Sofía Garaizar González de Echavarri

Evaluation of the Energy System in an Energy Efficient Commercial Building

Master's thesis in Mechanical Engineering
Supervisor: Trygve Magne Eikevik
Co-supervisor: Frode Børresen
June 2021

Master's thesis

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Preface and Acknowledgements

This thesis is the final project of my Master of Science degree in Industrial Engineering, with specialization in Thermal Energy, from the University of the Basque Country (UPV/EHU). It was written during my final year abroad at Norwegian University of Science and Technology (NTNU) as part of an Erasmus scholarship in Mechanical Engineering, at the Department of Energy and Process Engineering. Supervised by Trygve Magne Eikevik and in collaboration with GK Norge AS, this thesis has a total of 30 ECTS.

I would like to thank Trygve for his guidance, and Frode Børresen from GK, for his support and help through these months. To Jan Ingar Wollebek and Kaj Nordtvedt from GK, for providing me with important information and suggestions about the building. I would also like to thank my family and friends, for supporting me along the journey of becoming an engineer, especially my to parents, Carmen and Julián, who have always been a source of inspiration.

Abstract

This thesis focuses on the performance of energy systems in passive house standard office buildings in Norway. It is widely known that these buildings are built to minimize the energy use and energy loss, while maximising the use of heat produced by internal loads. While heating loads have decreased significantly, cooling loads have seen an increase. When designing an energy system for these types of buildings, cooling loads are therefore usually the limiting factor.

This master's thesis evaluates the performance of the energy system of an office building located in Oslo, Norway. The aim of the report is to analyse the system based on data gathered from 2019, to find possible failures and propose improvements. To do this, the latest technology was studied, and proposed applied to the building. In addition, a different size of the current system was investigated. It was discovered that due to an oversized system, part load operation decreased the heat pump systems COP by 15 % and 23 %, depending on whether the calculations are based on maximum or average loads. The same problem was found in the cooling machine, where the COP was decreased by 13 %. It was found that the part load operation alone supposed a loss of 5 278 NOK (maximum load) and 10 497 NOK (average load) each year. Three new options were proposed, the last of which improved the COP by 21,1 % for average loads and 18,59 % for maximum loads. These would save the company 6 512 NOK and 5 278 NOK every year respectively. However, this proposal is linked to the use of a peak load system and the use of thermal energy storage to lower the cooling demands. Finally, a full energy system was proposed based on recent technology advances and the analysis of the system.

Sammendrag

Denne oppgaven fokuserer på ytelsen til energisystemer i passivhus-kontorbygg i Norge. Det er kjent at disse bygningene konstrueres for å minimere bruk og tap av energi, samtidig som gjenbruk av varme fra bygget maksimeres. Mens oppvarmingsbehovet har blitt betydelig redusert i slike bygg, har kjølingsbehovet økt drastisk. Når man designer et energisystem for denne typen bygninger, er det derfor kjølingsbehovet som er den begrensende faktoren.

Denne masteroppgaven evaluerer ytelsen til energisystemet i et kontorbygg lokalisert i Oslo, Norge. Målet for rapporten er å analysere systemet basert på innsamlet data fra 2019, for å finne mulige feil og foreslå forbedringer. For å gjøre dette ble det sett på muligheten for å implementere ny teknologi, men også for å omdimensjonere det eksisterende systemet. Det ble oppdaget at systemet er grovt overdimensjonert, som gjør at vamepumpen opererer langt under optimale forhold. Dette gjør at COPen minker med 15 % til 23 %, avhengig av om beregningene er basert på høyeste belastning eller gjennomsnittlig belastning. Det samme problemet ble funnet i kjølemaskinen, hvor COPen blir redusert med 13 %. Det ble også oppdaget at det å operere systemet ved lavere effekt betydde et tap på 5 278 NOK (ved maksima belastning) og 10 497 (ved gjennomsnittlig belastning) hvert år. Tre nye muligheter ble foreslått for å forbedre systemet. Den siste forbedret COPen med 21,1 % for gjennomsnittlig belastning og 18,59 % for maksimal belastning. Dette vil spare selskapet for henholdsvis 6 512 NOK og 5 278 NOK hvert år. Dette forslaget er basert på et spisslastsystem i kombinasjon med termisk energilagring for å redusere kjølingsbehovet. Til slutt ble et nytt, fullverdig energisystem for kontorbygget foreslått basert på ny teknologi som har kommet etter bygget ble bygget, og med bakgrunn i den gjennomførte analysen.

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Acronyms

ASHP	*Air-Source Heat Pump*
BEMS	*Building Energy Management System*
COP	*Coefficient of Performance*
CDH	*Cooling Degree Hours*
CM	*Cooling Mode*
CM	*Cooling Machine*
CTES	*Cooling Thermal Energy Storage*
DR	*Data Rooms*
DX-SAHP	*Direct-Expansion Solar Assisted Heat Pump*
DHW	*Domestic Hot Water*
EAHE	*Earth-to-Air Heat Exchangers*
EB	*Electric Boiler*
GWP	*Global Warming Potential*
GSHP	*Ground-Source Heat Pump*
HP	*Heat Pump*
BRA	*Heated Floor Area*
HDH	*Heating Degree Hours*
HM	*Heating Mode*
HVAC	*Heating, Ventilation and Air Conditioning*

HC	*Hydrocarbons*
HFC	*Hydro-Fluro-Carbons*
ITES	*Ice Thermal Energy Storage*
MPC	*Model Predictive Control*
ODP	*Ozone Depletion Potential*
PLF	*Part Load Factor*
PLR	*Percentage of full load*
aPPD	*Percentage of Dissatisfied People during unoccupied hours*
PCM	*Phase Change Material*
PER	*Primary Energy Ratio*
PLC	*Programmable Logic Controller*
RH	*Relative Humidity*
SPF	*Seasonal Performance Factor*
SCOP	*Seasonal Coefficient of Performance*
SPF	*Specific Fan Power*
PVT	*Thermal Photo Voltaic*
VSD	*Variable Speed Drive*
WSHP	*Water-Source Heat Pump*

Chapter 1

Introduction and Motivation

The continuous population growth and its consequential rise in resource demand, has led to an inevitable ascent in CO_2 emissions. Nonetheless, a global effort is being made to avoid an increase of more than 1,5 ºC on earth's surface (compared to preindustrial levels) [1].

In Europe, the European Union has set a number of targets to eventually become in 2050 the first climate neutral continent in the world [2]. The "2020 climate and energy package" is a set of laws that opens the path to the 2050 goal, by promising a cut in 20 % in greenhouse gas emissions (from 1990 levels), a 20 % energy generation from renewable sources and a 20 % improvement in energy efficiency [3]. In a medium-term strategy, the European Union pretends to reduce greenhouse emission gases by 55 % compared to 1990 levels [4].

The building and construction sector is responsible for almost 40 % of energy- and process-related emissions globally [5]. If countries want to achieve the goals determined in the Paris Agreement, taking action in this sector is one of the most cost-effective measurements. However, reports indicate that the rate at which emissions from buildings are being lowered is not fast enough to reach those goals [6].

In 2018, buildings made up 30 % of the final energy consumption globally (8 % non-residential buildings and 22 % residential buildings) [7]. Also, according to global emissions, residential buildings accounted for 17 % and non-residential for 11 % (direct and indirect emissions) [8]. From 1990 there has been an increase of 10 % in CO_2 emitted by the residential sector according to [9]. One could also take into account the percentage of energy consumption and emissions that correspond to the construction and manufacturing of materials (6 % and 11 % respectively) [7] [8].

In the electricity production in 2020, only 38,2 % of the total produced came from renewable sources in the European Union [10]. Electricity consumption in the world in residential buildings has had a decreasing tendency over the last years, but is still higher than the one in the 90's according to [11] as seen in Figure 1.1. Commercial buildings are taken into account inside the "Services" sector, so the electricity consumption of building in general is higher than the showed numbers.

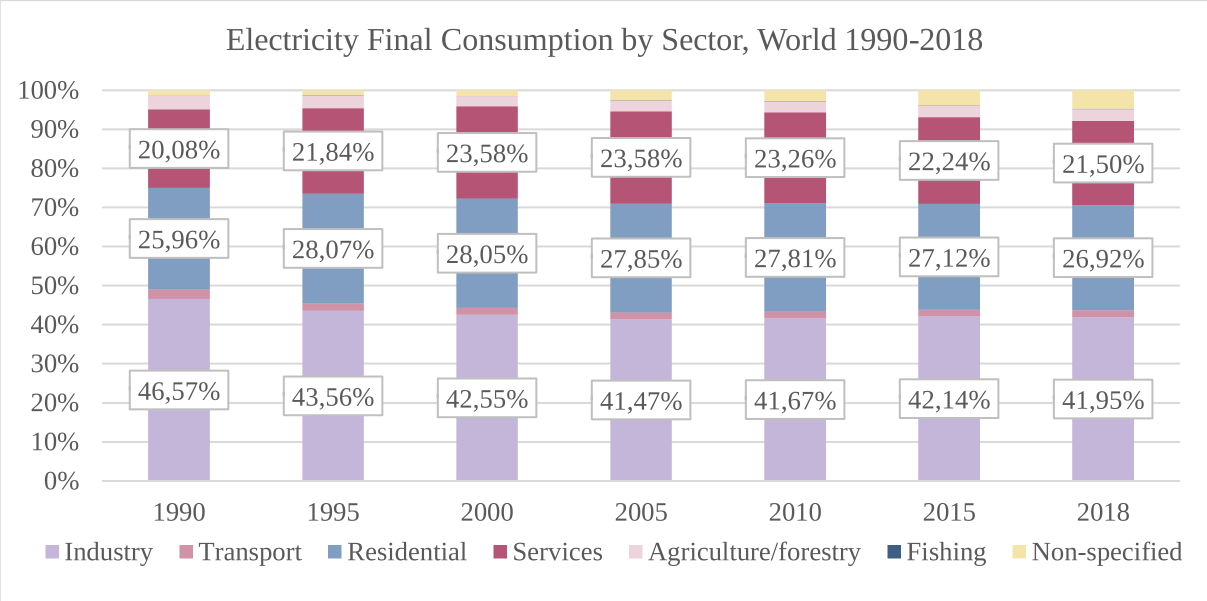


Figure 1.1 Electricity final consumption by sector in the world, from 1990 to 2018 [11].

1.1.1 Building Related Emissions and Energy Consumption: Situation in Norway

Due to the magnificent effort carried out by Norwegian authorities and the Norwegian society, emissions from this country have been considerably lower than its European neighbours. In 2018, according to [12], Norwegian households used 3 465 ktoe (kilo-tonne of oil equivalent) for electricity, while Ireland, a country with similar population, used 703 ktoe. However, when comparing CO_2 emissions, Norwegian households emitted 1 Mt CO_2 while Ireland's emitted 6 Mt CO_2. This comes as no surprise, seeing as the majority of Norwegian households use electric driven convection heaters or alternative sources of heat which do not use combustion [13].

In Norway, at the beginning of 2021, there were 1 681 hydropower plants, with a total capacity of 33 055 MW [14]. This results in 88 % of the electricity production coming from hydropower in a normal year. Other renewable sources contributing to the electricity production are wind (3 977 MW of installed capacity in 2021), solar (160 MW in 2021) and thermal energy (700 MW in 2020) [15] [14]. In 2020, the Nordic country produced the record quantity of 154,2

TWh of electricity, 10 TWh more than the average of the last 5 years [16]. Comparing again these two countries, in 2019 the Nordic country produced 125,8 TWh from hydropower, being wind power the second source with 5,5 TWh [12]. Meanwhile, Ireland's electricity production was dominated by natural gas (16 TWh) and wind energy (9,4 TWh), with a total production of 30 TWh [12].

In 2017, a 70 – 80 % of the energy used to heat Norwegian buildings was electricity, being oil or gas the rest [13]. However, from 2020 the use of oil and paraffins is not allowed to heat buildings in Norway. Both old and new, residential and commercial buildings are covered by this ban [17]. For this reason, the use of alternative heating systems has been very supported by Norwegian authorities.

1.2 Objective

The objective of this master's thesis is to analyse the energy central of a large office building located in Oslo, Norway. Built in 2012, Miljøhuset was the first office building built with the passive house standard in Norway. Therefore, the methods used to dimension the energy system were not enough proven or used. As a result, there is a suspicion that the energy system is not working as it should or as it was expected. This master's thesis will answer to the following questions:

- How does the energy system work?
- Does the system have the right size?
- Could it have been built differently?
- If the system were designed and built today, what could be done differently?

To do so, the energy system will be thoroughly studied and data from 2019 from the different sensors and meters will be analysed. First, relevant theoretical information will be exposed. Then, the building will be presented, providing all information needed for the analysis. After showing the methods used, an exhaustive study of the available data, as well as a discussion, will be presented. Possible improvements based on data and current technology will be presented at this point. Finally, conclusions will be drawn and further work will be proposed.

It must be noted that on a first agreement a simulation model of the energy system was to be developed. This model would be able to reproduce the behaviour of the system (especially the heat pump cycle), to later propose some improvements based on changes made in the simulation model (for example, changing inlet/outlet temperatures, mass flows etc.). However, when starting to gather information to develop the model, it was found that there was not enough information available to the student, such as energy demands of the building, mass flows and pressure data. An additional SIMIEN simulation was then proposed in order to obtain those demands needed for the model, but again, not enough information about the building characteristics was found. For this reason, a different way of finding an answer to the proposed questions was sought.

Before starting to analyse the building in depth, an analysis of the heat pump was proposed. Nonetheless, it was learnt that the heat pump was bought as a whole and that no interactions or changes in the current units could be performed. Consequently, no improvements of the current heat pump cycle will be considered. Nevertheless, improvements based on current technology will be explored, and different heat pump cycles or improvements will be described in Appendix A (even though they will not be able to be established in the current cycle).

The structure of this thesis is as follows:

Chapter 1: Introduction and Motivation. In this chapter, a brief introduction will be presented, as well as the objective and structure of the thesis. A brief study of the state-of-the-art for energy systems in office buildings will be performed, to better understand ongoing research in this field, and see how this thesis can be placed in today's situation.

Chapter 2: Theory. Here, several theoretical concepts will be explained in order to better understand the following chapters. It will also serve as a basis on which the discussion will be built.

Chapter 3: Case Study. A case study will be introduced, where an office building in Oslo (Miljøhuset GK) will be thoroughly studied. Information about the energy system, heat distribution and control strategies will be given.

Chapter 4: Methodology. An analysis of gathered data from the building will be performed, using data from 2019. Both sensor data and energy meter data will be used to later draw conclusions. An exergy analysis will be proposed as an additional tool to better understand the performance of the system.

Chapter 5: Results. Results of the analysis will be presented using graphs and tables, to be able to discuss the later. The results of some useful calculations will also be given.

Chapter 6: Discussion. Conclusions on the size of the energy system will be drawn, as well as proposals for an alternative sizing. Improvements of the system based on the analysis and on technology available will also be pinpointed. A final energy system will be proposed.

Chapter 7: Conclusions. Conclusions will be drawn based on the results presented and the discussed points.

Chapter 8: Further Work. Further research on the topic will be suggested, since the time dedicated to the thesis is limited and more work could be done in the future.

1.3 Literature Review

Heat pumps have been used as a heating and cooling technology since 1855 when Peter von Rittinger developed and built the first heat pump system in Austria [18]. It was in 1876 in Germany when Carl von Linde used ammonia as a refrigerant for the first time. Ever since, the technology has been developed to the point where it is considered one of the keys to a sustainable future in buildings.

Today, heat pumps are not only used for heating purposes but also cooling of spaces, refrigeration and chilling of products in different buildings and facilities.

Even though heat pumps seem to be a perfect tool to reduce emissions from buildings, they covered less than 5 % of heating demands in a global scale in 2019. Meanwhile, fossil fuel-based and electric heaters made up almost 80 % of new sales in 2020 [19]. Nonetheless, it is predicted that by 2030 heat pump sales will increase from 5 % to 22 % [20].

Back in 2010, SINTEF [21] proposed some guidelines to follow in favour of energy efficiency in office buildings such as reducing heat losses, reducing the cooling demand, reducing the electricity consumption, and selecting an energy source with low CO_2 emissions. Switching from traditional heating systems to more sustainable ones such as heat pumps in office buildings is one of the steps to follow. According to [22], in 2013 only 4 % of office buildings in Norway fulfilled the passive house standard.

In highly insulated office buildings, one of the major challenges is lowering the cooling demands. Since the building is designed to make the most of solar gains and reduce the heat losses, keeping an adequate temperature inside the building during summer months can be difficult [22]. Night cooling through the ventilation system, is according to Alonso et al. [22] a good strategy to lower the cooling demands in highly insulated buildings. However, Nord et al. [23] determined by simulating both scenarios (an air-to-water heat pump with and without night cooling), that it was preferable to not use night setback in winter if the overuse of the electrical boiler wanted to be avoided.

A proper dimensioning of the energy system is crucial to a well-functioning building. Analysis of heating and cooling loads, internal loads, hot water and climate must be carried out in order to design a system which is able to provide enough energy to the system, but without wasting money, space and energy. High variable loads carry a poor operation and therefore an over dimensioned system is not desired [24]. To avoid the efficiency losses attached to on/off cycling of heat pumps [25], especially in over-dimensioned systems, variable speed compressors (VSD) which adapt to the load are a good alternative [26].

Non-traditional technologies have been studied to offer an alternative to standard systems and to seek higher efficiencies, better performances and higher energy savings. The use of dual source heat pumps has started to become popular, especially the ones combining the use of solar resources and traditional heat pumps, as pointed out by Bertram et al. [27] and Alonso et al. [21]. The first to propose the use of this technology were Sporn et al. back in 1955 [28]. Other technologies used recently are thermal energy storage systems, which store energy and allow a constant and steady operation of the system even if the source is intermittent (such as solar thermal energy). Not only heat can be stored, but also cooling energy, which is very useful in highly insulated buildings where cooling demands are usually high [29] [30] [31]. The use of PVT panels has also become very popular, since studies show that it increases the electricity production (0,2 – 0,5 % gets lost for every 1 ºC increase [32]) by cooling the solar panels while recovering thermal energy that can be used later in the building [32].

When designing an energy system for a building, control strategies are fundamental to exploit the full potential of the system. Advanced control technologies for heat pump systems were studied by [33], [34] and [35], concluding that an optimal control system improves the operation of the system, sometimes by shifting loads or by reducing energy consumption (further studied in Section 2.5.3).

Chapter 2

Theory

In this chapter, important background theory regarding heat pumps in office buildings will be studied. Equations and parameters used for evaluation of energy systems are introduced, as well as different factors to take into account when performing an analysis to energy systems. This will help set a base on which later establish the discussion. Finally, other office buildings in Norway will be included and current technology used in these kinds of building will be pin-pointed.

2.1 Heat Pumps in Integrated Systems

2.1.1 Principles of Heat Pumps

Heat pumps use the vapor compression cycle or reverse Carnot cycle (see Figure 2.2) to extract heat from a heat source and deliver it to a heat sink. It consists of four basic components: a compressor, a condenser, an expansion valve and an evaporator. A traditional heat pump system is represented in Figure 2.1. The refrigerant enters the evaporator as a mixture of vapor and liquid, and evaporates as it absorbs the heat provided by the heat source. Then, as a saturated vapor, it is compressed in the compressor increasing its temperature along the way. In the condenser, it rejects heat to the cooling medium until it turns into saturated liquid. Finally, it expands in the expansion valve and flows back to the evaporator. One of the characteristics of these kinds of cycles is that the efficiency drops when the difference between evaporation and condensation temperatures increases. Furthermore, the loss of expansion work in throttling devices further contributes to the loss of efficiency.

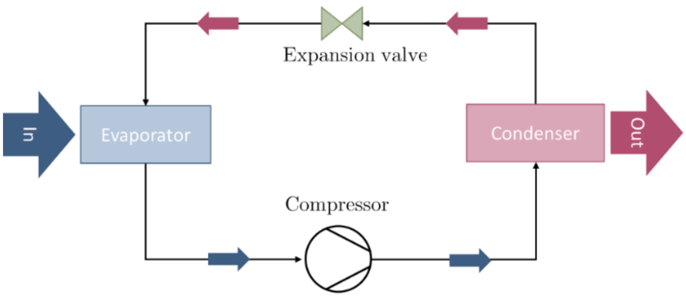


Figure 2.1 Heat Pump's basic principles.

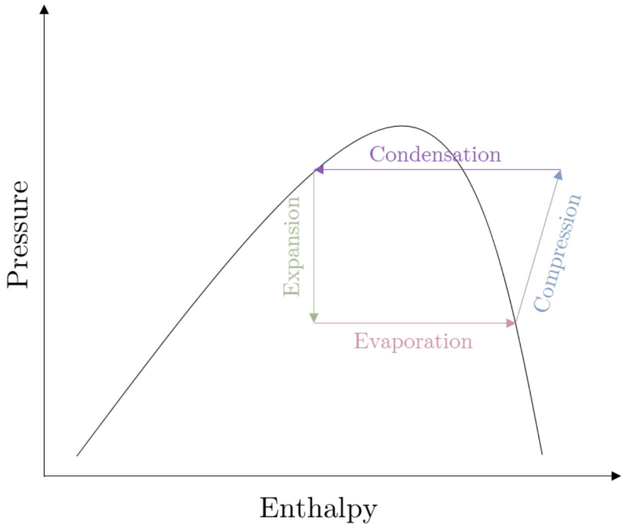


Figure 2.2 Vapor compression cycle in heat pumps, basic representation of a cycle.

In the heating and cooling sector, one refers to an integrated system when talking about a single system, simple or complex, which includes several different energy systems, such as cooling, heating, ventilation, and Domestic Hot Water (DHW). Usually, it is assigned as HVAC (heating, cooling and air conditioning) when the system provides heating and cooling and is also in charge of air conditioning indoors. In the case of heat pumps, it is especially adequate to integrate these kinds of systems, for the nature of the technology lies on the transport of heat from one space to the other. By using integrated systems, both the implicated company and the environment get the best of the bargain. Less equipment and less materials lead to fewer emissions and lower costs (installation, operation and maintenance). Also, heat pumps can be reversed, if prepared to do so, and work as a cooling system during warmer months.

