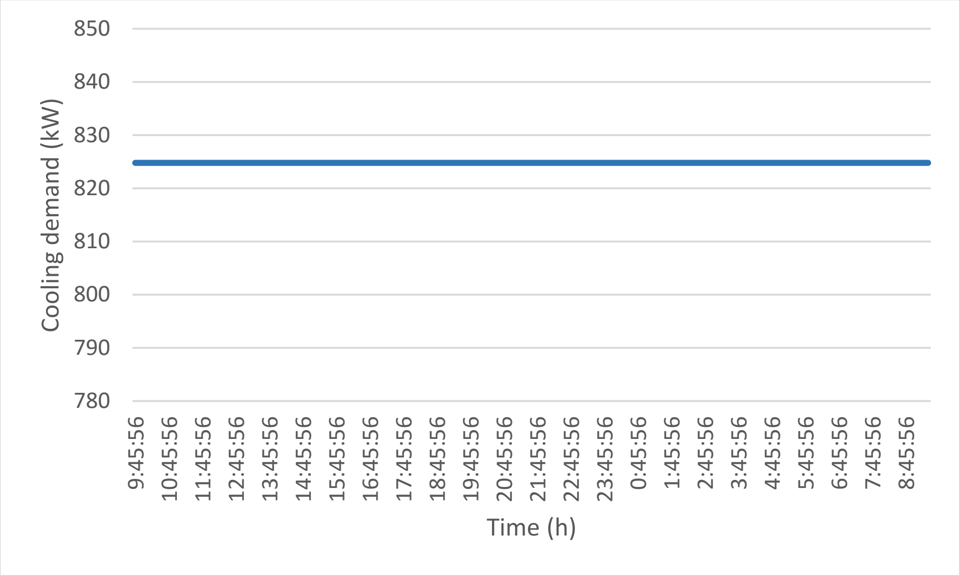

Figure 48. Cooling demand in Gløshaugen (16/05/2016)

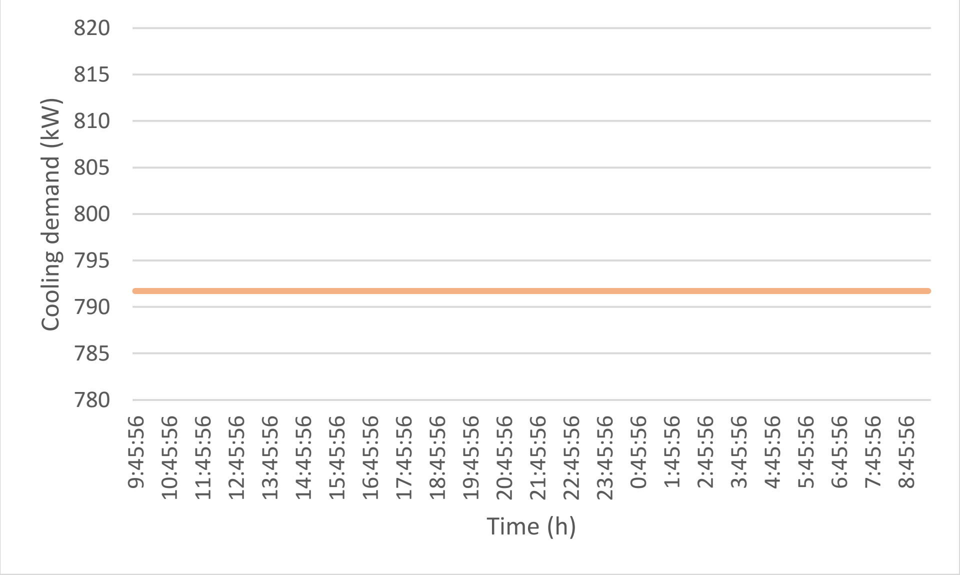

Figure 49. Cooling demand in Gløshaugen (01/06/2016)