2.1.2 Heat Pump Types

Air-Source Heat Pumps (ASHP)

Air-Source Heat Pumps (ASHP) use the available outdoor air as a heat source in order to transfer heat from the outside to the inside (when used in heating mode). This type of heat pumps is the most used and sold in the market, since they are easily installed and work for most climates. However, there are also drawbacks to this kind of systems, such as variations in the system performance, freezing problems in the evaporator, and low efficiency at low outdoor temperatures [36]. To circulate air through the evaporator, the use of fans is required. These devices demand power and can be noisy, so their location must be strategic.

In cold regions, low COP (Coefficient of Performance, explained in Section 2.2.1) at low outdoor temperatures (< -5 ºC [37]) is a major concern for classic ASHP systems. At low outdoor temperatures, the evaporation temperature drops and results in an increase of mass flow rate, leading to an increase of compression power. As mentioned earlier, a decrease in the evaporation temperature will increase the temperature difference with the condenser, thus reducing the COP.

Frost formation in the evaporator in humid conditions is also a big issue for ASHP. When the surface temperature of the evaporator is below freezing temperature and below air dew temperature, water will start to condense and freeze at the surface. Frost build-up occurs when outside temperature is lower than 6 ºC and relative humidity (RH) is higher than 60 % [38]. Frost will build up, hinder heat transmission, and eventually lead to a sizeable failure of the system. To avoid this, defrosting procedures must take place regularly, decreasing the system's COP. One way of getting rid of the frost is by running the heat pump in reverse mode (if the heat pump is reversible). By doing this, the evaporator will now act as a condenser and its temperature will increase, melting the built-up frost.

Water Source Heat Pumps (WSHP)

Water source heat pumps, use shallow water available in Earth's surface at wells, sea or waste water to extract heat and deliver it to the refrigerant. Due to the Gulf Stream, the Norwegian coastline has relatively high temperatures which is very convenient for the heat pump application. Salt water has a lower freezing point than fresh water, which is a benefit, but it is more corrosive.

When using sea water as a heat source, some problems have to be faced. Firstly, biological activity and shells may cause clogging of the pipes, thus both coarse and fine filters should be installed. Secondly, leakages may occur causing environmental damages, so a leak proof design must be carried out. Lastly, sea water is highly corrosive and may damage pipes and other expensive equipment. A corrosion-proof design, for example using titanium heat exchangers, is a must when working with this fluid [98].

Ground water can also be a source of heat. Water temperature from wells or aquifers is stable and moderate during the year, with a typical value of 2 – 8 ºC, which is a positive aspect regarding the performance of the system [98].

Ground-Source Heat Pumps (GSHP)

Ground-Source Heat Pumps (GSHP) allow heat extraction from the earth. Ground source heat exchangers are buried in the ground to absorb high temperature heat, resulting in high COPs. Because the soil acts as a thermal storage, after the heating season reheating of the ground may be needed in order to use it as a heat source the next heating season (this is done by storing heat in the ground during cooling season) [39]. This system is also widely used with cooling cycles, due to the enormous thermal capacity and volume of earth, working as a heat sink. These properties also keep temperatures roughly constant throughout the season.

A summary of the different heat pump types is presented in Table 2.1.

Table 2.1 Summary of benefits/drawbacks of the main heat pump types.

Source	**Source temperature**	**Benefits**	**Drawbacks**
Air	Depends on outdoor temperature.	Cheap. Available everywhere.	Frost formation in evaporator, defreezing process. Fluctuating capacity and COP due to unstable outdoor temperatures. Need of auxiliary heating system during coldest periods. Use of noisy fans.
Water	4 – 10 ºC [39].	More stable temperatures over the course of a year than air.	Large initial investment. Problems with salt water.

			Not available in all locations. Leakages.
Ground	T= T_{out} + 2 ºC 50 m ↓ 1 ºC ↑ 0 – 10 ºC [39].	More stable temperatures. Higher temperatures. Long lifetime. Minimum maintenance.	High initial cost. Not all soils allow drilling. Not available in all locations. Leakages.

2.1.1 Reversible Heat Pumps

Reversible heat pumps are able to work in both directions in order to deliver both heating and cooling. This is done by using a reversible valve or a 4-way valve which allows the refrigerant to flow both directions. In a reversible heat pump, the heating capacity is usually higher than the cooling capacity, due to design factors such as heat exchanger surface and size.

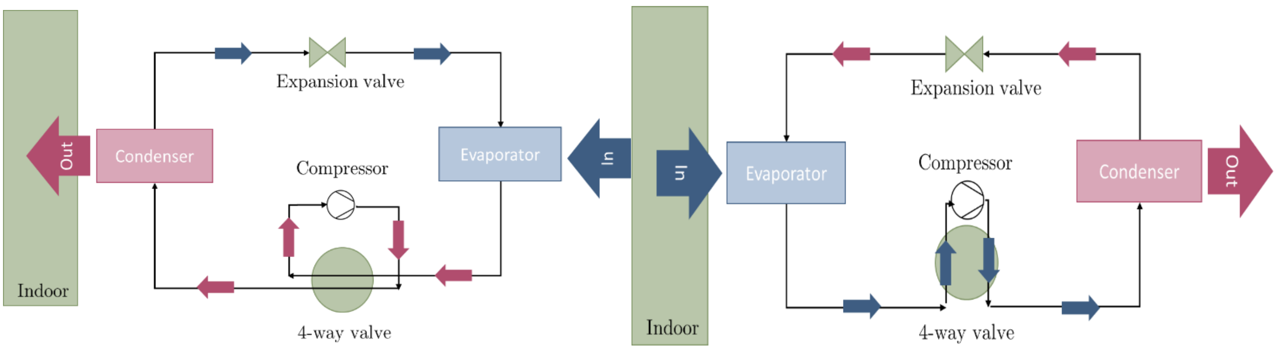


Figure 2.3 Reversible heat pump sketch. Heating mode (left) and cooling mode (right) of a reversible heat pump [40].

2.1.2 Refrigerants Used in Heat Pumps

When designing a heat pump, the selection of refrigerant is of great importance. The refrigerant of choice must first cover all the requirements from the cycle, such as critical temperature and pressure, thermal conductivity and density to name a few. However, awareness against climate change and environmental impact has brought up attention to the negative impact that traditional refrigerants have had and still have on our atmosphere. Although refrigerants should stay within the refrigeration cycle, leakages occur damaging the atmosphere. Moreover, the treatment that those substances receive after their service life leaves much to be desired.

The majority of these traditional refrigerants are typically hydrofluorocarbons (HFC), a type of fluorinated gas (F-gas), which has a global warming potential of 2000 times that of CO_2. HFC emissions rose from almost zero in 1990 to 1 100 Mt of CO_2 eq. in 2010 [41]. Nevertheless F-gas emissions have been falling since 2015 in the EU, due to EU legislation. By 2030, EU bets on reducing F-gas emissions by two thirds compared to 2014 [42]. This legislation also stimulates the use of other more sustainable alternatives.

Optimum properties for refrigerants are to name a few: high critical temperature, high critical pressure, low density, low viscosity and high thermal conductivity. Furthermore, other factors are also important when choosing a refrigerant such as price, accessibility and compatibility with the system.

Historically Used Refrigerants

R134a

This is a refrigerant used typically in high temperature applications. It has a very high efficiency compared to other HFC but lower than R717 (ammonia). At atmospheric pressure, its critical temperature is -26,3 ºC. Because it is a very stable substance, it will not react with other substances before reaching the higher layers of the atmosphere and will damage it. It is to some extent banned in the European Union, and will be completely banned in 2030 [43].

R407c

Originally used as a replacement of R22, it is mainly used in air conditioning systems and medium temperature applications. It was banned from EU the 1st of January of 2020 since its Global Warming Potential (GWP) exceeds 150 (it is 1732) [44].

R410a

Due to its high GWP (675), the U.S Environmental Protection Agency (EPA) has listed R410a has “unacceptable” [45]. According to [46], 1 kg of this substance, has the same greenhouse impact as 2 t of CO_2.

Natural Refrigerants: New Trend in Sustainable Systems

R744 – Carbon Dioxide

The use of CO_2 in heat pump systems is not something new. It was one of the first refrigerants to be used, mainly in air conditioning and refrigeration systems. However, with the development of synthetic refrigerants, CO_2 was left aside. It was not until the prohibition of several of those refrigerants, when interest in the use of CO_2 was regained mainly thanks to the work of Gustav Lorentzen at NTNU. Having an Ozone Depletion Potential (ODP) of 0, and a GWP of just 1, it seems that CO_2 is key to solve the problem between refrigerants and the environment.

After the reintroduction of CO_2, it was mainly used as a mobile air conditioner and heat pump water heater refrigerant [47]. Because of limitations in that time's equipment, low theoretical efficiency and high-pressure levels, the scientific community was not very confident of the future development of CO_2 as a refrigerant for a wide range of applications [47].

One of the challenges of working with CO_2, is that its critical point is 73.8 bar and 31.1 ºC, a high pressure and very low temperature compared to other refrigerants. This leads to large expansion and superheating losses. Above this critical point, CO_2 will never condense. Working in this area is called "trans-critical operation". Instead of using a condenser, a gas-cooler is used, with the same goal: transfer heat to the heating medium. A CO_2 heat pump will reject heat at constant pressure but gliding temperature, and since the COP is determined by the outlet temperature from the gas cooler, high COPs are achieved when hot water at high temperatures is required. This operation mode will occur most typically when the heat pump is located in a hot climate.

In cold climates, where the cycle stays subcritical, CO_2 heat pumps have uncountable benefits. It is cheap, not flammable, sustainable and has a very good performance. It is more suitable for applications where big temperature glides are needed, such as DHW heating (max, outlet water temperature 80-90 ºC [48]). However, if the application does not require such high temperatures, other natural refrigerants are more suitable such as propane or ammonia.

R290 – Propane

Propane has been used as an alternative for HFC refrigerants for some time now. This hydrocarbon (HC) has many advantages compared to HFC, such as zero ODP and lower GWP (3) and in general better efficiency and lower compressor discharge temperatures [49]. For heating and cooling of buildings, propane's characteristics are more than suitable. Nevertheless, a notorious drawback of these kind of systems is safety regarding high flammability. Due to this, HC are only used in heat pumps with a capacity lower than 100-200 kW [50].

In order to deliver safe systems, a reduction of the refrigerant charge is one of the solutions. By having less refrigerant, the flammability risk and toxicity exposure is considerably decreased. This can be achieved by selecting high thermal efficient and low internal volume condensers [51].

R717 – Ammonia

Ammonia has excellent thermodynamic properties that have been tested and used for a long time. Its properties make it a perfect fit for industrial applications, such as heat pumps with capacities above 100 kW [52]. One factor to take into account with these kinds of systems, is the high discharge temperature. To limit this, some actions can be taken such as increasing the temperature glide at the condenser, installing flooded evaporator systems or using accurate control with electronic expansion valves. Other measures can be designing two stage systems or using compressors with integrated cooling.

However, one of the major drawbacks of this substance is the corrosion of copper in contact with water moisture. It is also a highly toxic gas, but due to its strong odour, risk of poisoning is very small.

When designing an Ammonia heat pump, several things have to be taken into account [50]:

- Tubing and components in steel or aluminium (due to corrosion).
- There is a high discharge temperature which requires special measures.
- An insoluble lubricant must be used.
- Special safety measures: gas-tight machinery room, two-stage ventilation systems, NH_3 absorbers and alarm systems activated by leak detectors.

A comparison between the presented refrigerants is shown in Table 2.2.

Table 2.2 Comparison of characteristics of different working fluids. [50] [52].

Refrigerant	**R134a**	**R407c**	**R410a**	**R744**	**R290**	**R717**
ODP	0	0	0	0	0	0
GWP	1300	1732	675	1	3	0
Boiling Point, 1 bar [ºC]	-26,1	-43,8	-51,4	-78,03	-42,1	-33,3
Critical T [ºC]	101,05	86,7	74,7	31,1	96,74	132,3
Critical P [bar]	40,6	46,2	51,7	73,8	42,5	113,5

2.2 Evaluation and Analysis of Energy Systems

2.2.1 Parameters Used for Evaluation

In order to evaluate the performance of different systems, and being able to compare different ones, efficiency analyses are often carried out. In heat pump systems, Coefficient of Performance (COP), Seasonal Performance Factor (SPF) and many other parameters are used to determine the system's efficiency.

Coefficient of Performance (COP)

This coefficient compares the work used by the compressor, which is the main energy source that comes as a cost, with the energy provided to or by the system, depending on the function of the heat pump (heating/cooling). Different typical COP values are shown in Table 2.3.

$$COP_{heating} = \frac{Q_{out}}{W_{in}} \qquad \text{Eq. 2.1}$$

$$COP_{cooling} = \frac{Q_{in}}{W_{in}} \qquad \text{Eq. 2.2}$$

If the heat pump is used in an integrated system and it is used for cooling and heating purposes, it's COP will be notoriously higher:

$$COP_{heating+cooling} = \frac{Q_{out} + Q_{in}}{W_{in}} \qquad \text{Eq. 2.3}$$

In order to compare COP from different systems effectively, the measurements must always be carried out at the same outside temperature.

However, the COP can also be expressed in terms of the temperature difference in the heat pump:

$$COP = \eta_c \frac{t_c}{t_c - t_e} \qquad \text{Eq. 2.4}$$

Where:

η_c: is the Carnot efficiency [-],

t_c: is the condensation temperature [ºC],

t_e: is the evaporation temperature [ºC].

Table 2.3 Coefficient of Performance (COP) for different heat pumps types.

Heat pump type	**COP**
ASHP	2 – 4[1] [36]
WSHP	3,3 – 4,4 heating, 4,1 – 5,8 cooling [53]
GSHP	3 – 5 [54]

Seasonal Coefficient of Performance (SCOP)

This factor is used to see the overall efficiency of the system throughout a season. It is the average COP over the full season, whilst the COP itself is a punctual measurement, fluctuating with outdoor temperatures, heating demands etc.

Seasonal Performance Factor (SPF)

Another factor to take into account when assessing a heat pump's efficiency is SPF. It is defined as the 'net seasonal coefficient of performance in active mode', being the ratio between annual usable energy provided by the heat pump and the energy supplied to the whole heating system [54]. Here, not only compressor power is taken into account, but also fan power, auxiliary heaters for peak loads, pumps and also any other heating source that is used, such as fossil fuelled boilers. Typical values for SPF can be found in Table 2.4.

$$SPF = \frac{Q_{out}}{W_{in} + E_{fan} + E_{aux}} \qquad \text{Eq. 2.5}$$

Table 2.4 Seasonal Performance Factor (SPF) for different heat pump types.

Heat pump type	**SPF**
Air source heat pump	1,8 – 2,2[2] in cold climates [54]
Ground source heat pump	2,6 – 3,6 [54]

SPF has been increasing steadily since 2010, reaching almost 4. In colder climates this value is lower, while in milder climates can reach up to 4.5 and higher [19].

It comes as no surprise that ASHP have a lower SFP, for the use of fans to move the air demands a great deal of energy.

[1] Air source heat pump's COP varies substantially with outside temperature.
[2] A study in Canada [101], determined that the equivalent COP would be 2,3 – 3,5.

Primary Energy Ratio (PER)

Defined as the ratio of useful heat delivered to primary energy input, it is used to compare heat pumps which are driven from different energy sources, such as electricity from renewable sources (hydro, wind, solar) or natural gas motors [39].

$$PER = \eta\, COP \qquad \text{Eq. 2.6}$$

Being η the efficiency at which the primary energy is being transformed to shaft work in the compressor. If the electricity used to power the compressor, comes from a coal power plant, η could be as low as 25 % [39].

Relative Energy Saving (ΔE)

It is the relative energy saved when using a heat pump instead of an alternative source of heat.

$$\Delta E = \left(\frac{1}{\eta_{alt}} - \frac{1}{COP}\right) \cdot 100\% \qquad \text{Eq. 2.7}$$

Where;

η_{alt}: Efficiency of the alternative source [-],

ΔE: Relative energy savings [%],

COP: Coefficient of performance of the heat pump.

In Eq. 2.7 the relative energy savings of a heat pump compared to an alternative source is shown [55]. The representation of Eq. 2.7 can be seen in Figure 2.4.

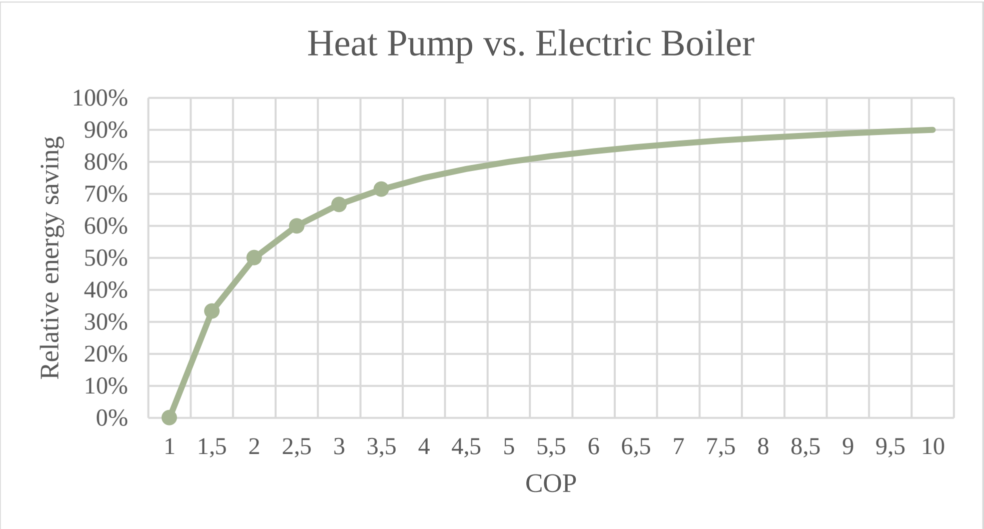


Figure 2.4 Relationship between the relative energy saving and the COP. System with a heat pump and an electric boiler with an efficiency of 1.

According to Figure 2.4, a small variation of the COP in systems with low COP, leads to a large variation in the relative energy saving. Nevertheless, systems with a larger COP are less sensitive to these fluctuations.

2.2.2 Mass, Energy and Exergy Balances

In order to see how the different elements of the system are working in terms of efficiency and to be able to put forward some improvements, an energy and exergy analysis will be performed on the different elements. A steady-flow, steady-state process will be considered. It will also be considered that the specific heat stays constant during the whole process.

Conservation of Energy

To calculate the different properties of each of the states, one will assume that the known property is temperature and that the fluid is water, so the following equations will be used:

$$h_x = C_p * T_x \qquad \text{Eq. 2.8}$$

$$s_x = C_p * Ln\left(\frac{T_x}{T_0}\right) + s_{water} \qquad \text{Eq. 2.9}$$

Where:

h_x is the enthalpy of the studied point *x* [kJ/kg],

s_x is the entropy of the studied point *x* [kJ/kg·K],

C_p : is the specific heat [kJ/kg·K],

T_0 : is the reference temperature [K].

Taking into account the enthalpy at the inlet and outlet of the component, following the conservation of energy principle, the work supplied will be:

$$\dot{m}\, h_{in} + \dot{W}_{in} = \dot{m}\, h_{out} \qquad \text{Eq. 2.10}$$

Where:

$\dot{W}_{in}$ is the work input [kW],

h_{in} , h_{out} are the specific enthalpy of inlet and outlet respectively [kJ/kg].

If one wanted to take into account the irreversibilities, the isentropic efficiency and the isentropic work could be used, defined by:

$$\eta_{is} = \frac{h_1 - h_2}{h_1 - h_{2s}} \quad \text{Eq. 2.11}$$

$$\dot{W}_{is} = \dot{m}\,(h_{2s} - h_1) \quad \text{Eq. 2.12}$$

If there is a heat transfer between the component and the ambient, following the same procedure this heat can be calculated:

$$\dot{m}\,h_{out} + \dot{Q}_{out} = \dot{m}\,h_{in} \quad \text{Eq. 2.13}$$

$$\dot{m}\,h_{in} + \dot{Q}_{in} = \dot{m}\,h_{out} \quad \text{Eq. 2.14}$$

Where:

$\dot{Q}_{out/in}$ is the heat transfer with the ambient [kW].

Global energy conservation principle determines:

$$\dot{W}_{in} + \dot{Q}_{in} = \dot{Q}_{out} \quad \text{Eq. 2.15}$$

Non-Conservation of Entropy

Entropy balance can be written as:

$$\dot{m}\,s_{in} + \dot{\sigma} = \dot{m}\,s_{out} \quad \text{Eq. 2.16}$$

Where:

$\dot{\sigma}$: is the entropy generation rate in the process [kJ/kg·K],

s_{in}, s_{out} is the specific entropy of inlet and outlet respectively [kJ/kg·K].

If there is a heat transfer with the ambient:

$$\dot{m}\,s_{in} + \dot{\sigma} = \dot{m}\,s_{out} + \frac{\dot{Q}_{out}}{T_H} \quad \text{Eq. 2.17}$$

Where:

T_H is the temperature of the hot medium (heat sink) [K].

$$\dot{m}\,s_{out} = \dot{m}\,s_{in} + \frac{\dot{Q}_{in}}{T_C} + \dot{\sigma} \quad \text{Eq. 2.18}$$

Where:

T_C is the temperature of the cold medium (heat source) [K].

Defining Exergy

Exergy is the maximum shaft work or electrical work a system can carry out when it is brought into thermodynamical equilibrium with its environment, this is, work obtained from a reversible process. In engineering, this parameter is used to detect possible improvable points or points where reversible work is lost. Even though energy balances can determine the energy needs of the system, they cannot provide enough information about how efficiently is this energy being used.

In this thesis, only the physical specific exergy will be taken into account, which can be defined as:

$$b_x = (h_x - h_0) - T_0 \cdot (s_x - s_0) \quad \text{Eq. 2.19}$$

Where:

h_0 is the enthalpy of the reference state [kJ/kg],

T_0 is the reference temperature [K],

s_0 is the entropy of the reference state [kJ/kg·K].