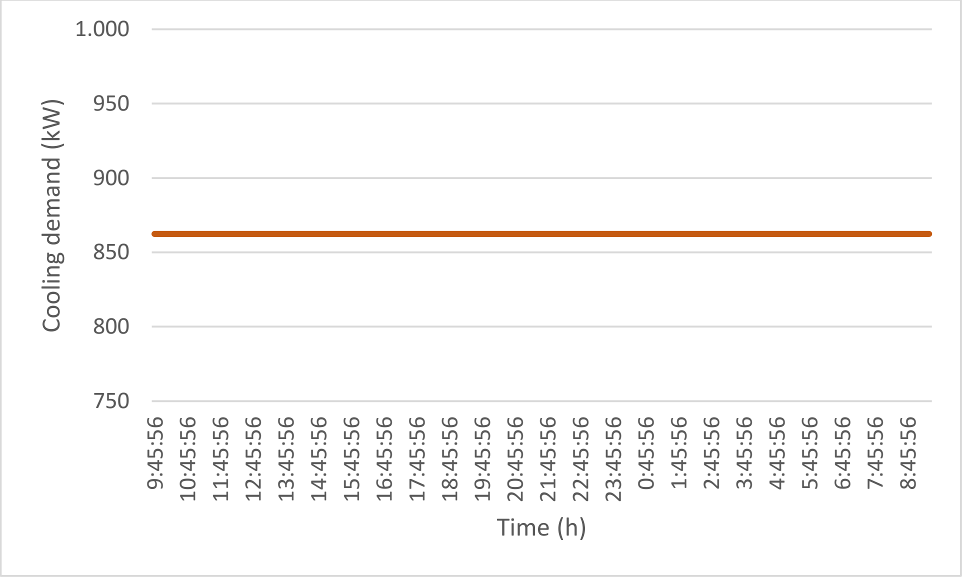

Figure 50. Cooling demand in Gløshaugen (07/06/2016)

Some variations can be found from approximately 650 to 950 kW. Based on these graphs, the cooling demand graph was designed.

ANNEX II

In this annex, the complete results for each chiller/heat pump are displayed.

- One-stage ammonia chiller/heat pump (example from NTNU) [27]:

Qcooling (kW)	*Qheating (kW)*	*W (kW)*	*Part load heating(%)*	*Tsupply (ºC)*	*COPh*	*COPc*	*Relative enery saving(%)*
800	1314	514	100	80	2,56	3,56	60,88
548	919,8	371,8	70	80	2,47	3,47	59,58
382	657	275	50	80	2,39	3,39	58,14
800	1102	302	100	60	3,65	4,65	72,60
548	771,4	223,4	70	60	3,45	4,45	71,04
388	551	163	50	60	3,38	4,38	70,42
800	965	165	100	40	5,85	6,85	82,90
553	675,5	122,5	70	40	5,51	6,51	81,87
393	482,5	89,5	50	40	5,39	6,39	81,45

Table 11. Results from the one-stage ammonia chiller/heat pump in NTNU (Trondheim)

- Two-stage ammonia chiller/heat pump [25]:

Qcooling (MW)	*Qheating (MW)*	*Part load (%)*	*Tsupply (ºC)*	*COPh*	*COPc*	*Relative enery saving (%)*
860	1263	100	80	2,91	3,91	65,64
799	-	100	60	3,66	4,66	72,68
812	974	100	40	5,3	4,3	81,13
812	974	100	40	3,91	4,91	74,42
575	693	70	40	4,66	5,66	78,54
426	505	50	40	5,15	6,15	80,58
265	316	30	40	5,51	6,51	81,85

Table 12. Results from the two-stage ammonia chiller/heat pump from Hanna Risnes

- Hybrid cascade two-stage ammonia-water chiller/heat pump [25]:

Part load (%)	*Tsupply (ºC)*	*COPh*	*COPc*	*Relative enery saving (%)*
100	65/80	2,67	3,67	62,55
70	65/80	2,58	3,58	61,24
50	65/80	2,67	3,67	62,55
30	65/80	2,58	3,58	61,24
100	45/60	3,28	4,28	69,51
70	45/60	3,17	4,17	68,45
50	45/60	3,28	4,28	69,51
30	45/60	3,17	4,17	68,45
100	34/40	5,59	6,59	82,11
70	34/40	5,4	6,4	81,48
50	34/40	5,59	6,59	82,11
30	34/40	5,35	6,35	81,31

Table 13. Results for the hybrid cascade two-stage chiller/heat pump from Hanna Risnes