Exergy Analysis

When doing an exergy analysis, there are two main goals: determine the exergy destruction and the exergy efficiency in each of the components. By doing so, one will see which components have more room for improvement.

The following criteria will be used to develop the functional analysis:

Exergy input – Exergy Output = Irreversibilities

Or:

Fuel (F) = Product (P) + Losses (L) + Destruction (D) Eq. 2.20

If there is a work contribution:

Table 2.5 Exergy balance summary of a component with work contribution.

F	$\dot{W}_{in}$
P	$\dot{m}\, b_{out} - \dot{m}\, b_{in}$
L	0
D	$\dot{D}$

$$\dot{W}_{in} = \dot{m}\, b_{out} - \dot{m}\, b_{in} + \dot{D} \quad \text{Eq. 2.21}$$

Where:

$\dot{D}$ is the exergy destruction rate in the process [kJ/kg],

b_{in}, b_{out} is the specific exergy of inlet and outlet respectively [kJ/kg].

If there is a heat transfer to the ambient:

Table 2.6 Exergy balance summary of a component with heat delivery to the ambient.

F	$\dot{m}\, b_{in} - \dot{m}\, b_{out}$
P	$\dot{Q}_{out}(1-\frac{T_0}{T_H})$
L	0
D	$\dot{D}$

$$\dot{m}\, b_{in} \; - \; \dot{m}\, b_{out} \; = \dot{Q}_{out}(1-\frac{T_0}{T_H}) \; + \; \dot{D} \qquad \text{Eq. 2.22}$$

Table 2.7 Exergy balance summary of a component with heat intake from the ambient.

F	$\dot{Q}_{in}(1-\frac{T_0}{T_C})$
P	$\dot{m}\, b_{out} \; - \; \dot{m}\, b_{in}$
L	0
D	$\dot{D}$

$$\dot{Q}_{in}(1-\frac{T_0}{T_C}) \; = \; \dot{m}\, b_{out} \; - \; \dot{m}\, b_{in} \; + \; \dot{D} \qquad \text{Eq. 2.23}$$

Overall exergy destruction in the system can be determined by summing up all exergy destructions in each and every component:

$$\dot{D}_{total} \; = \; \dot{D}_1 \; + \; \dot{D}_2 \; + \; \ldots \; + \; \dot{D}_n \qquad \text{Eq. 2.24}$$

By doing an overall analysis of the system:

Table 2.8 Exergy balance summary of the whole system.

F	$\dot{W}_{in} \; + \; \dot{Q}_{in}(1-\frac{T_0}{T_c})$
P	$\dot{Q}_{out}(1-\frac{T_0}{T_H})$
L	0
D	$\dot{D}_{total}$

$$\dot{W}_{in} \; + \; \dot{Q}_{in}(1-\frac{T_0}{T_c}) \; = \dot{Q}_{out}(1-\frac{T_0}{T_H}) \; + \; \dot{D}_{total} \qquad \text{Eq. 2.25}$$

The exergy efficiency (ψ) can be defined as the relationship between the product and the fuel and can be determined for every component and for the whole cycle:

$$\psi = \frac{P}{F} \qquad \text{Eq. 2.26}$$

Table 2.9 Summary of the exergy efficiencies of the components and the system.

$\boldsymbol{\psi_1}$	$\boldsymbol{\psi_2}$	$\boldsymbol{\psi_3}$	$\boldsymbol{\psi_{system}}$
$\frac{\dot{m}\, b_{out} - \dot{m}\, b_{in}}{\dot{W}_{in}}$	$\frac{\dot{Q}_{out}(1-\frac{T_0}{T_H})}{\dot{m}\, b_{in} - \dot{m}\, b_{out}}$	$\frac{\dot{m}\, b_{out} - \dot{m}\, b_{in}}{\dot{Q}_{in}(1-\frac{T_0}{T_C})}$	$\frac{\dot{Q}_{out}(1-\frac{T_0}{T_H})}{\dot{W}_{in} + \dot{Q}_{in}(1-\frac{T_0}{T_C})}$

2.2.3 Heating and Cooling Degree Hours

In order to evaluate if a period of time was particularly hot or cold, a calculation of the heating/cooling degree hours/days can be performed. These parameters give an insight of how much hotter or colder the outside temperature was over a period of time. They are used in numerous calculations, among the building industry mainly. It is calculated using a reference temperature as a base 10 ºC [56].

Heating Degree Hours (HDH) measure how much and for how long, the heating system was on. It can also be calculated daily instead of hourly and the base temperature is usually around 10 ºC for highly insulated buildings.

$$HDH = (T_{base} - T_{out}) \cdot hours \qquad \text{Eq. 2.27}$$

Cooling Degree Hours (CDH) define how much and for how long the temperature outside was above a certain value that the cooling system was on. The base temperature in this case is around 25 ºC.

$$CDH = (T_{out} - T_{base}) \cdot hours \qquad \text{Eq. 2.28}$$

In Figure 2.5 a graphic representation of these parameters can be seen. When the outside temperature is higher or lower than the base temperature, then all the hours above/below that set temperature would count into CDH or HDH.

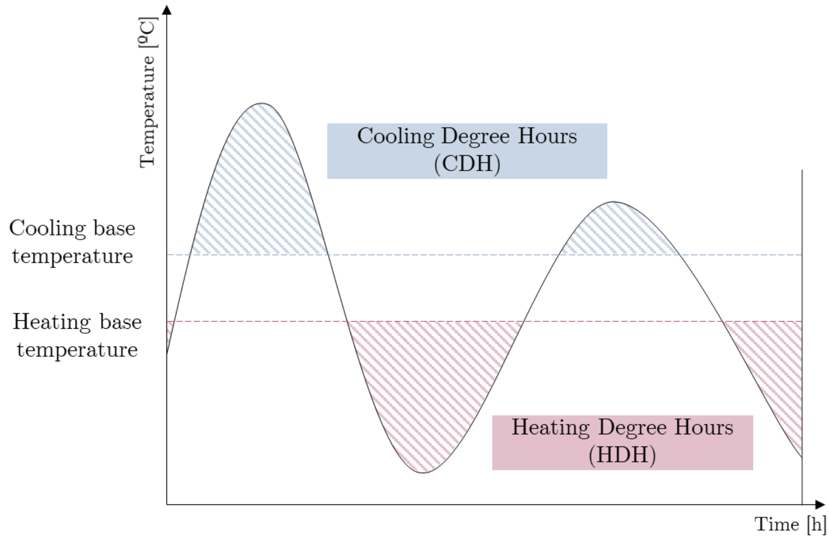


Figure 2.5 Heating Degree Hours and Cooling Degree Hours. [56].

The choice of the base temperature, is affected by several parameters [57]:

1) Indoor temperature: A higher indoor temperature will need more heating and therefore the base temperature for heating will rise.

2) Internal loads: Different loads such as lighting and equipment, transfer heat to the ambient, needing less heating during colder months and more cooling during warmer months. This will influence base temperatures.

3) Heat loss to the ambient: the required heating and cooling will depend on the amount of heat lost to the ambient.

According to this, a passive house with very good insulation, will need a lower base temperature for heating than a regular building.

However, since these calculations will be merely for comparison between different years in the same location, the base temperature is not of extreme importance.

2.3 Dimensioning of Heat Pump Systems

When dimensioning a heat pump system, a decision must be taken regarding the use of a peak load system. If a peak load system is not used, then the heat pump will cover 100 % of the heating loads throughout the year. This leads to a waste of energy and money most of the days, because the system will be over dimensioned and will only work at full capacity for a few days each year, thus leading to poor efficiency most of the year. If a peak load system is indeed used, one must decide which percentage of the heating loads will this system cover. If the percentage selected is too high, one will not make the most of the potential of a heat pump system, because a bigger peak load system than needed will be selected. In conclusion, it is crucial to determine this ratio properly, in order to save energy and money.

Power Coverage Ratio β

This parameter shows the percentage covered by each of the systems, with a typical value between 40 – 70 % [58].

$$\beta = \frac{P_{HP-dim}}{P_{HP} + P_{AS}} \qquad \text{Eq. 2.29}$$

Where:

P_{HP-dim}: Design power of the heat pump [kW],

P_{HP}: Power delivered by the heat pump [kW],

P_{AS}: Power delivered by the auxiliary system [kW].

Energy Coverage Ratio α

It represents the share of the energy coverage covered by the heat pump, with a typical value of 70 – 95 % [58].

$$\alpha = \frac{Q_{HP}}{Q_{HP} + Q_{AS}} \qquad \text{Eq. 2.30}$$

Where:

Q_{HP}: Heat delivered by the heat pump over a year [kWh],

Q_{AS}: Heat delivered by the auxiliary system over a year [kWh].

Equivalent Operation Time τ

It is the relationship between the yearly heat production and the dimensioned power of the heat pump. It quantifies how good the heat pump is used throughout the year. If this value is close to 8 760 h, it means that the system is working at almost full capacity throughout the year. Usually, in a residential building, this value is around 1 500 – 3 000 h [58].

$$\tau = \frac{Q_{HP}}{P_{HP-dim}} \quad \text{Eq. 2.31}$$

In Figure 2.6 a representation of the energy coverage ratio, the power coverage ratio and the equivalent operation time can be seen.

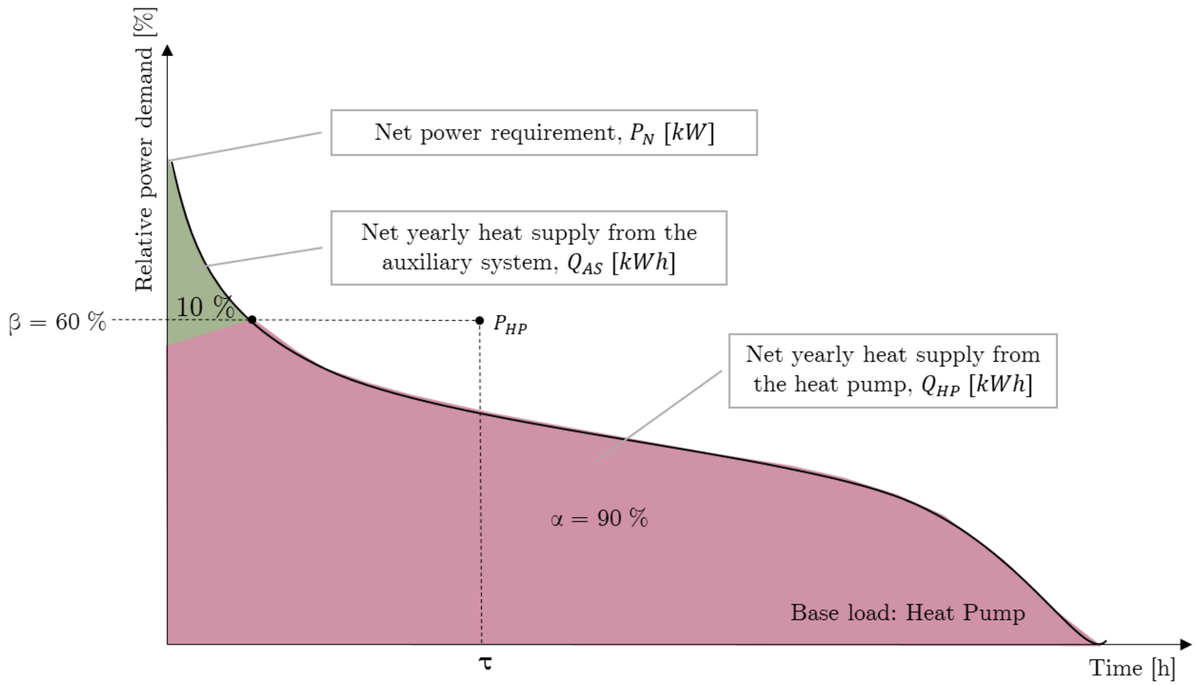


Figure 2.6 Example given to show the above-mentioned parameters. System with a heat pump covering 90 % of heating demands and a power coverage ratio of 60 % (modified from [58]).

Optimal Power Coverage Ratio

The relative energy saving parameter of the heat pump, depends on the systems SPF, the energy coverage ratio α and the efficiency of the auxiliary system η_{AS}:

$$\Delta E = \left[1 - \left(\frac{\alpha}{SPF} + \frac{(1-\alpha)}{\eta_{AS}}\right)\right] \cdot 100 \quad \text{Eq. 2.32}$$

In Figure 2.7 the representation of the above-described formula is shown.

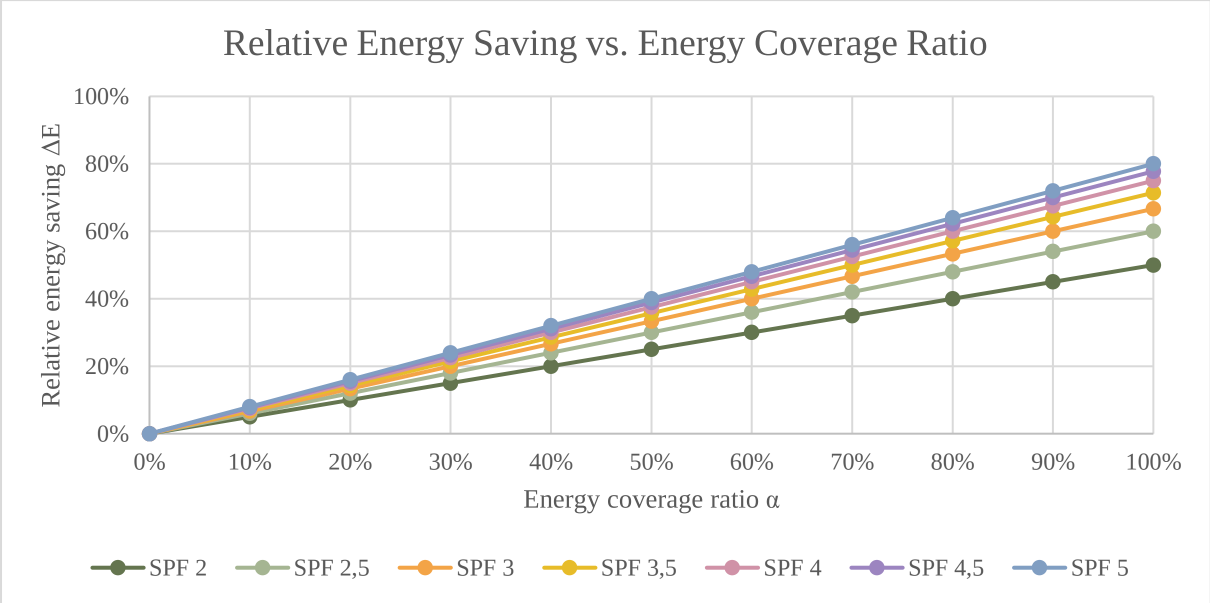


Figure 2.7 Relationship between relative energy saving and energy coverage ratio supposing an electric boiler as auxiliary system with an efficiency of 1 (modified from [58]).

Figure 2.8 shows the relationship between SPF, power coverage ratio and yearly costs. The lower SPF, the higher yearly costs. However, power coverage ratio has value at which yearly costs minimize. This value is around 0,7.

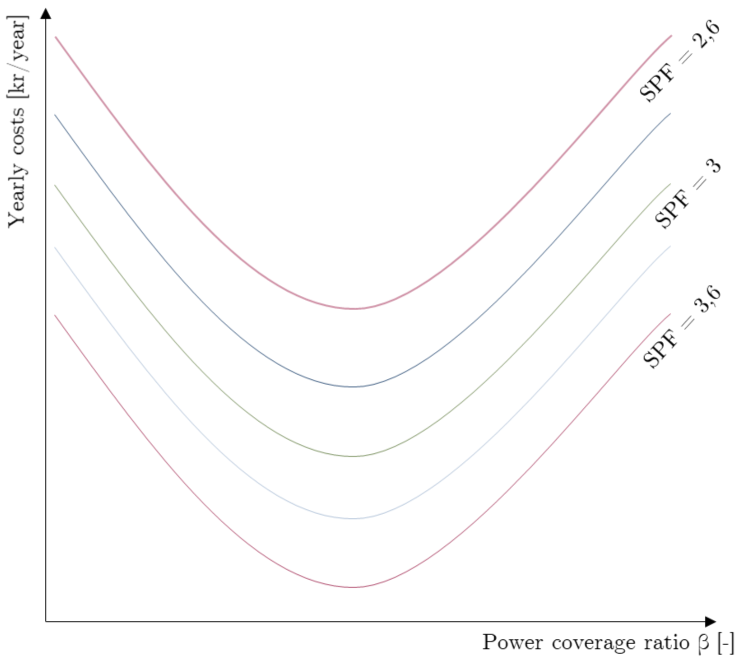


Figure 2.8 Heat pump system with high heat output (Tromsø) (modified from [58]).

2.3.1 Dimensioning of Combined Heat Pump/Cooling Systems

When the designed system also covers the cooling needs of the building, this is, when a reversible heat pump is designed, additional factors have to be taken into account.

If the cooling demand is moderate, the system is designed based on the heating demand, if it is capable of covering the entire cooling demand [58].

If the cooling demand is large, the system can cover the demand partially or fully. If it covers the whole cooling demand, the compressor will be designed in accordance to this demand. If it covers the cooling demand partially, the heat pump will cover the base load while an auxiliary system will cover the peak loads. Thereby the compressor will work under less partial loads [58].

To better understand the scale of the problem, two parameters will be introduced, the heating and cooling oversizing parameters [59].

$$\alpha_h = \frac{P_{installed} - P_{design\ heating}}{P_{design\ heating}} \quad \text{Eq. 2.33}$$

$$\alpha_c = \frac{P_{installed} - P_{design\ cooling}}{P_{design\ cooling}} \quad \text{Eq. 2.34}$$

A value close to 0, would mean that the selected system provides the necessary heat, a value lower than 0, would mean that the system is under-dimensioned, and a value higher than 0 would mean that the system is over dimensioned.

2.3.2 Part Load Operation

If the system is a fixed-capacity one, with no modulation possible, the system will cycle between on/off to reach the desired capacity. This leads to losses in performance which have to be considered [25].

Part Load Factor (PLF)

In order to take into account that part load functioning negatively affects the performance of heat pumps, the Part Load Factor (PLF) will be introduced. The European standard EN14825 [60] proposes these equations:

$$PLF = 1 - C_d(1 - PLR) \quad \text{Eq. 2.35}$$

This equation can be used for air-source heat pumps. However, for water-source heat pumps, the parasitical electrical energy consumption at stand-by has to be taken into account. For this reason, Eq. 2.36 is used:

$$PLF = \frac{PLR}{(1 - C_d) + C_d PLR} \quad \text{Eq. 2.36}$$

Where;

C_d: Cyclic degradation coefficient [-],

PLR: Percentage of the full load delivered by the heat pump [-].

$$PLR = \frac{P_{max}}{P_{design}} \quad \text{Eq. 2.37}$$

To be noted is that the coefficient C_d can be determined for cooling mode and used as well in heating mode calculations. This value can be determined experimentally or be 0,25 by default for air-to-air and water-to-air heat pumps, or 0,9 for air-to-water and water-to-water heat pumps [60].

On-off cycles have a strong impact on the efficiency of the system, because when the compressor restarts, it has to restore the pressure difference between condenser and evaporator [59]. Corberán et al. [49] studied experimentally the partialization losses of on/off operation of water-to-water heat pump units and concluded that the start-up losses were not relevant when studying these kinds of systems. These losses are especially relevant in systems using capillary tubes. However, they concluded that the stand-by losses were something to take into account at loads below 20 %, which contributed to a degradation of the PLF (in water-to-water systems). These losses, according to Corberán et al., are mainly caused by consumption of the valves (to keep them energized), consumption of the electronics and consumption of the crankcase heater.

2.3.3 Problems Related to Oversized Systems

Oversized systems often lead to numerous undesired consequences such as poor efficiency and increased costs. An oversized system will have a higher investment cost than a properly sized system and relatively high operating costs with generally low energy savings [48]. Higher capacity often implies larger pipes, pumping systems and valves, which increase the cost of the total installation. Usually, larger motors require higher currents, so the electricity cost increases as well. Oversized systems, lead to over-cycling and excessive low-load operation [24].

Some tips to prevent oversizing of systems are listed below [24]:

- Reduce the use of safety factors.
- Use precise methods of calculation.

- Use simulation programs that can estimate the thermal storage in the building.
- Use equipment which enables part load operation: Variable speed pumps and fans.
- Allow flexibility if the internal loads are not fully known.

2.4 Control Strategies in Heat Pump Systems

Designing a proper control strategy is crucial for the well-functioning of the energy system. It brings together different parts of the system such as distribution systems, heat pumps and peak load systems by using sensors and controllers.

Many factors have to be taken into account before designing a control strategy. First, a correct dimensioning of the system must be made, adjusting to the heating/cooling demands of the building and an appropriate heat source system must be selected. Then energy efficient components should be selected in order to achieve the maximum COP [26].

Compressors and Heat Pumps: capacity control and minimum start/stops.

Variable Speed Drive (VSD), unloading of cylinders and on/off control are factors that help operate compressor at maximum efficiency. VSD compressor frequency can vary from 30 to 90 Hz, which can be controlled by using the return water supply or the outdoor temperature [61]. One of the advantages of using capacity control, is the decrease in thermal power, which reduce the temperature jump leading to higher COPs [62].

In Figure 2.9 three scenarios are studied. The first, a system with four scroll compressors and no regulation available, this is on/off control. The second, two piston compressors with cylinder unloading. And the third, one piston compressor with Variable Speed Drive. It can be seen that the scenario with regulation available can adapt better to the building's needs.

Peak Load System: It should only deliver heat when heat pumps are working at full capacity and they do not cover the heat demand (and in extreme cases such as very low outside temperature in air-to-air heat pumps or when the system fails).

As seen in Figure 2.7, energy savings drop significantly when the peak load system covers more percentage of the energy demand.

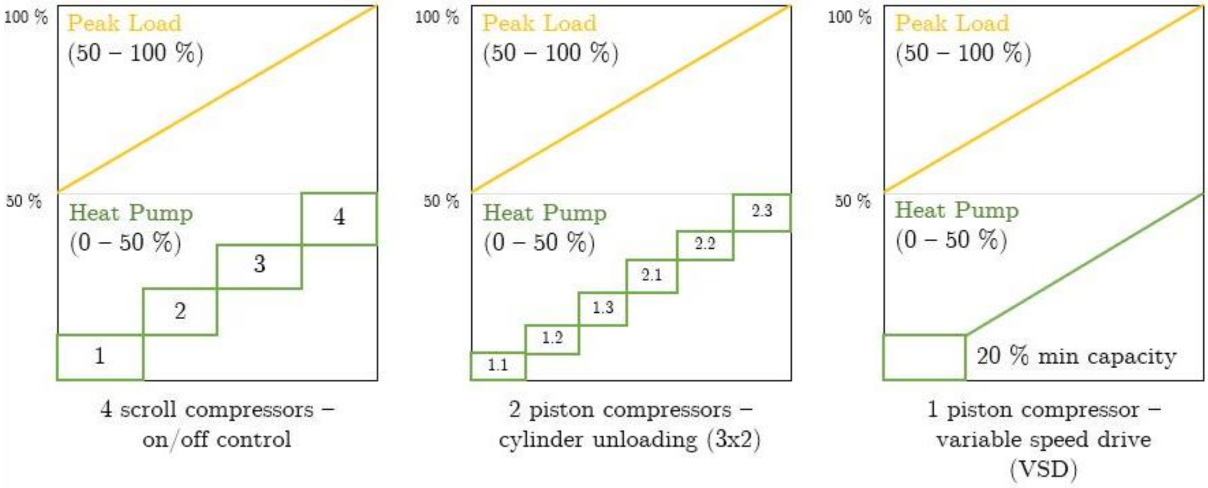


Figure 2.9 Comparison of three different control strategies in a one-unit heat pump system (modified from [26]).

Distribution Systems: for heating, the supply temperature should be as low as possible, and for cooling as high as possible. This is because minimizing the temperature lift of the heat pumps will increase the COP (Eq. 2.4) [26].

Cooling Mode: Avoid changing to cooling mode if "free cooling" is an option. Night setback could be a feasible option.

In addition, several safety equipment should be installed to ensure safe operation [26]:

- Pressure switches (evaporator side (min. pressure), condenser side (max. pressure) and compressor (oil pressure)).
- Thermostats (compressor, electric motor).
- Active and passive safety devices (safety valves, rupture discs, overflow valves)

It is of great importance to install the instrumentation in the right place, otherwise the measurements will not be as accurate and could lead to a misuse of the control system (for example, if a temperature sensor is too far from the accumulator tank, the system will think that more heat is needed and will turn on the heat pumps, when in reality it is not needed).

Using these installed devices, the next step is to instal a PLC (Programmable Logic Controller) which will control in open or closed loops the compressor, expansion valves, liquid vessels, pumps and fans and defrosting of air-cooled evaporators as well as monitor data. The last step is to communicate with the users to set up alarms and optimize the energy use. These are the steps followed by the Building Energy Management System (BEMS).

2.4.1 Building Energy Management System (BEMS)

A Building Energy Management System (BEMS), is a tool that helps monitor and manage the facility. It is a computer-based system which gives the people in charge a central system from which control energy related building operations, such as lighting and HVAC. BEMS are also able to store, process and monitor data at a basic level [63].

That one may make the most of these kinds of systems and gain knowledge to make smart decisions, some data inputs might be useful [63]:

- Total energy consumption of the building and its equipment. It is important to know the electrical consumption of the building and how each device contributes.
- Occupants' behaviour. Depending on the activity and comfort level and behaviour patterns, one may decide between one or another saving strategy.
- Weather data.

With this data, some patterns in the energy consumption might be visible, which could then potentially predictable and used for saving energy, and consequentially, money.

2.5 Latest Technology

In order to see which technologies are available nowadays, or which have been developed lately, an analysis of the latest technologies will be carried out. Several objectives will be pursued: lower heating or cooling demands, lower electricity consumption, shifting loads or better efficiencies, among others.

Fresh Air Heat Recovery

This technology, utilizes the heat extracted from the ventilation system to preheat fresh air entering the building. In severe cold/cold regions like Norway, these systems use usually anti-freezing measures such as electrical preheating or Earth-to-Air Heat Exchangers (EAHE). Nevertheless, these freezing problems may result in lower efficiency and must therefore be further studied [64].

Thermal Photovoltaic Cells (PVT)

Thermal Photovoltaic Cells use heat accumulated in the cells as well as produce electricity. The use of PVT allows the recovery of lost heat while increasing electrical efficiency. When coupling this technology with a heat pump, the compressor's electrical work could be covered by the PVT. According to [32], a photovoltaic module, uses only 18 % of solar irradiation to turn it into electricity, therefore 82 % of the irradiation is lost. Furthermore, heat dissipation increases the cell's temperature, diminishing its efficiency. Consequently, cell efficiency gets reduced 0,2 – 0,5 % for every 1 ºC temperature increase [32]. As a refrigerant both water and air can be used, although fluids that have lower phase change temperature are more suited for this application.

Night Cooling

Night cooling is a low-cost, easy technique to lower the temperature during the night when the outside temperature during the day is too high. Although it is not a new technology and has been used for years, it is worth mentioning its importance when lowering cooling demands.

Wang et al. [65] studied in 2009 an office building with lightweight construction in three different cities in China. They pointed out that ventilation rates, ventilation duration, climatic conditions and building mass were the parameters that affected the most the performance of night cooling. They concluded that energy saving in buildings with night cooling in northern China were higher than in buildings without the strategy.

In 2020, Guo et al. [66] studied the effects of night ventilation in the Percentage of Dissatisfied during unoccupied hours (aPPD), which showed an increase from 7,5 % to 15 %.

However, very high total cooling energy consumption savings were reached (8,8 – 82,5 %) using three different SPF night cooling systems.

Thermal Mass

On the contrary to light constructions, heavy constructions are able to store some of the solar energy that hit the windows. In cold climates, the use of heavy mass in interior walls, ceilings and floors is useful to control overheating [21]. In terms of materials, concrete and bricks are one of the best regarding thermal capacity and heat conductivity. A high thermal capacity and moderate heat conductivity, make these materials a perfect energy reservoir, due to the charge and discharge duration, which are according to the diurnal cycle. This makes it possible to charge the material during the day, and discharge it during the night, when the loads are lower.

Figure 2.10 shows the comfort improvement due to the use of thermal mass in several Norwegian cities during summer.

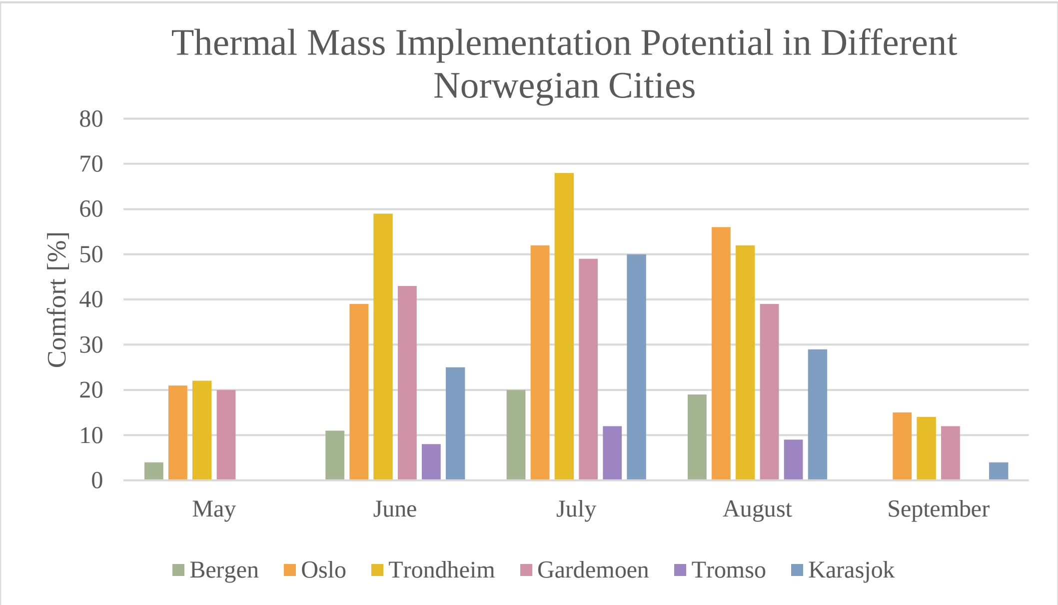


Figure 2.10 Monthly thermal comfort improvement potential in Norway if thermal mass is used [21]. Originally from [67].

However, room acoustics can be a challenge so special attention must be paid to this matter.

2.5.1 Dual Source Heat Pumps

These systems, unlike traditional heat pumps, use not one but several heat sources. The advantages of these heat pumps, is that they make the most of two sources, obviating the defects of a single one.

Ground-Source Heat Pump and Thermal Photovoltaic Cells (PVT)

Back in 2012, Bertram et al. [27] studied the possibility of coupling a ground source heat pump (12 kW) with an unglazed photovoltaic-thermal collector. PVT were used to increase the temperature level of the heat pump and therefore improve its performance. They concluded, that the SPF improvement was of 0,36 the first year and 0,51 after 20 years, which is equal to a 9 % and 13 % of electricity saving accordingly. However, the simulated system was located in Frankfurt, Germany, so results cannot be directly extrapolated to Norwegian climate, due to for example, snow covering of the cells during winter.

A combination of GHSP that provides heating and cooling, with thermal photovoltaic cells (PVT) and an air-to-brine heat exchanger that maintain a constant inlet temperature was installed in 2020Park in Stavanger, Norway in 2017 [68]. The addition of the two extra heat sources helped in this case again with the problem of unstable inlet temperatures at the GSHP, obtaining a fairly constant COP throughout the year. The PVT have micro channels installed behind the PV cells, so some cooling is provided. This helps with both the electricity production and the heating of brine precedent from the boreholes, as mentioned before [68]. The air-to-brine heat exchanger works when the outside temperature is above 2ºC and reduces the heat needed from the ground. As peak load system, the installation is connected to the district heating grid to cover peak loads. Alonso et al. [68] suggests that the excess heat produced in the summer could be stored in the ground, but that due to the configuration of the current system, it is not possible. In their research of the data gathered from a year of using the installations, their most remarked result is the positive outcome of using the air-to-brine heat exchanger to lift the temperature from the boreholes. If all the heat was to be extracted from the ground, it would have reached freezing levels. However, monitoring of the ground temperature would have been crucial to determine this. They also suggest the use of a thermal storage system (TES) to store the excess heat from the cooling season, such as PCM or thermochemical storage.

Direct-Expansion Solar-Assisted Air-Source Heat Pumps (DX-SAHP)

First proposed by Sporn et al. back in 1955 [28] and later studied by Zaheer-Uddin et al. [69] in 1987, coupling solar technologies with heat pump systems has been studied. In DX-SAHPs, the solar collector acts as an evaporator, resulting in a lower evaporating temperature than the ambient, enabling the evaporator to reduce heat loss and even recover heat from the environment. Later, in 1998, Torres et al. [70] analysed the exergy efficiency and optimization of a solar assisted heat pump system, concluding that large irreversibilities occurred at the collector-evaporator. However, they proposed an increase in the heat recovery area of the condenser to lower the total exergy losses of the system. Chow et al. [71] found in their study conducted with the Hong Kong climate, that the yearly COP rose up to 6,46, much better than the one in the conventional configuration.

One of the main drawbacks of ASHP systems, as explained earlier, is the frost accumulation in the evaporator. A research carried out by Huang et al. [72], investigated the effect of ambient conditions to avoid freezing problems in air-source heat pumps when using use solar-assisted direct expansion. Their research concluded that even solar irradiance as low as 10 W/m^2 would totally prevent frost accumulation when the temperature is above -3 ºC and the RH is 70 %.

Solar Collector and CO_2 Heat Pump

Because solar irradiation is both unpredictable and intermittent, in the majority of climates it cannot be used alone for heating purposes. Chen et al. [73] studied the combination of a solar collector with a CO_2 heat pump for DHW and space heating purposes. Thanks to the CO_2 trans-critical cycle characteristics, it is especially suited for hot water heating applications. The conclusions made were that the COP of the system was improved and that electricity was saved, up to 53,6 %.

2.5.2 Thermal Energy Storage

Thermal energy storage comes as a solution to the problem of mismatch between availability and demand of energy. It allows shifting of peaks to periods where the electricity price is lower, as well as provides a solution to unstable availability.

Phase Change Materials (PCM)

These substances absorb/release large quantities of latent heat when changing their physical state (solid to liquid, liquid to gas etc.).

A study performed by Tan et al. [31], which was focused on the use of PCM for space cooling of office buildings in Gothenburg, determined that even though the system used only 36 % of the theoretical storage capacity, still 99 kWh of cooling was shifted from peak hours. This is of significant importance especially in highly insulated office buildings, because of the high cooling demands during summer months.

A comparison carried out by Moreno et al. [29], determined that a PCM tank was able to store 35,5 % more cooling energy than a regular water tank (sensible heat). However, the charging time was 4,55 times higher.

PCM materials can be sorted in three types: Organic, Inorganic and Eutectics. Organic compounds have the advantage of homogenous melting, while inorganic compounds have a higher thermal energy storage capacity [74].

Hybrid Space Cooling System with Night Ventilation and Thermal Storage

Another solution to high cooling loads is proposed by Zhou et al. [75]. They combine PCM with night cooling to obtain 76 % savings in daytime cooling energy consumption (Beijing climate). The thermal storage technology used are PCM plates attached to inner surfaces of walls and ceilings, on which cool is charged during night hours.

PCM and Ice Thermal Energy Storage (ITES) Tank

Rahdar et al. [30] compared two different technologies for air-conditioning systems in office buildings in Iran. They compared an ice thermal energy storage tank (ITES), with a PCM tank. Both of the systems were installed in the evaporator of the heat pump. The tanks accumulate heat extracted from the air handling unit, and get refrigerated by the evaporator. The conclusions made were that PCM had higher savings in power consumption (7,58 % vs. 4,59 % of the ITES) but also a higher payback period (5,56 years vs. 3,16 years).

Low Temperature Seasonal Thermal Energy Storage

Thermal loss is a big problem in thermal energy storage systems, but it can be lowered if the temperature of the stored energy decreases. However, as studied by Hesaraki et al. [76], the combination of these systems with heat pumps can carry multiple advantages. For example, solar energy and heat excess from the building can be stored in the storage system during summer, and used as a heat source during winter. Furthermore, heat pumps help keep the temperature stratified inside the tank, which is beneficial. According to Ghaddar [77], a 20 % increase in energy delivery is achieved when using a stratified tank, compared to a fully mixed one.

2.5.3 Control Strategies

Model Predictive Control (MPC)

As explained in Section 2.3.2, part load operation of heat pumps systems leads to losses in efficiency. For this reason, the installation of variable speed compressors is a good option if a better adjustment to the heating and cooling loads wants to be reached. After installing a variable speed system, a proper control of the system is crucial to reach the best adjustment possible, as well as reducing energy consumption and emissions. For this, Model Predictive Control (MPC) strategies are very useful. A MPC strategy is one that takes into account predictions during the design of the control system while satisfying the operation constraints. These constraints can be indoor temperatures, DHW temperatures to avoid the growth of legionella or maximum energy consumption [78]. In a study conducted by Pean et al. [35] it was concluded that a MPC strategy is able to shift loads from high periods to low periods very efficiently using thermal energy storage systems. They studied a reversible heat pump (11 kW heating, 7,2 kW cooling) using variable speed compressors, concluding that the HP using MPC had a higher COP than the reference case (no MPC) due to milder temperature supply. However, they also pointed out problems while charging the DHW tank, due to lack of information of local installed controllers. For this they recommended realizing tests to the heat pump to characterize its behaviour, which increases the development costs.

Combined Feedforward and Feedback Structure

In a system using both available solar thermal energy and an air-to-water heat pump, an optimal control strategy was studied by Hosseinirad et al. [34]. The previously studied similar situations [33], where focused on obtaining a constant temperature in the DHW system, rather than an interval where users would be satisfied, mainly because the studied system was a power plant. In addition, they were not focused on reducing the electrical consumption, which is in a residential/commercial building, something to aim to. The proposed control structure in [34] consists of three sub-controllers (offline, robust feedforward and feedback). Using weather forecast, historical data and the mathematical model, the signals of the disturbances were calculated 24 h ahead. Nevertheless, these could be changed based on real-time values of the disturbances. If the DHW temperature is inside the allowable temperature range, the feedforward controller is active, if not, the feedback controller is working, adjusted to bring the temperature back in the desired range, which is faster than if done with the feedforward.

In Figure 2.11 the mentioned control strategy can be seen. Instead of focusing on maintaining a certain indoor temperature, the allowed value lies on an interval where the occupants would feel comfortable.

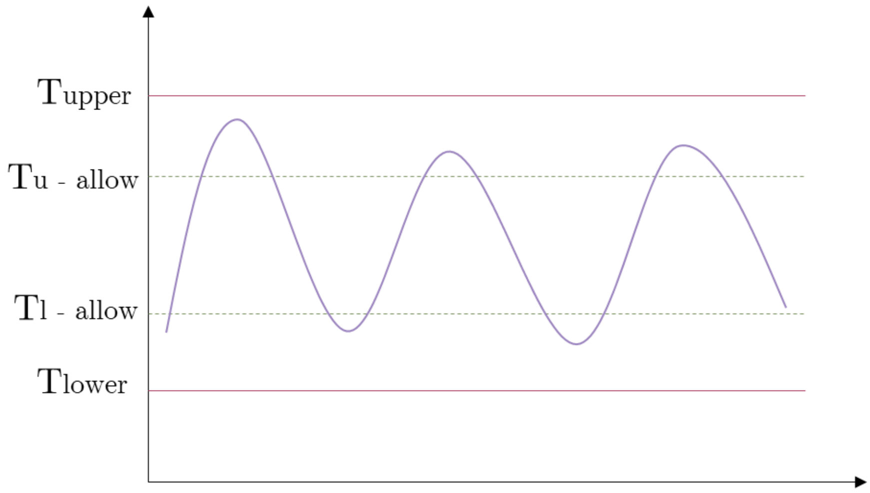


Figure 2.11 Comparison between desired and allowable temperatures (modified from [34]).

2.6 Energy Centrals in Large Buildings in Cold Climates

That one may have a bigger understanding of large energy systems, a few office buildings in Norway will be studied.

2.6.1 Powerhouse Kjørbo, Sandvika (1985/2014).

Located in Sandvika and built in 1985, this office building was refurbished in 2014 with the ambition of turning it into a more sustainable building (building standard plus energy building) [79]. With a heated area of 5 180 m^2, it has two heat pump units to cover space heating, ventilation air heating and DHW. As working fluids one of the heat pump uses R410a and the other R407c, and both use bedrock as a heat source. Radiators and ventilation heater batteries make up the heat distribution. Finally, as a back-up system it uses district heating. This building, does not have an auxiliary system to cover peak loads, instead has a heat pump system that covers 100 % of the heating and cooling demand [79]. The borehole system provides cooling during summer months.

Table 2.10 Technical characteristics of Powerhouse Kjørbo [79].

Thermal Energy demand [kWh/m^{2}a]	Space heating + Ventilation air heating: 19,1 DHW: 1,9 Space cooling: 1,8
Installed power	HP1: 65 kW HP2: 8,5 kW
Energy coverage factor	HP1: 100 % HP2 100 %
Seasonal COP	HP1: 3,9 HP2: 2,9
Temperature levels	Space heating: 50/40 °C Ventilation air heating: 50/25 °C

2.6.2 Powerhouse Brattørkaia, Trondheim (2019).

Built between 2017 and 2019, Powerhouse Brattørkaia is the biggest energy-positive building in Norway (ZEB-COM%EQ) [79]. Low U-values, low thermal bridges, low air leakage rates and an outstanding and revolutionary use of solar power define Powerhouse Brattørkaia. In order to heat both its 14 000 m^2 of heated area and its DHW demand, a seawater heat pump system of 140 kW was installed. This system takes care of 100 % of space heating and 50 % of DHW production [79]. Distribution of heat throughout the building is done by radiant floor heaters and large radiators to achieve a low-temperature distribution. Regarding the cooling demand of the building, it is entirely taken care by sea water.

Table 2.11 Technical characteristics of Powerhouse Brattørkaia [79] [80].

Total net energy demand [kW/m^{2}a]	46
Heat Pump [kWh/a]	185 000
Electricity generation	3 000 m^2 of PV 485 000 kW/a
Energy consumption reduction	Advanced BEMS

2.6.3 The SWECO Building, Bergen (2016).

An 80 % of energy consumption reduction compared to a standard TEK10 building is a reality at the SWECO building in Bergen [81]. Heating and cooling demands are covered by reversible heat pumps with natural refrigerants (R717), using geothermal energy as energy source. With a 110-kW heating and DHW demand and a 195-kW space cooling and process cooling demand, the SWECO building covers its demands with a 195-kW ground-source heat pump. The auxiliary system to cover peak loads and to act as a back-up system is district heating. Waste heat from shops is used to preheat DHW [48].

2.6.4 Norsk Hydro, Vækerø, Oslo (2003).

A refurbished building of 60 000 m^2, this office building uses a two-stage ammonia (R717) heat pump with a desuperheater to cover its energetic demands (2 MW of maximum heating and 2 MW of maximum cooling) [82]. Cooling demand is usually entirely covered by sea water, but in case of need the heat pump can cover those needs. The distribution system is made up of radiators, heater batteries and cooling batteries. The heat pump system, operates with two tonnes of ammonia and employs sea water as a heat source. As peak load cover system (3 MW), existing oil-fired boilers and electric boilers are used [82].

Due to the use of a very toxic substance as it is ammonia, safety measures were taken when designing the system. Gas-tight machinery room, leak detectors and 2-stage ventilation systems to name a few [82].

2.6.5 Alnafossen kontorpark, Brynseng (2003).

With an area of approximately 35 000 m^2, this office building uses two R407c units with a total of 1 200 kW of heating capacity [82]. The heat pump units use bedrock as a source, by means of 54 energy wells. Cooling demands of the building are largely covered by free cooling by using cooling batteries and it is distributed by a hydronic system. Space heating distribution is carried out by radiators and surplus heat from the building is supplied to the boreholes to maintain annual thermal balance [82].

*Chapter **3***

Case Study: GK Miljøhuset

In this chapter, a case study will be brought forward for study. The building in question is GK's headquarters in Oslo. The energy system will be thoroughly studied, as well as all the aspects required for the later analysis and discussion.

3.1 Context

GK is a technical contractor and service partner with competences in HVAC, building automation, electrical installations and plumbing. It is the leader in Scandinavia in its sector, with over 3 000 employees over 80 offices in Norway, Sweden and Denmark [83].

Located in Ryen, Oslo (Norway), Miljøhuset is the headquarters building of GK in Norway. Built between 2010 and 2012, the company had no intentions of building one of Norway's most sustainable buildings and the first office building passive house in Norway at the time. Nevertheless, an analysis of the possibilities of the building, made GK realise that with a minimal extra investment, the benefits for both themselves and the environment were uncountable. By increasing the investment costs in 8 million NOK, of which 4 million NOK were covered by Enova, the building would stick to the passive house standard (NS3701). Calculations made at the time, showed that Miljøhuset (marked A+) would reduce in 1,1 GWh the use of energy compared to a C-marked building, which was the government's requirement at the time [84].

Table 3.1 Comparison between NS3701, TEK17 and Miljøhuset values [84].

Parameters	NS 3701	TEK17 (block of flats)	Papirbredden 2	Miljøhuset
U-value outer walls [W/m^{2}K]	0,15	0,18	0,15	0,14
U-value roof [W/m^{2}K]	0,13	0,13	0,13	0,1
U-value floors [W/m^{2}K]	0,15	0,10	0,15	0,07
U-value windows and doors [W/m^{2}K]	0,8	0,8	0,8	0,78
Air leakage rate per hour at 50 Pa pressure difference	0,6	0,6	0,6	0,23

3.2 The building

3.2.1 Description and General Characteristics

With a heated area (BRA) of 13 650 m^2, Miljøhuset has two air-source heat pumps which can work on winter or summer mode. In summer, the system works as a cooling machine and provides chilled water to the installations. In winter, the heat pumps produce hot water for the installations, combined with heat extracted from data rooms and helped if needed by an electrical boiler. During summer operation, the electric boiler is switched off. To avoid unnecessary switching from summer to cooling mode due to unstable weather, the system has a fixed switching hysteria of 5 ºC [85]. However, nowadays the change between seasons is being done manually by a chosen date.

The working fluid used in the heat pumps is R410a, with a mass of around 67,5 kg, whilst the working fluid used in the distribution system is a glycol-mixed water [86]. The distribution of heat around the building is done through the ventilation system.

The building also has a cooling system which works independently in order to cool data rooms. Heat extracted is used to preheat DHW as well as to preheat water entering the accumulator tank located before the heat pump. This system works all year round, with variation only in the temperature set point needed in the preheat [85]. On top of this, the building has a solar shading system on every façade but the north one to limit the cooling demand during the summer [87].

In order to produce DHW, heat from the data rooms is used to preheat the water that later will be accumulated and heated with an electric resistance to the desired temperature (approx. 70 ºC).

The building counts with a solar production site with 180 panels and covers an area of roughly 293 m^2 [88].

Table 3.2 Seasonal mode summary of the energy system.

	Winter mode	**Summer mode**
Heat Pumps	Heating	Cooling
Electric Boiler	On (40 %)	Off
Data Room Cooling	On	On
DHW	On	On
Dry Cooler	On if necessary	On

The ventilation system consists of six balanced ventilation systems (Specific Fan Power (SPF) set to 1,2 kW/m^3s) all of them equipped with highly efficient rotary heat exchangers. Air leakage of the building is 0,23 at 50 Pa [87]. One of this ventilation systems can be seen in Figure 3.1.

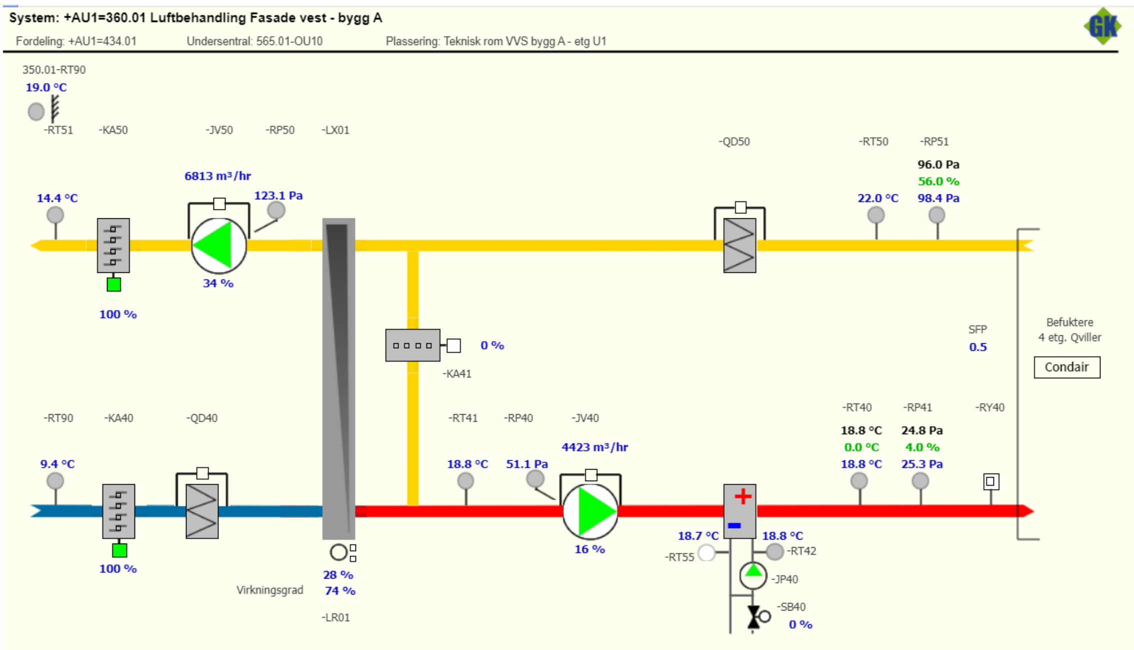


Figure 3.1 One of the ventilation units at Miljøhuset [88].

The entrance to the garage (approx. 125 m^2) has a snow melting system which prevents snow to accumulate and makes the circulation of vehicles easier during winter months.

Table 3.3 Summary of the energy demands of the building from a source[89].

Space Heating	**Space Cooling**	**Domestic Hot Water (DHW)**
4,3 [kWh/m^{2}a]	7,8 [kWh/m^{2}a]	5 [kWh/m^{2}a]

Because of lack of data, one cannot know for sure if these demands are correct. Table 3.3 shows data from a simulation done by GK in 2012, according to [89]. This simulation was not found as a first source.

In terms of the building itself, it has five stores above ground and one underground. The underground floor is built with watertight concrete and isolation (350 mm EPS). The roof consists of perforated deck elements externally insulated (450 mm standard insulation) [87]. It has three different areas, named Building A, B and C respectfully. Each building has 2 ventilation systems.

Building A

In data found from the company's servers, in this building only office space is taken into account. Therefore, heating and cooling loads are the only ones considered. The snow melting system is also taken into consideration in this building.

Building B

In this building a canteen and kitchen can be found, as well as office spaces and the garage. On top of the heating and cooling loads, also electricity for the kitchen is taken into account.

Building C

In building C also, a garage is considered, but in this case, there are also electric car chargers.

3.3 Energy System: Used technology

3.3.1 Main Energy Systems

Energy Central System

As mentioned earlier, the building has two air-source heat pumps which work on R410a. Both of the heat pumps used in the building are EAGLE HP T.240.P4.Y2 D from the Italian company RcGroup.

Table 3.4 Summer working mode characteristics of EAGLE HP T.240.P4.Y2 D [86].

Cooling capacity	kW	226
Compressor power input	kW	73,4
Water temperature (in/out)	ºC	12/7
Water flow rate	m^3/h	43
Pressure drop	kPa	57
Ambient temperature	ºC	35

Table 3.5 Winter working mode characteristics of EAGLE HP T.240.P4.Y2 D [86].

Heating capacity	kW	276
Compressors power input	kW	58,2
Water temperature (in/out)	ºC	23,8/30
Ambient temperature	ºC	2,5

In Figure 3.2 one can see one of the heat pumps of the system (IK40). It consists of 5 main components: an air heat exchanger, 4-way valve, scroll compressors, expansion valve and water heat exchanger. Since the system has two circuits, the components' quantity is doubled. Because the system has to be reversible, it is fundamental to have a 4-way valve to make it possible. This allows the refrigerant to flow in the other direction when needed. This can be seen in Figure 3.3.

In cooling mode, the water heat exchanger works as an evaporator, absorbing heat from water, while in heating mode it works as a condenser.

The working principle of the system, relies on the acquisition of heat from the data rooms, which is used as a base. If the heat recovered from the data rooms is not enough, the heat pumps will start working until the indoor temperature reaches the desired temperature. On the other hand, if the building is sufficiently heated, heat from data rooms will be rejected to the hot water system or to the air, and if some cooling is needed, the heat pumps will switch to cooling mode.

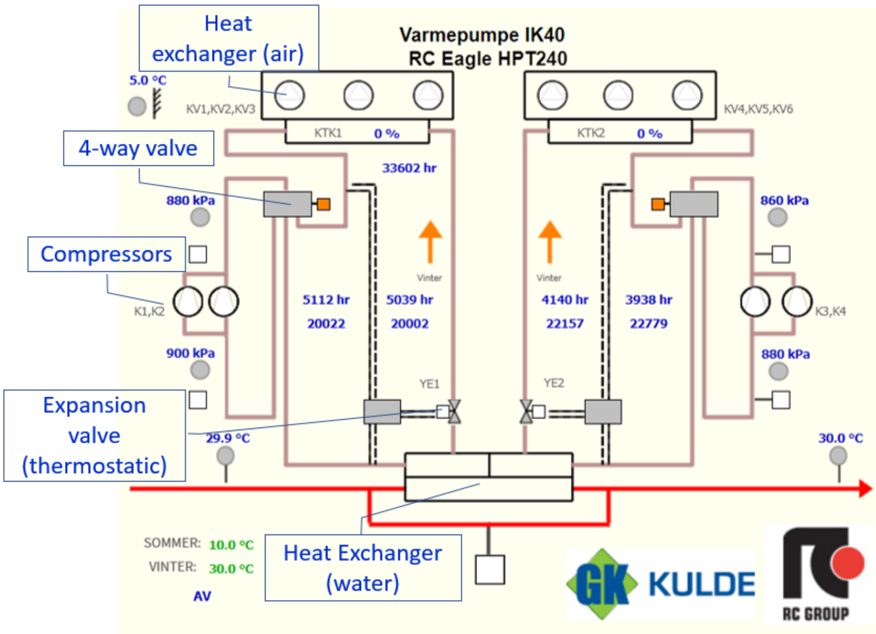


Figure 3.2 Insight of one of the heat pumps (modified from GK's servers) [88].

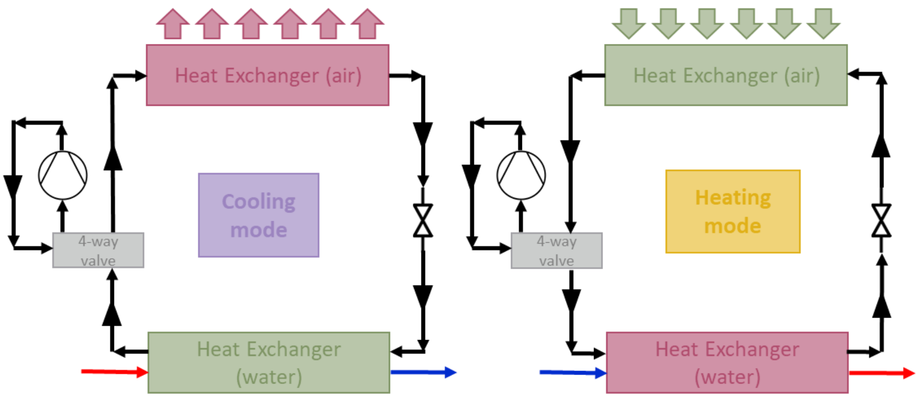


Figure 3.3 Heating/cooling modes of the heat pumps.

Apart from the heat pump system, the energy central has an accumulator tank that accumulates water before the heat pump so there is always supply of hot water to the system. This device allows for the heat pumps to not be working continuously. However, according to [90], the accumulator tank is not being used as it is nowadays and the heat pumps are turned on/off when needed. It also has an air separator which is in charge of removing the air from the fluid which could damage pumps and other equipment and even hinder heat transmission.

The system is equipped with three pair of twin pumps to transport water from/to the cooling system, to circulate water through the electric boiler and to allow water to flow from the heat pumps to the accumulator tank. The pumps take turns at working, with equal working times. When one of them fails, the other works as a back-up. The pair of pumps that pump hot water to the building (JP42/JP43) are variable speed pumps. They use a pair of pressure sensors located at the inlet and outlet of the system to be able to regulate their speed.

As auxiliary system to cover peak loads and to act as a back-up system, the system has an electric boiler system (approx. 130 kW) and 200 W electric heating rods located in the offices [22]. The electric boiler is connected in parallel with the heat pump. The capacity of the electric boiler is fixed to 40 % when working simultaneously with the heat pumps. Nevertheless, if the temperature outside is so low that the heat pumps work very poorly in terms of efficiency (-15 ºC), the electric boiler can be turned on and be used at a 100 %. This device is bypassed when not in use.

The energy central provides heating and cooling to the building through a hydronic distribution system, which then heats the air inside each zone in the building via the ventilation system. This allows a better heat distribution that the usual radiators and lowers the power consumption [89]. A full overview of the system is provided in Figure 3.4.

Finally, the system provides heat to a snow melting system which only works during winter.

Table 3.6 Other important characteristics of EAGLE HP T.240.P4.Y2 D [86].

Compressors	Quantity	nr.	4
	Type	--	scroll
	Capacity steps	nr.	4
Plant water	Water volume	l	20,3
	Max. water flow rate	m^3/h	55,6
Axial fans	Quantity	nr.	6
	Airflow	m^3/h	132000
	Power input	kW	11,64
Refrigerant	Type	--	R410a
	Total charge	kg	67,5
	Gas circuits	nr.	2

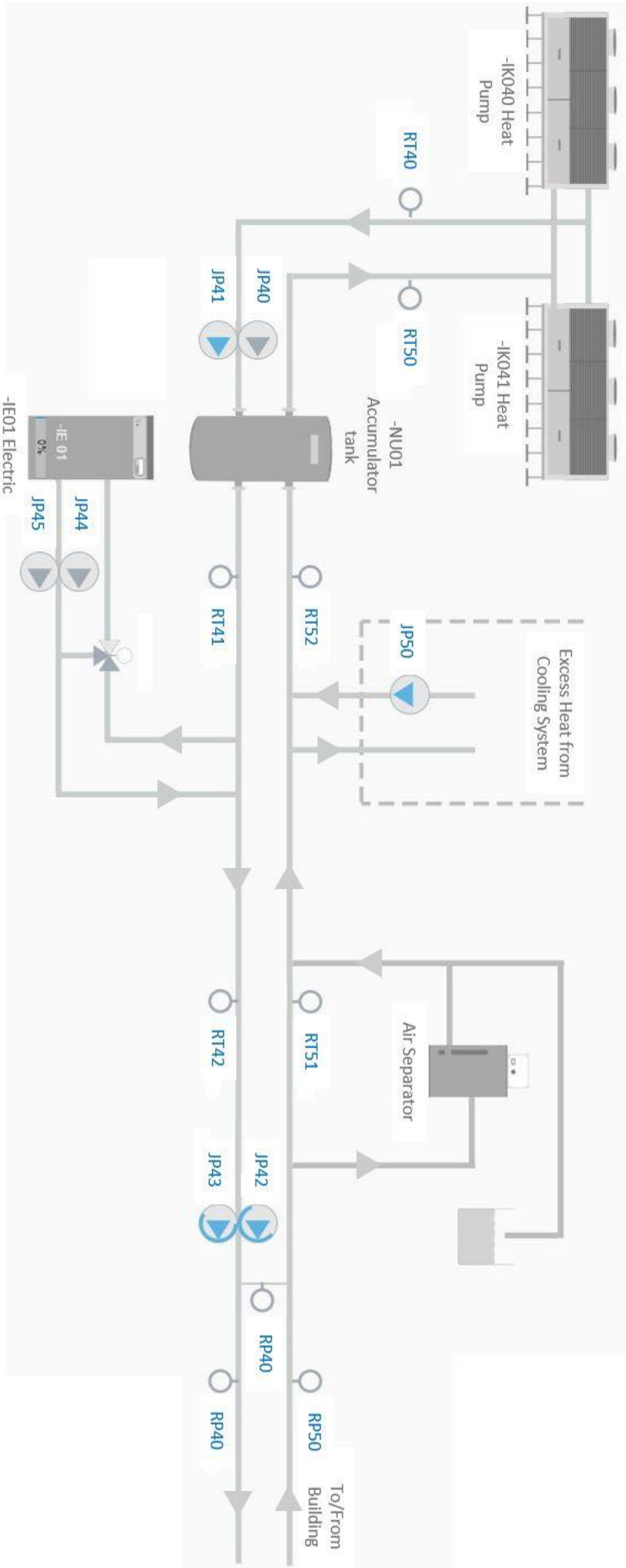


Figure 3.4 Energy Central of the building (modified from GK's servers [88]).

Cooling system

The cooling system is in charge of collecting heat from data rooms. Excess heat is then used to preheat water (both for DHW and for the building) all year round. In the case that there is still excess heat, a dry cooler is used to further condensate the glycol mixture. This dry cooler has 72,2 kW of capacity, working with a temperature difference of 44/38,4 ºC. A glycol mixture is needed to avoid freezing of the fluid during the coldest days (30 % of glycol). This equipment can be bypassed when not needed. Usually, this device would be most used in summer, because excess heat from data rooms will not be needed to preheat neither the building or DHW. It transfers heat to the ambient and is located next to the evaporators of the heat pumps in the façade of the building [90].

Since this cooling system is used throughout the year, the only difference between seasons is the temperature set point for the preheating of water.

The main device in this system is the cooling machine (IK42) which transfers heat from the water coming from the data rooms to the glycol mixture that will transfer it either to the energy central, DHW or dry cooler [88] (see Figure 3.5). This cooling machine is a water-to-water heat pump with a capacity of 55,7 kW, working always in heating mode [85]. It also runs with R410a, but the mass is much lower than the ones in the main energy system, being 4,8 kg [85]. Heat from each of the data rooms (three in total) is extracted with 9 kW heat pumps. A full overview of the system can be seen in Figure 3.6.

Table 3.7 Summary of the characteristics of the MANTA T.48.P2.D.J7 cooling machine.

Cooling capacity	kW	55,7
Compressor power input	kW	15,5
Water temperature evaporator (in/out)	ºC	15/10
Water flow rate	m^3/h	9,57
Pressure drop	kPa	29
Water temperature condenser (in/out)	ºC	37/43
Gad circuits	nr.	2

It is equipped with two pairs of twin pumps, so “ice water” is transported to the data rooms and water can reach the dry cooler.

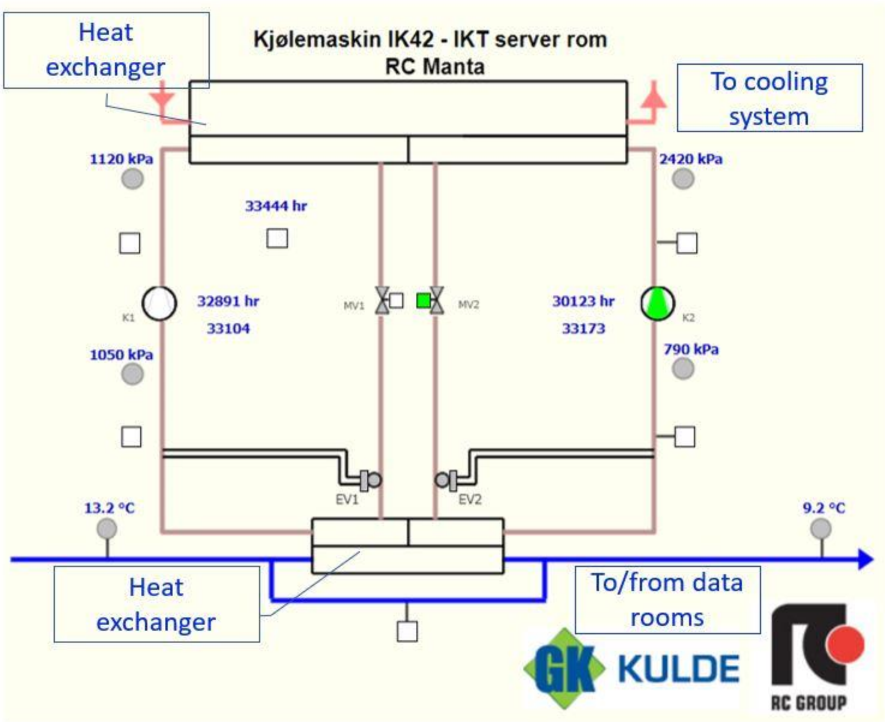


Figure 3.5 Cooling machine IK42 (modified from GK's servers) [88].

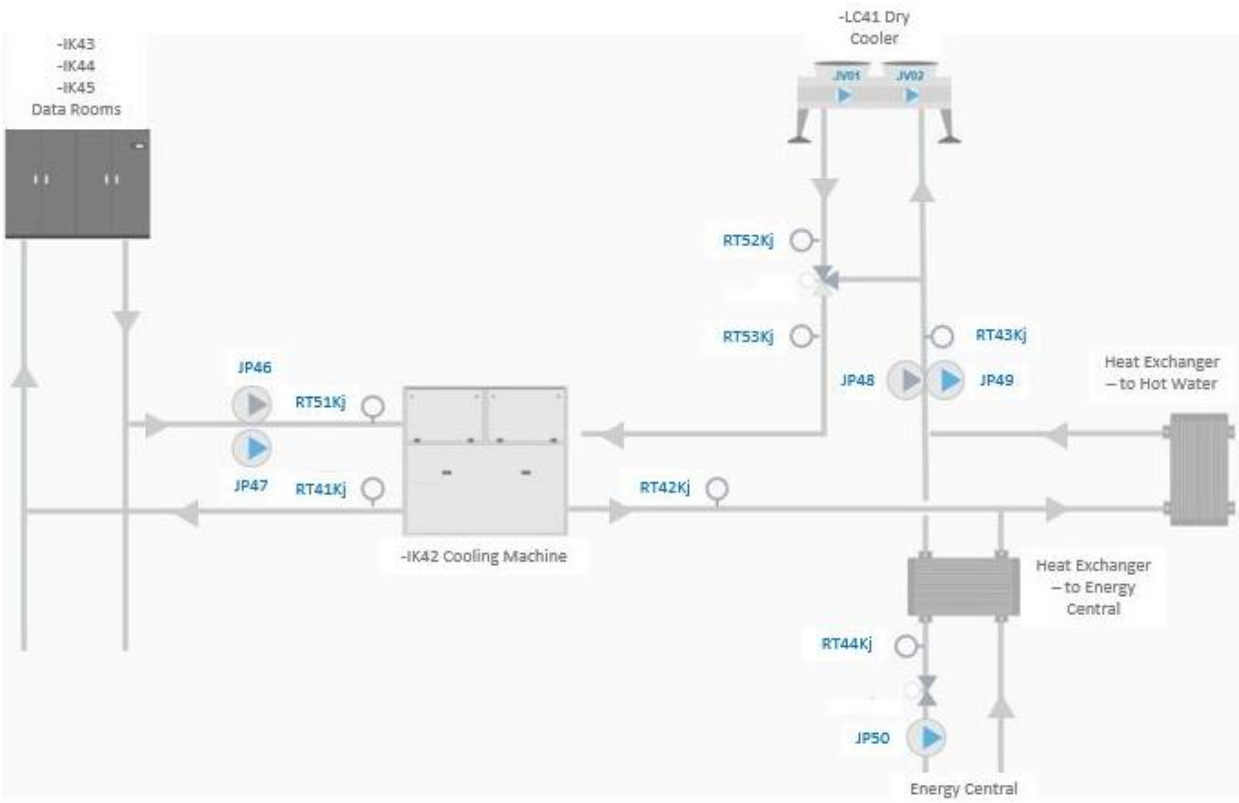


Figure 3.6 Cooling System of the building (modified from GK's servers) [88].

Domestic Hot Water System (DHW)

Preheat from the cooling system is used throughout the year to preheat DHW. In addition, one of the tanks has an electrical resistance (14 kW) which heats the water up to the desired temperature (70 ºC) [88]. In total there are three tanks of 400 L each. The distribution system for hot water is direct heating, which means that the mixture between hot water and cold water is done far from the user (55 ºC). This can lead to higher waiting periods for hot water to come out of the tap. The JP42 pump is controlled by the RT41Tv temperature sensor. To avoid pressure to increase when increasing the water temperature, an expansion vessel of 200 L is installed on the cold-water line before the first boiler [88]. The system can be seen in Figure 3.7.

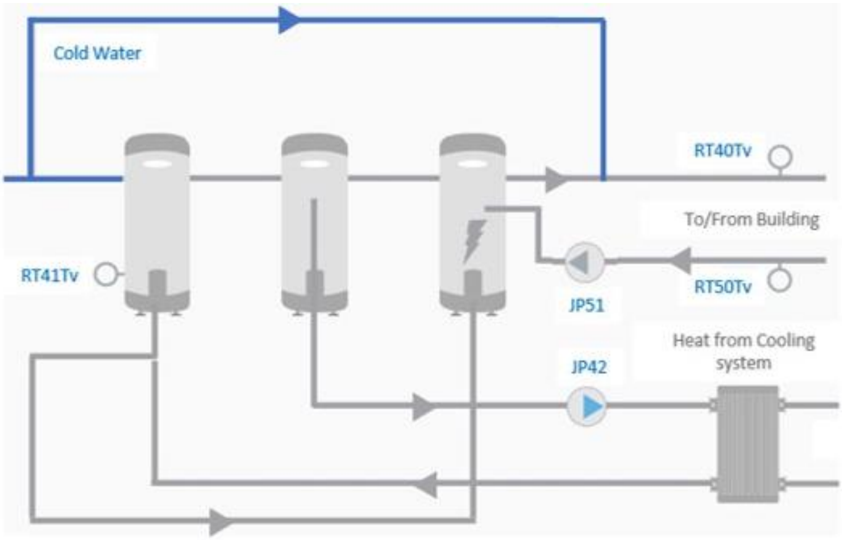


Figure 3.7 DHW system of the building (modified from GK's servers) [88].

3.4 Control Systems and Strategies

The system is controlled by measuring the temperature inside rooms, and actuating accordingly.

The control strategy followed in the building regarding the auxiliary electric boiler, according to [90] , uses the sensor RT41 to regulate the start/stop of the boiler. It has a setpoint with 2 ºC hysteresis, which enables the boiler to turn on if the temperature is more than 2 ºC below the setpoint temperature (this hysteresis avoids unnecessary switching on/off of the electric boiler).

Chapter ***4***

Methodology

In this chapter, the followed methodology will be explained. The gathering, processing and presentation of data will be presented.

4.1 Gathering of data from GK servers

Data from 2019 (January 1st to December 31st) will be gathered from several GK servers on the cloud. First, data recovered from “GK Cloud” from temperature sensors will be analysed (see Figure 4.1). It is to be noted that a short time slot has to be selected if hourly data[3] is to be seen. If the time slot chosen is too large, daily data will be shown and no thorough analysis could be performed.

Secondly, data from energy meters will be recovered from “ESight” (see Figure 4.2). In this platform, one can see energy meters from the building, from electricity meters to heat transfer calculations. This data will be particularly effective in the analysis of the yearly energy use and sizing of the energy system.

The meters analysed are as follows:

Table 4.1 Energy meters used for the analysis of the building [88].

System	**Energy meter**	**Time step**	**Unit**
Heat Pump	Total electricity consumed	Hourly	kW
	Total heat delivered	Hourly	kW
	Total cooling delivered	Hourly	kW
	Total heating and cooling	Hourly	kW
	COP	Hourly	-
Cooling machine	Electricity consumed	Hourly	kW
Electric Boiler	Electricity consumed	Hourly	kW
	Total heat delivered	Hourly	kW
Data Rooms	Total heat delivered to the building	Hourly	kW
	Total heat delivered to DHW	Hourly	kW
	Total electricity consumed	Hourly	kW
DHW	Total heat delivered by El. Tank	Hourly	kW
	Total electricity consumed	Hourly	kW
	Total electricity delivered to the distribution system	Hourly	kW

[3] The hourly data obtained is an average of the measurements saved every 10 min.

Snow Melting	Total heat delivered	Hourly	kW
Sun	Total electricity produced	Hourly	kW
Building A, B, C	Total heat delivered	Hourly	kW
	Heat delivered by each of the ventilation units	Hourly	kW
	Total cooling delivered	Hourly	kW
	Cooling delivered by each of the ventilation units	Hourly	kW
	Electricity delivered to each of the floors	Hourly	kW
General	Total electricity consumption	Hourly	kW

After recovering the necessary data, an Excel sheet (see Figure 4.3 and Figure 4.4) will be carried out, in order to see the different data from meters and sensors, and to be able to connect them together. This tool is also useful when drawing yearly graphics. The developed sheets are: hourly data and graphics, daily analysis, energy and electricity hourly use, heating and cooling degree hour calculations, sensor data and exergy analysis.

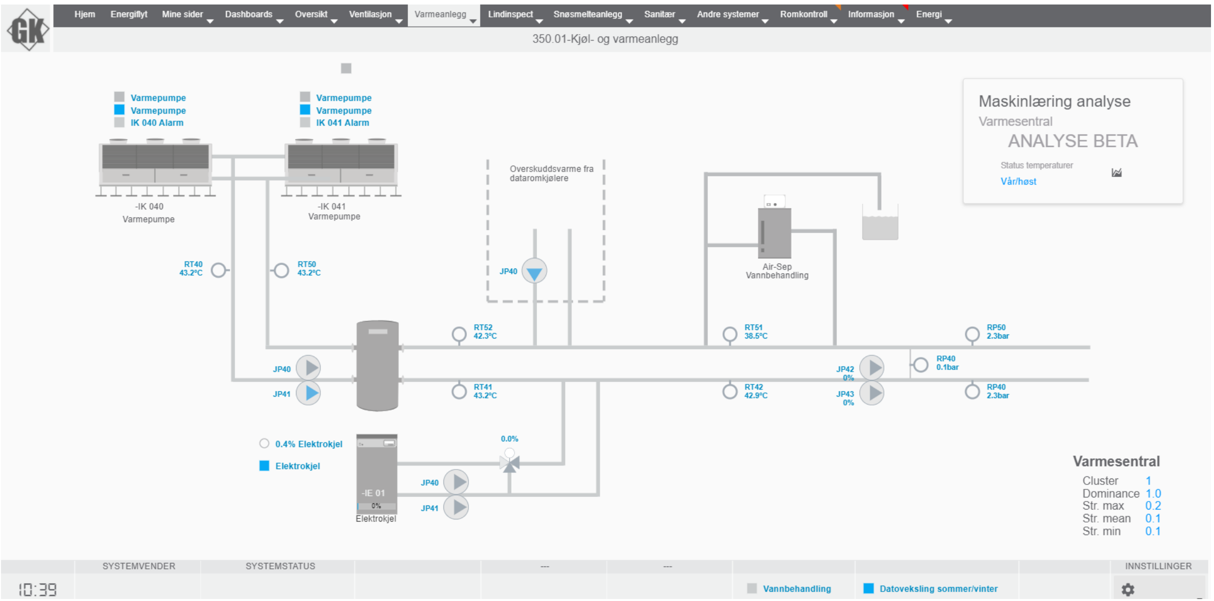


Figure 4.1 GK Cloud server on the cloud [88].

Figure 4.2 ESight server on the cloud [88].

	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R	S
1			HEAT PUMPS											ELECTRIC BOILER				DATA ROOM SPACE HEATING	
2-3			El. HP [kW]	Heat HP [kW]	Heat HP [%]	*Heat HP Sr. [kW]*	*Heat HP Sr. [%]*	Cooling HP [kW]	*Cooling Sr. [kW]*	Cooling (-) [kW]	Sum Heat + Cooling HP [kW]	*Sum Heat + Cooling HP Sr. [kW]*	COP HP	El. EB [kW]	Heat EB [kW]	*Heat EB Sr. [kW]*	*Heat EB Sr. [%]*	Heat from DR [kW]	*Heat from DR Sr. [kW]*
4	01/01/2019 0:00	0	2,8	29	51,8%	139	100,0%	0	207	0	29	139	10,36	0	0	121	100,0%	27	32
5	01/01/2019 1:00	1	8,3	27	50,0%	132	95,0%	0	197	0	27	129	3,25	0	0	96	79,3%	27	32
6	01/01/2019 2:00	2	7,7	8	22,2%	120	86,3%	0	181	0	8	120	1,04	0	0	95	78,5%	28	32
7	01/01/2019 3:00	3	9,8	22	45,8%	115	82,7%	0	179	0	22	113	2,24	0	0	95	78,5%	26	32
8	01/01/2019 4:00	4	8,5	30	57,7%	109	78,4%	0	173	0	30	104	3,53	0	0	95	78,5%	22	32
9	01/01/2019 5:00	5	9,7	19	38,0%	104	74,8%	0	166	0	19	104	1,96	0	0	95	78,5%	31	31
10	01/01/2019 6:00	6	4,4	29	54,7%	102	73,4%	0	163	0	29	101	6,59	0	0	94	77,7%	24	31
11	01/01/2019 7:00	7	12,5	33	54,1%	98	70,5%	5	162	-5	28	98	3,04	0	0	94	77,7%	28	31
12	01/01/2019 8:00	8	7,6	10	27,0%	98	70,5%	0	160	0	10	98	1,32	0	0	94	77,7%	27	31
13	01/01/2019 9:00	9	10,1	23	45,1%	98	70,5%	0	159	0	23	98	2,28	0	0	94	77,7%	28	31
14	01/01/2019 10:00	10	1,2	31	51,7%	98	70,5%	0	158	0	31	98	25,83	0	0	94	77,7%	29	31
15	01/01/2019 11:00	11	7,3	18	41,9%	97	69,8%	0	156	0	18	97	2,47	0	0	94	77,7%	25	31
16	01/01/2019 12:00	12	7,1	5	15,6%	97	69,8%	0	154	0	5	97	0,7	0	0	93	76,9%	27	31
17	01/01/2019 13:00	13	7,7	21	43,8%	97	69,8%	0	151	0	21	97	2,73	0	0	93	76,9%	27	31
18	01/01/2019 14:00	14	1,2	24	47,1%	97	69,8%	0	150	0	24	97	20	0	0	92	76,0%	27	31
19	01/01/2019 15:00	15	6,3	19	40,4%	97	69,8%	1	150	-1	18	96	3,17	0	0	92	76,0%	28	31
20	01/01/2019 16:00	16	9,2	21	44,7%	96	69,1%	0	150	0	21	96	2,28	0	0	87	71,9%	26	31
21	01/01/2019 17:00	17	10,5	11	31,4%	96	69,1%	0	146	0	11	96	1,05	0	0	82	67,8%	24	31
22	01/01/2019 18:00	18	3,2	35	53,8%	96	69,1%	0	143	0	35	96	10,94	0	0	81	66,9%	30	31
23	01/01/2019 19:00	19	7,9	14	35,9%	96	69,1%	0	142	0	14	96	1,77	0	0	81	66,9%	25	31
24	01/01/2019 20:00	20	9,2	19	40,4%	96	69,1%	0	142	0	19	96	2,07	0	0	81	66,9%	28	31
25	01/01/2019 21:00	21	10,1	26	51,0%	96	69,1%	0	142	0	26	95	2,57	0	0	80	66,1%	25	31
26	01/01/2019 22:00	22	10,5	24	48,0%	95	68,3%	0	141	0	24	95	2,29	0	0	80	66,1%	26	31
27	01/01/2019 23:00	23	10	32	52,5%	95	68,3%	0	140	0	32	95	3,2	0	0	79	65,3%	29	31
28	02/01/2019 0:00	24	5,2	31	56,4%	95	68,3%	0	139	0	31	94	5,96	0	0	78	64,5%	24	30
29	02/01/2019 1:00	25	8	35	54,7%	94	67,6%	0	138	0	35	94	4,37	0	0	78	64,5%	29	30
30	02/01/2019 2:00	26	10,8	21	44,7%	94	67,6%	0	137	0	21	94	1,94	0	0	78	64,5%	26	30
31	02/01/2019 3:00	27	12,9	18	40,9%	94	67,6%	0	137	0	18	93	1,4	0	0	78	64,5%	26	30
32	02/01/2019 4:00	28	10,7	32	54,2%	93	66,9%	0	137	0	32	93	2,99	0	0	77	63,6%	27	30
33	02/01/2019 5:00	29	16,1	36	57,1%	93	66,9%	0	135	0	36	93	2,24	0	0	76	62,8%	27	30
34	02/01/2019 6:00	30	17,1	46	62,2%	93	66,9%	0	135	0	46	92	2,69	0	0	76	62,8%	28	30
35	02/01/2019 7:00	31	18,9	86	78,2%	92	66,2%	0	135	0	86	92	4,55	0	0	75	62,0%	24	30
36	02/01/2019 8:00	32	17	40	58,8%	92	66,2%	0	133	0	40	92	2,35	0	0	75	62,0%	28	30
37	02/01/2019 9:00	33	18,2	47	64,4%	92	66,2%	0	133	0	47	91	2,58	0	0	74	61,2%	26	30
38	02/01/2019 10:00	34	6,9	69	75,8%	91	65,5%	0	132	0	69	91	10	0	0	73	60,3%	22	30
39	02/01/2019 11:00	35	11,6	45	62,5%	91	65,5%	1	132	-1	44	91	3,97	0	0	73	60,3%	27	30

Figure 4.3 Part of the excel sheet developed for the analysis (1).

	A	D	E	F	G	H	I	J	K	L	M	N
1		BUILDING A					BUILDING B					
2	From Timestamp	Cooling vent. 36002 [kW]	Heatin g vent. 36002 [kW]	Total Heating A [kW]	Total Cooling A [kW]		Cooling cantine [kW]	Heating cantine - Fancoil [kW]	Cooling vent. 36003 [kW]	Heating vent. 36003 [kW]	Cooling Vent. 36004 [kW]	Heating Vent. 36004 [kW]
2843	29/04/2019 8:00	0	0	0	0		0	0	0	0	1E-05	1E-05
2844	29/04/2019 9:00	0	0	0	0		0	0	0	0	0	0
2845	29/04/2019 10:00	0	0	0	0		0	0	0	0	1E-05	1E-05
2846	29/04/2019 11:00	0	0	0	1		0	0	1	0	0	0
2847	29/04/2019 12:00	5	0	0	11		0	0	10	0	1E-05	1E-05
2848	29/04/2019 13:00	10	0	0	20		0	0	16	0	0	0
2849	29/04/2019 14:00	14	0	0	26		0	0	17	0	1E-05	1E-05
2850	29/04/2019 15:00	15	0	0	30		0	0	15	0	0	0
2851	29/04/2019 16:00	14	0	0	27		0	0	13	0	1E-05	1E-05
2852	29/04/2019 17:00	10	0	0	21		0	0	11	0	0	0
2853	29/04/2019 18:00	8	0	0	17		0	0	7	0	1E-05	1E-05
2854	29/04/2019 19:00	9	0	0	17		0	0	9	0	0	0
2855	29/04/2019 20:00	4	0	0	9		0	0	4	0	1E-05	1E-05
2856	29/04/2019 21:00	4	0	0	7		0	0	1	0	0	0
2857	29/04/2019 22:00	1	0	0	1		0	0	0	0	1E-05	1E-05
2858	29/04/2019 23:00	0	0	0	0		0	0	0	0	0	0
2859	30/04/2019 0:00	0	0	0	0		0	0	0	0	1E-05	1E-05
2860	30/04/2019 1:00	0	0	0	0		0	0	0	0	0	0
2861	30/04/2019 2:00	0	0	0	0		0	0	0	0	1E-05	1E-05
2862	30/04/2019 3:00	0	0	0	0		0	0	0	0	0	0
2863	30/04/2019 4:00	0	0	0	0		0	0	0	0	1E-05	1E-05
2864	30/04/2019 5:00	0	0	0	0		0	0	0	0	0	0
2865	30/04/2019 6:00	0	0	0	0		0	0	0	0	1E-05	1E-05
2866	30/04/2019 7:00	0	0	0	0		0	0	0	0	0	0
2867	30/04/2019 8:00	0	0	0	0		0	0	0	0	1E-05	1E-05
2868	30/04/2019 9:00	0	0	0	1		0	0	0	0	0	0
2869	30/04/2019 10:00	2	0	0	5		0	0	0	0	1E-05	1E-05
2870	30/04/2019 11:00	7	0	0	12		0	0	2	0	0	0
2871	30/04/2019 12:00	9	0	0	17		0	0	7	0	1E-05	1E-05
2872	30/04/2019 13:00	13	0	0	24		0	0	11	0	0	0
2873	30/04/2019 14:00	19	0	0	31		0	0	18	0	1E-05	1E-05

	O	P	Q	R	S	T	U	V	W	X	Y	Z	AA	A	AC	AD	AE
1						BUILDING C											
2	Cooling garage 36008 [kW]	Heating garage 36008 [kW]	Total Heating B [kW]	Total Cooling B [kW]		Cooling vent. 36005 [kW]	Heating Vent. 36005 [kW]	Cooling vent. 36006 [kW]	Heating Vent. 36006 [kW]	Cooling garage 36009 [kW]	Heating garage 36009 [kW]	Total Heating C [kW]	Total Cooling C [kW]		Heating Snowmel ting [kW]	Cooling from HP [kW]	Heating from HP [kW]
2843	0	0	0,00001	1E-05		0	0	0	0	1E-05	0	0	1E-05		0	0	7
2844	1E-05	1E-05	0,00001	1E-05		2	0	5	0	0	1E-05	1E-05	7		0	34	3
2845	0	0	0,00001	1E-05		7	0	10	0	1E-05	0	0	17		0	21	0
2846	1E-05	1E-05	0,00001	1		9	0	12	0	0	1E-05	1E-05	21		0	19	0
2847	0	0	0,00001	10		15	0	17	0	1E-05	0	0	32		0	34	0
2848	1E-05	1E-05	0,00001	16		19	0	20	0	0	1E-05	1E-05	39		0	58	0
2849	0	0	0,00001	17		25	0	23	0	1E-05	0	0	48		0	85	0
2850	1E-05	1E-05	0,00001	15		26	0	27	0	0	1E-05	1E-05	53		0	94	0
2851	0	0	0,00001	13		24	0	23	0	1E-05	0	0	47		0	90	0
2852	1E-05	1E-05	0,00001	11		18	0	19	0	0	1E-05	1E-05	37		0	83	0
2853	0	0	0,00001	7		15	0	13	0	1E-05	0	0	28		0	61	0
2854	1E-05	1E-05	0,00001	9		12	0	12	0	0	1E-05	1E-05	24		0	53	0
2855	0	0	0,00001	4		11	0	9	0	1E-05	0	0	20		0	43	0
2856	1E-05	1E-05	0,00001	1		9	0	8	0	0	1E-05	1E-05	17		0	29	0
2857	0	0	0,00001	1E-05		7	0	7	0	1E-05	0	0	14		0	19	0
2858	1E-05	1E-05	0,00001	1E-05		4	0	5	0	0	1E-05	1E-05	9		0	9	1
2859	0	0	0,00001	1E-05		4	0	4	0	1E-05	0	0	8		0	7	0
2860	1E-05	1E-05	0,00001	1E-05		3	0	4	0	0	1E-05	1E-05	7		0	5	2
2861	0	0	0,00001	1E-05		1	0	1	0	1E-05	0	0	2		0	9	3
2862	1E-05	1E-05	0,00001	1E-05		2	0	3	0	0	1E-05	1E-05	5		0	6	2
2863	0	0	0,00001	1E-05		1	0	1	0	1E-05	0	0	2		0	0	4
2864	1E-05	1E-05	0,00001	1E-05		0	0	2	0	0	1E-05	1E-05	2		0	1	6
2865	0	0	0,00001	1E-05		0	0	1	0	1E-05	0	0	1		0	4	4
2866	1E-05	1E-05	0,00001	1E-05		1	0	2	0	0	1E-05	1E-05	3		0	11	1
2867	0	0	0,00001	1E-05		3	0	2	0	1E-05	0	0	5		0	9	1
2868	1E-05	1E-05	0,00001	1E-05		6	0	5	0	0	1E-05	1E-05	11		0	8	1
2869	0	0	0,00001	1E-05		10	0	10	0	1E-05	0	0	20		0	19	0
2870	1E-05	1E-05	0,00001	2		13	0	14	0	0	1E-05	1E-05	27		0	31	0
2871	0	0	0,00001	7		20	0	17	0	1E-05	0	0	37		0	54	0
2872	1E-05	1E-05	0,00001	11		21	0	20	0	0	1E-05	1E-05	41		0	66	0
2873	0	0	0,00001	18		25	0	24	0	1E-05	0	0	49		0	84	0

Figure 4.4 Part of the excel sheet developed for the analysis (2).

4.2 Exergy Analysis

That one may understand better the real efficiency of the main components, an exergy analysis will be performed. Following the equations Eq. 2.8 to Eq. 2.19, the properties of each of inlet/outlet of each component will be calculated, as well as the destruction of exergy and the energetic and exergetic efficiency. Following the principle in Figure 4.5:

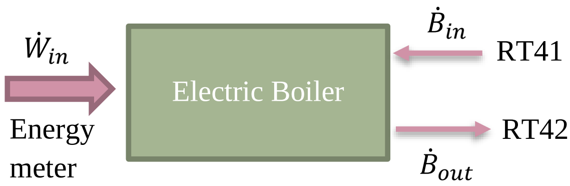


Figure 4.5 Example of the procedure followed in the energy/exergy analysis.

Exergy efficiency and exergy destruction, will give a better insight of how the components are working, and will provide information about how to increase its performance.

4.3 Energy Flows

In order to see how electricity and energy are distributed in the building, an analysis of the energy flows will be carried out. In this analysis, all known electric loads and energy loads are taken into account. Unfortunately, some loads are unknown, but this analysis will help in the determination of these unknown loads. To create the energy flows, data from the meters was used, as well as the Excel sheet developed previously.

In Figure 4.6 one can see the electricity flow of the building, on which the distribution of the electrical loads around the building is shown. Meters from around the building are taken into account, separated by building and purpose. Information from the meters show that all electricity produced in the building is used in the same building. When talking about “Lighting and Equipment” one refers to the electricity consumed in each of the buildings, not related to heating/cooling of spaces, ventilation or other matters such as electric car chargers (however includes elevators etc.).

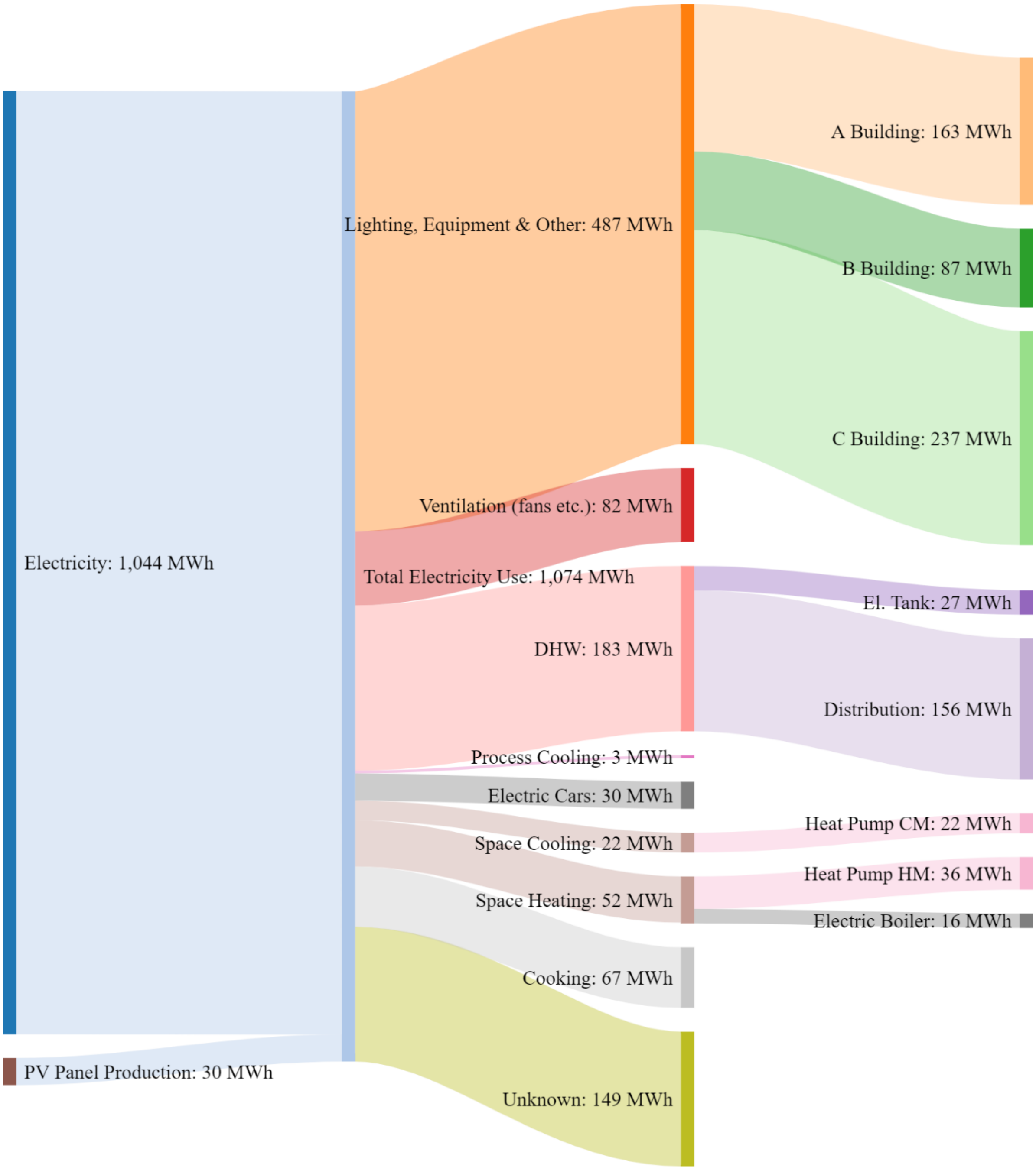


Figure 4.6 Electricity flow of the building. Image built using http://sankeymatic.com/build/ [88].

In Miljøhuset there are two main heating sources, heat recovered from data rooms and heat pumps, as seen in Figure 4.7. apart from these sources, an electric boiler is installed as an auxiliary boiler and electric heated tanks are installed to heat up DHW, as explained in Section 3.3.1. However, other heat sources such as solar gains or heat produced by people, are not measured even though they actively affect the change in the indoor temperature of the building.

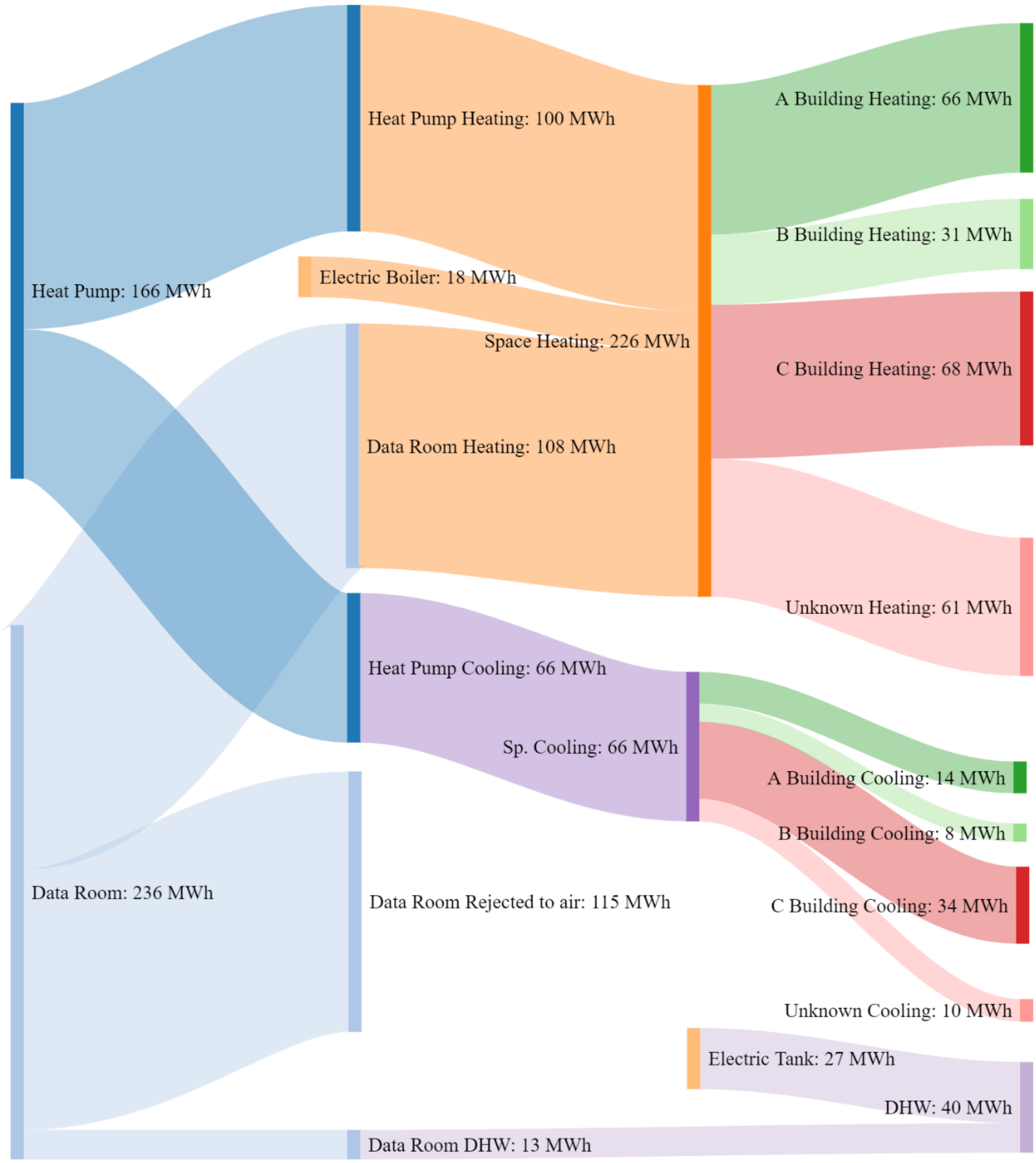


Figure 4.7 Energy flow of the building. Image built using http://sankeymatic.com/build/ [88].

Chapter ***5***

Results

In this chapter, data gathered from GK's servers (GK-Cloud, eSight and GK Sentraldrift) from the year 2019 will be presented. Graphics and tables will show the data, and some general calculations will be made to better illustrate the results. Graphics on energy and electricity loads will be presented. Results of the exergy analysis will be given.

5.1 Annual analysis – Heating and Cooling loads

Using GK's servers, data from the year 2019 was gathered. Due to the Covid-19 pandemic, the building was not used as normal in 2020, so data from 2020 will not be taken into account in this project. From the analysis performed to the data gathered from GK servers, some conclusions will be made.

It must be noted, that from the 28th of August to the 10th of September, no data was recovered. The reason for this is unknown, but it may be due to a failure of the system.

In Figure 5.1 the total heating/cooling load duration curve from 2019 can be seen. Since heat pumps cover 100 % of the cooling loads, the total HP cooling and the space cooling coincide.

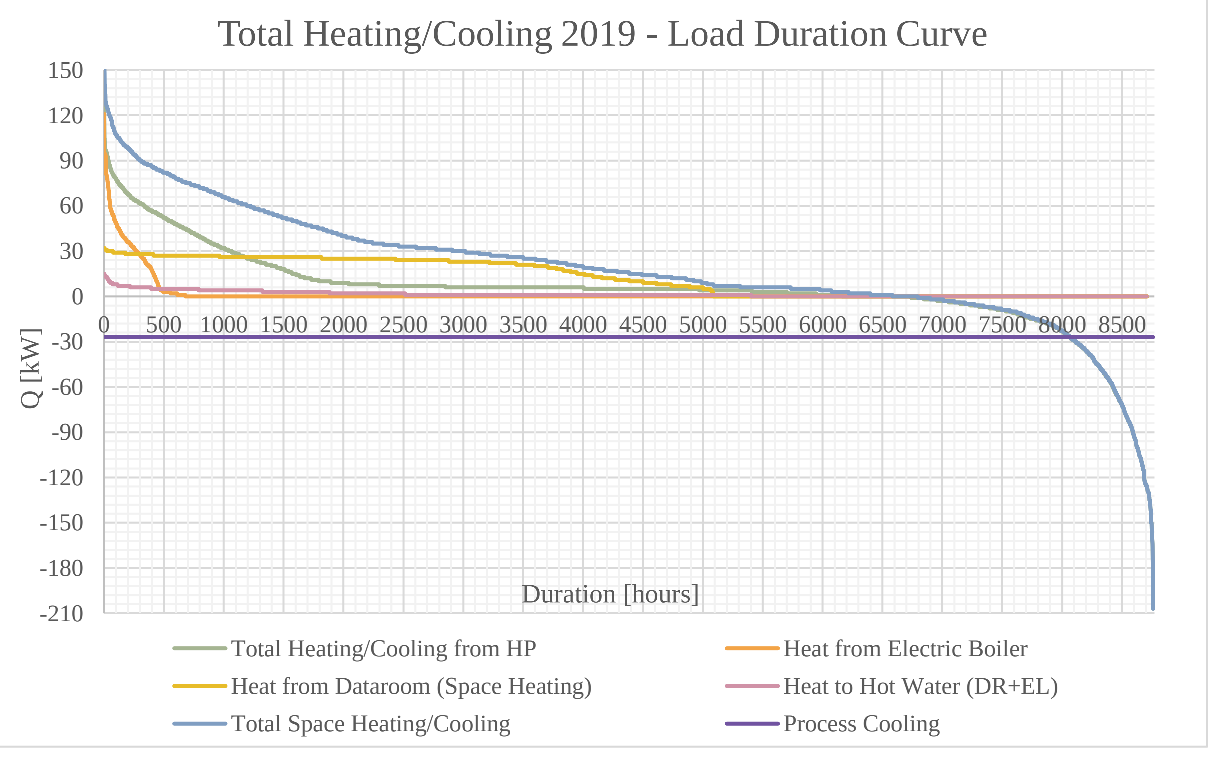


Figure 5.1 Total heating and cooling load duration curve from 2019 [88].

Heat from data rooms is more or less constant throughout the heating season and covers 48 % of the total space heating load of the building (see Figure 5.1).

Figure 5.2 shows the space total heating and cooling loads from all the energy providers in the building (this is, heat pumps, data rooms and electric boiler). In green can also be seen the hours when the outside temperature was lower than -15 ºC, which would be of used later in the discussion.

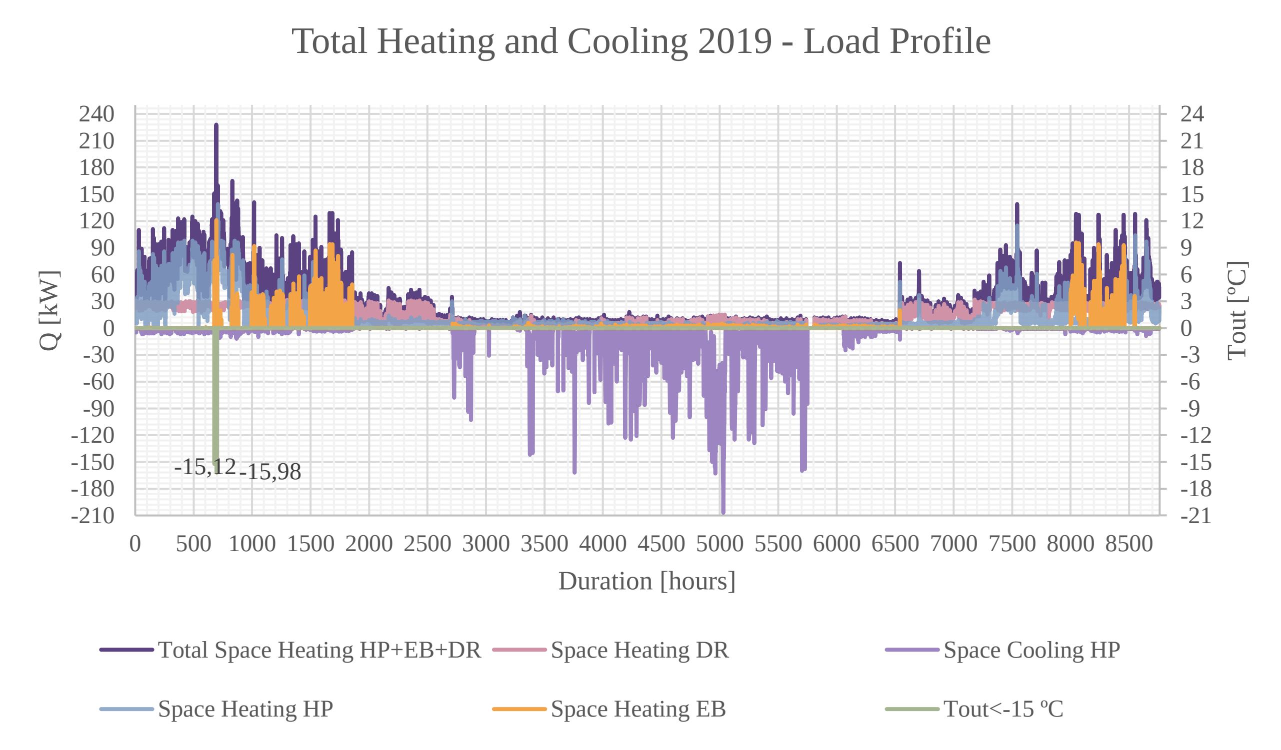


Figure 5.2 Total heating and cooling load profile from 2019. Not shown in the graphic: heat delivered to water from data rooms and heat delivered to the snow melting system (for the clarity of the graph) [88].

From Figure 5.2 can be calculated, that about 44 % of the heat delivered to the building corresponds to the heat pump units. However, as will be seen later in Figure 5.6, the electricity consumption of the heat pumps is only 5 % of the total electricity consumption of the building. The maximum heating load is 139 kW with an average of 14 kW during the 7169 h that the heat pump was on heating mode. The maximum cooling loads is 207 kW with an average of 24 kW during the 3518 h that the heat pump was in cooling mode[4]. Although heat pumps work longer on heating mode (7169 h in heating mode versus 3518 h in cooling mode), the maximum cooling load is larger than the maximum heating load (207 kW of cooling vs. 139 kW of heating).

The electric boiler was on for 485 h in 2019, with a maximum peak load of 121 kW. The average heat load was 36 kW.

The snow melting system used 18 MWh (or 144,78 kWh/m^2) in 2019 from heat produced in the building (an 8,7 % of the total 207 952 kWh).

[4] The heat pumps work multiple hours a year with a capacity lower than 10 kW, both in heating and cooling mode. If only the hours that the heat pumps provide more that 10 kW are considered, the average heating load would be 48,5 kW and the average cooling load 38,5 kW. Nevertheless, these numbers will not be taken into account in this thesis.

In order to cover the hot water demand, 2,6 kWh/m^2 of heat is provided, of which 36 % is obtained from data rooms.

In Figure 5.3 the distribution of the heat recovered from the data rooms can be seen. In total, 108 250 kWh were used in space heating, 115 400 kWh were rejected to air and 12 900 kWh were used in preheating water. This is respectively, 45,76 %, 48,78 % and 5,45 %. Rejected heat to air is not measured by the energy meters, so it was estimated by assuming that the data room heat extraction was constant throughout the year.

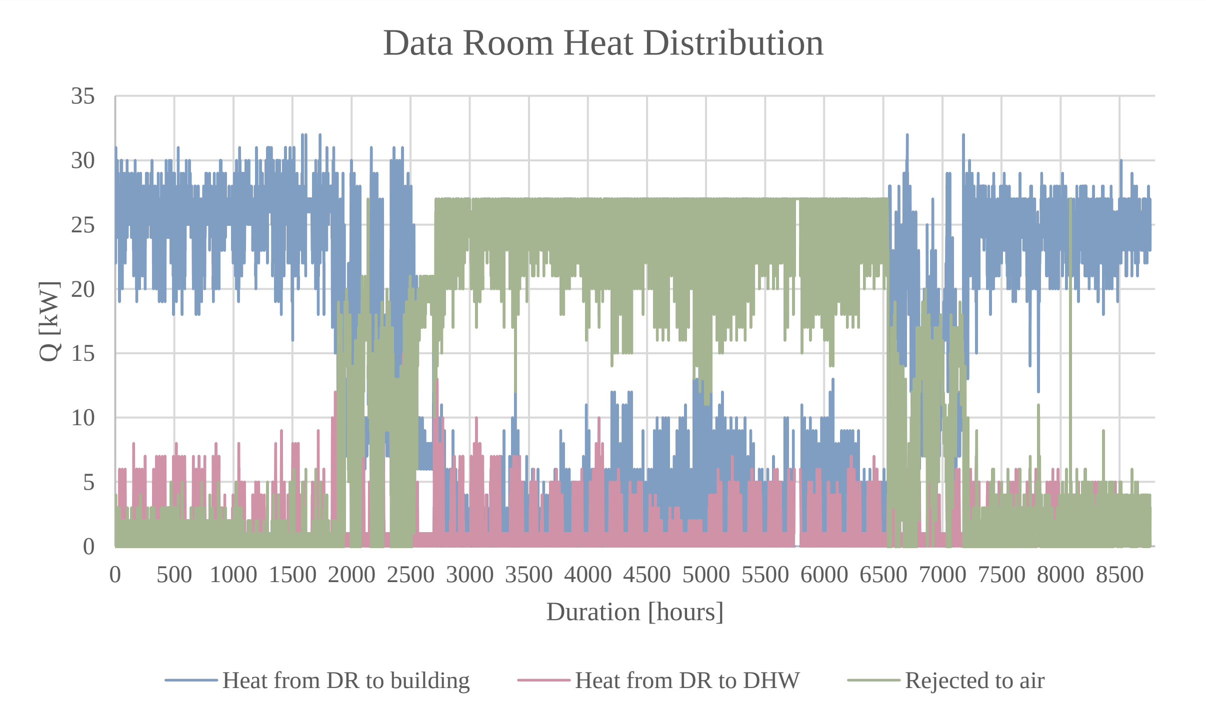


Figure 5.3 Data Room heat distribution between building, DHW and rejected to air [88].

Since the cooling machine used heat extracted from the data coolers, its maximum load is limited by them. There are three data coolers with a capacity of 9 kW each, so a maximum of 27 kW can be extracted. There is no data on the heat extracted from the data rooms, but looking at Figure 5.3 one can see that it is around 27 kW. However, the cooling machine's capacity is 55,7 kW, meaning that the machine works at 48,47 % capacity.

Using equations Eq. 2.29 and Eq. 2.30, the Energy Coverage Ratio (α) and the Power Coverage Ratio (β) will be calculated:

$$\alpha_{HP-heating} = \frac{Q_{HP-heating}}{Q_{TOTAL}} = \frac{99\ 709\ kWh}{225\ 493\ kWh} = 0{,}4421$$

$$\alpha_{HP-cooling} = \frac{Q_{HP-cooling}}{Q_{TOTAL}} = \frac{65\ 216\ kWh}{65\ 216\ kWh} = 1$$

$$\alpha_{DR} = \frac{Q_{DR}}{Q_{TOTAL}} = \frac{108\,243\ kWh}{225\,493\ kWh} = 0{,}48$$

$$\alpha_{EB} = \frac{Q_{EB}}{Q_{TOTAL}} = \frac{17\,541\ kWh}{225\,493\ kWh} = 0{,}077$$

These parameters and Figure 5.4 show what was explained earlier, that the heat pump system covers 44 % of the heating loads, the data rooms 48 % and the electric boiler 8%.

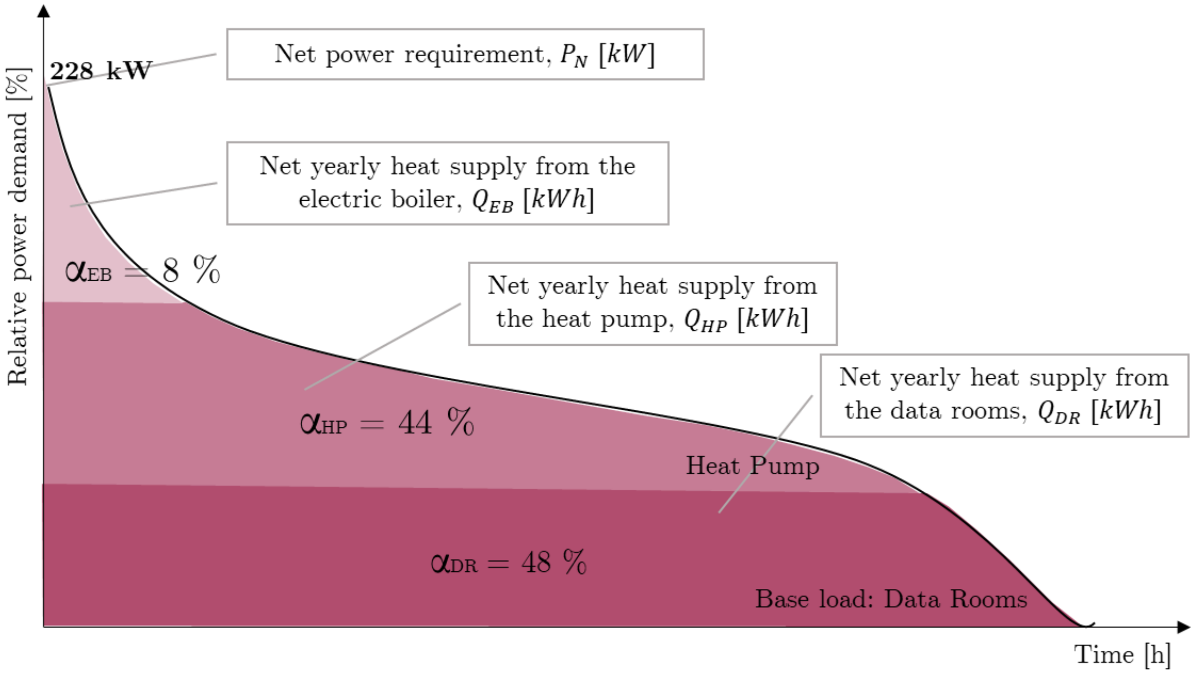


Figure 5.4 Load duration curve of the building.

It would be of interest to know how much of the power was covered by each of the systems. Since the design parameters are not available, the calculations have been done with the maximum load of each of the systems.

$$\beta_{HP-heating} = \frac{P_{HP-heating}}{P_{TOTAL}} = \frac{139\ kW}{228\ kW} = 0{,}6096$$

$$\beta_{HP-cooling} = \frac{P_{HP-heating}}{P_{TOTAL}} = \frac{207\ kW}{207\ kW} = 1$$

$$\beta_{DR} = \frac{P_{DR}}{P_{TOTAL}} = \frac{32\ kW}{228\ kW} = 0{,}1403$$

$$\beta_{EB} = \frac{P_{EB}}{P_{TOTAL}} = \frac{121\ kW}{228\ kW} = 0{,}5307$$

However, these parameters do not show the real distribution, because the day that the electric boiler had its peak load, the heat pumps were not at full load. Consequently, the above calculated parameters cannot be used to conclude that the electric boiler is designed to cover 53 % of the peak loads, for example.

It is also of interest to know the Equivalent Operation Time (τ) defined in Eq. 2.31, since it would give an insight of how much the systems were used during the year 2019:

$$\tau_{heating} = \frac{Q_{HP-heating}}{P_{HP-dim-heating}} = \frac{225\,493\ kWh}{2 * 276\ kW} = 408{,}5\ h$$

$$\tau_{cooling} = \frac{Q_{HP-cooling}}{P_{HP-dim-cooling}} = \frac{65\,216\ kWh}{2 * 226\ kW} = 144{,}28\ h$$

$$\tau_{cooling\ machine} = \frac{Q_{cooling\ machine}}{P_{CM-dim}} = \frac{108\,243 + 12\,870 + 115\,407\ kWh}{55{,}7\ kW} = 4\,246{,}3\ h$$

These values are very low compared to the standard office building ones, which are around 1 500 – 3 000 h for heat pump systems [58].

To sum up, the most relevant data will be gathered in Table 5.1, Table 5.2, Table 5.3 and Table 5.4.

Table 5.1 Space heating loads by source [88].

Total heating from HP	[MWh]	99,7	[kWh/m^2]	7,3
Total heating from DR	[MWh]	108,25	[kWh/m^2]	7,92
Total heating from EB	[MWh]	17,55	[kWh/m^2]	1,28
Total space heating	[MWh]	225,5	[kWh/m^2]	16,51

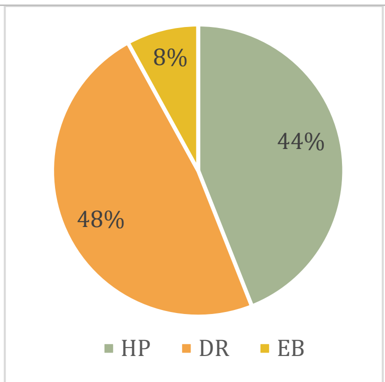


Table 5.2 Space cooling loads by source [88].

Total space cooling	[MWh]	65,22	[kWh/m^2]	4,77	100 %

Table 5.3 Heat delivery from Data Rooms by sink [88].

Total heat from DR to space heating	[MWh]	108,25	[kWh/m^2]	7,92
Total heat from DR to DHW	[MWh]	12,9	[kWh/m^2]	0,94
Total heat from DR rejected to air	[MWh]	115,4	[kWh/m^2]	8,45
Total heat recovered from DR	[MWh]	236,5	[kWh/m^2]	17,33

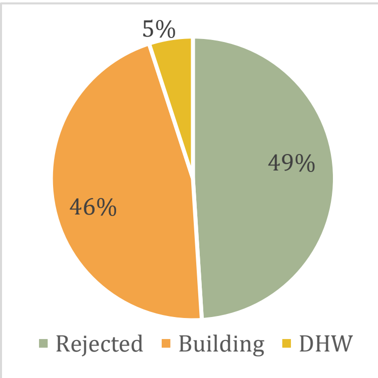
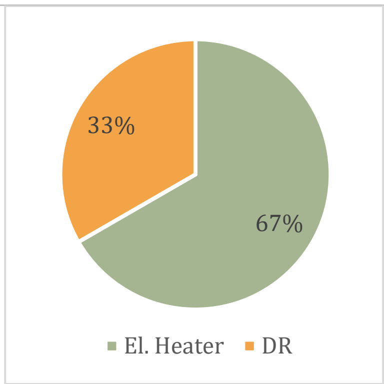


Table 5.4 Heat delivered to DHW by source [88].

Total heat to DHW from DR	[MWh]	12,9	[kWh/m^2]	0,94
Total heat to DHW from El. Boiler	[MWh]	26,64	[kWh/m^2]	1,95
Total heat to DHW	[MWh]	39,5	[kWh/m^2]	2,89



In Figure 5.5, a comparison between the simulated data obtained from [89] and the measured data is given. It can be appreciated that the simulations do not represent the reality of the building's needs.

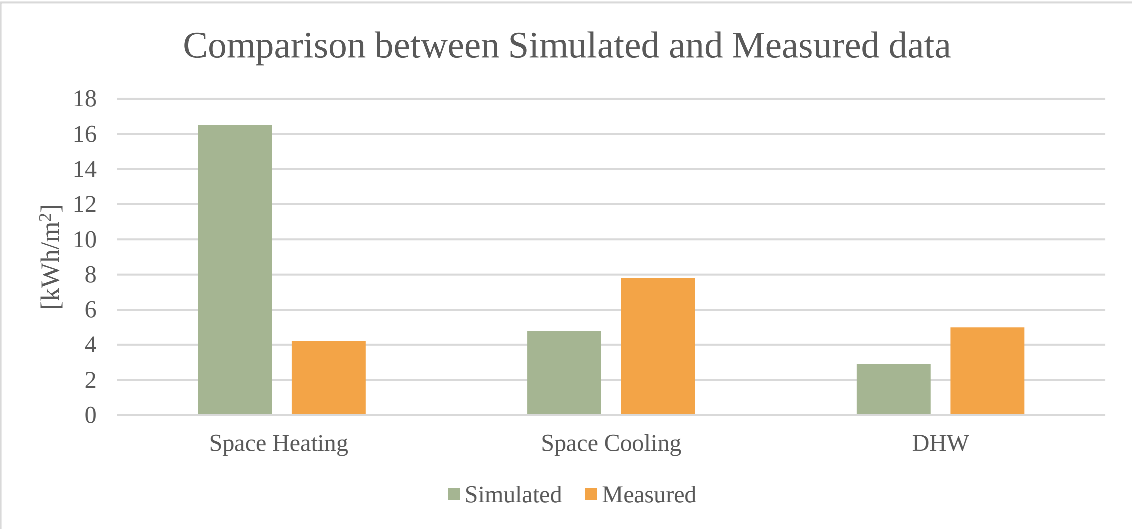


Figure 5.5 Comparison between simulated and measured data [89] [88] respectively.

5.2 Annual analysis: Electricity loads

Figure 5.6 shows the annual electricity loads from 2019. It can also be easily seen that buildings C, B and A, in order, demand the most electricity, which accounts for lightings, equipment, kitchen and canteen, electrical car charging stations, snow melting system and others. To better see the different loads, the graphic will be divided in two: building loads in Figure 5.7 and loads related to heating/cooling of spaces and DHW in Figure 5.8. Solar electricity production covers roughly 3 % of the electricity consumed in the building with a production of around 30 280 kWh in 2019 which is close to the expected value (around 40 000 kWh annually). The lower electricity production can be due to snow covering, lower performance or maintenance periods.

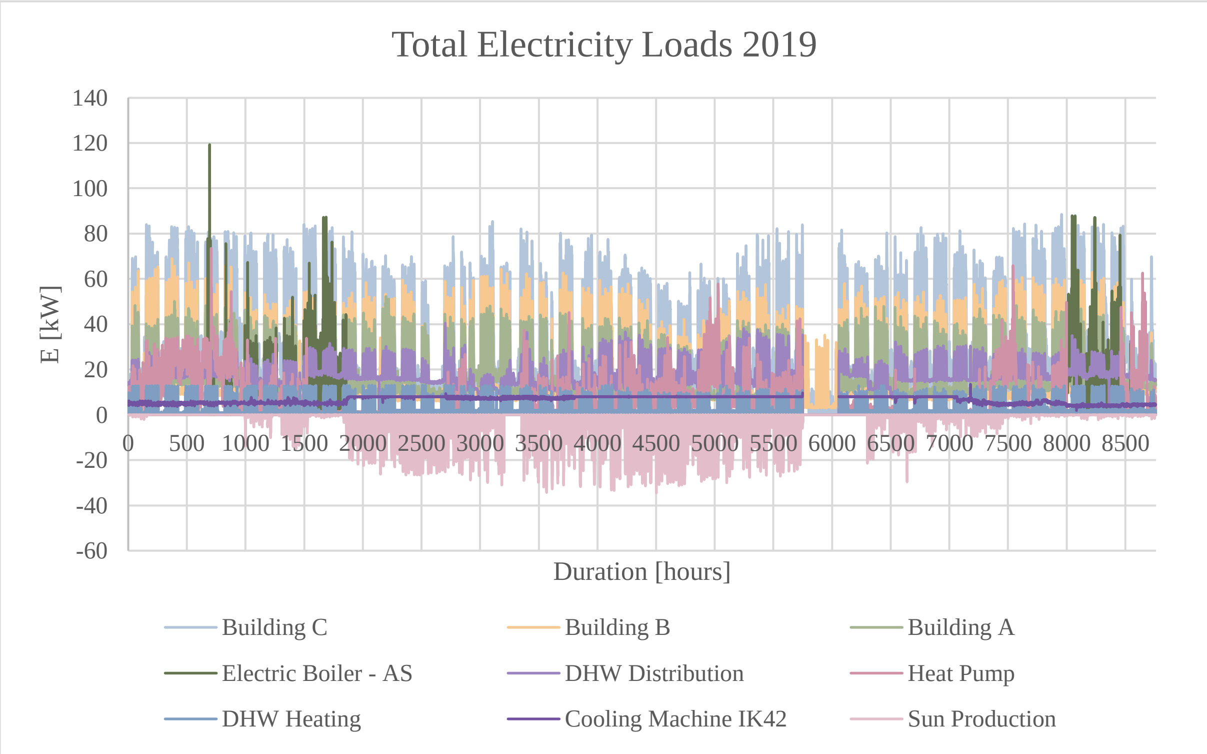


Figure 5.6 Annual electricity loads from 2019 [88].

In Figure 5.7 one can see that building C has a substantially higher electricity load than the other two buildings, almost being double the load of building A. The source for this difference is unknown, since there is not enough information about the inside of the building. It can be also seen the weekly pattern in the electricity use, with lower consumption during the weekends.

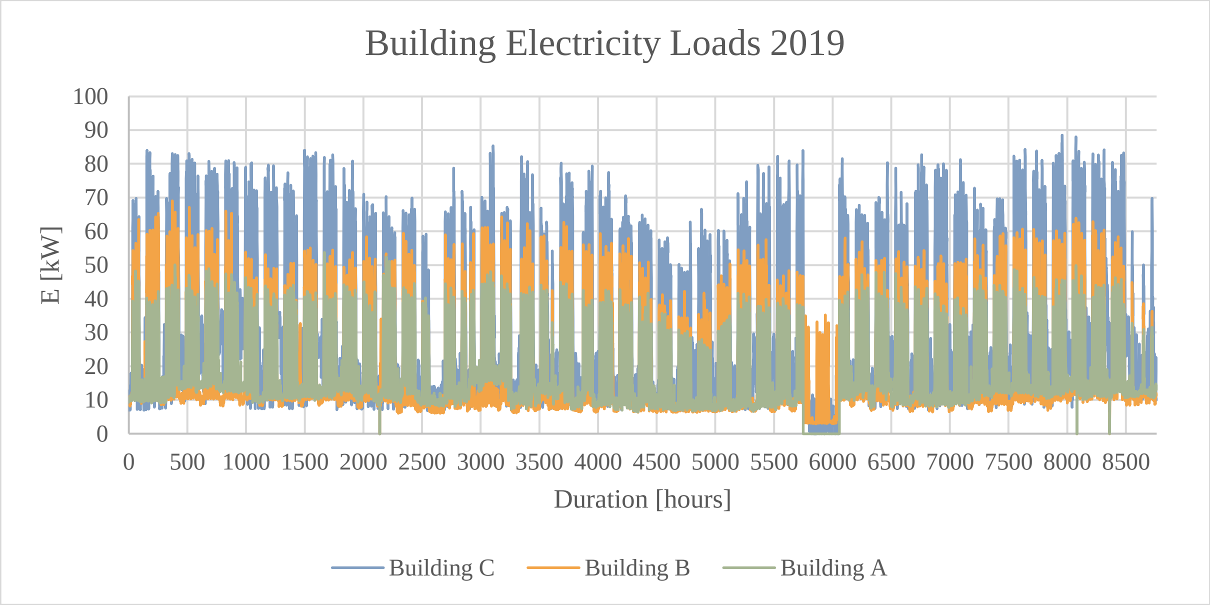


Figure 5.7 Electricity loads from 2019 from the buildings A, B and C [88].

The electricity consumption of the electrical boiler, is one of the highest when it is turned on, as shown in Figure 5.6 and Figure 5.8. The electrical consumption of the DHW distribution system is surprisingly high, being six times as high as the electricity used to heat it in the tank.

The electrical consumption of the cooling machine IK42, as can be seen in Figure 5.8, is more or less constant throughout the year, which is consistent with the heat transfer data.

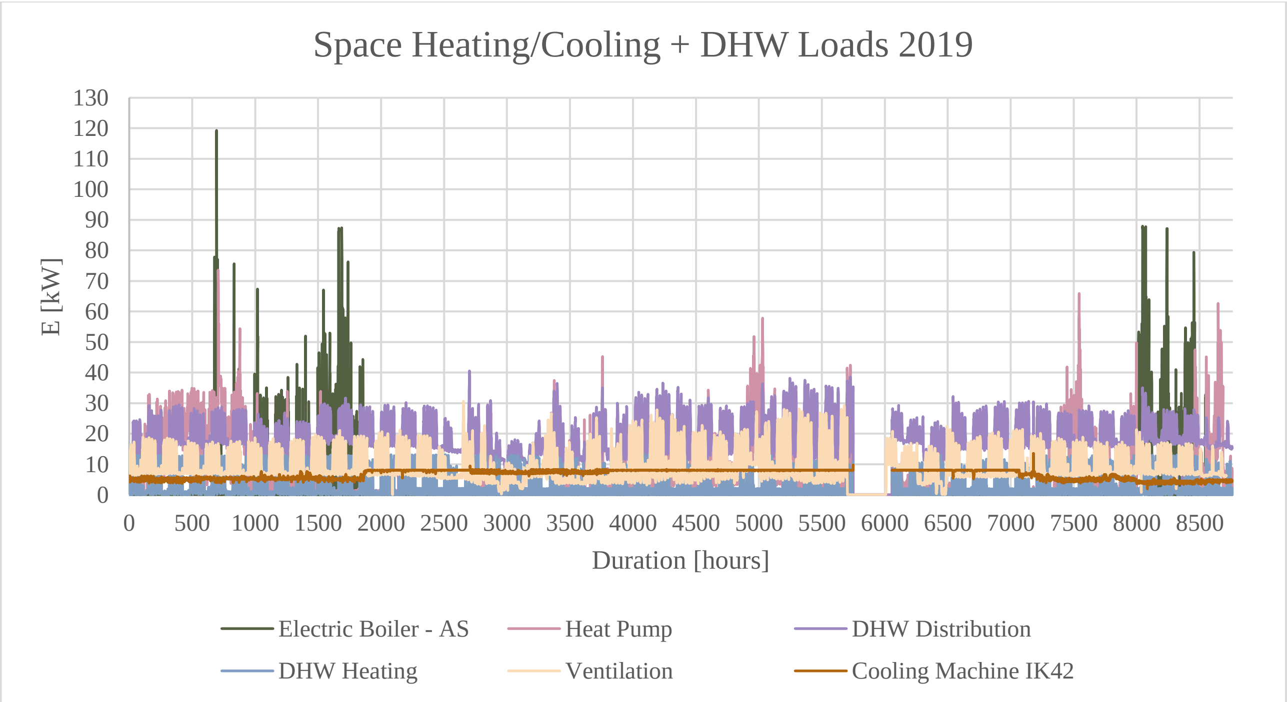


Figure 5.8 Electricity loads from space heating/cooling and DHW from 2019 [88].

5.3 Exergy analysis

While performing the energy and exergy analysis, several limitations where encountered. First, not all sensors have a storing system installed. Therefore, a live value can be seen but no annual data can be recovered. This happened to the inlet and outlet temperature sensors located at the air inlet of the heat pumps. As presented in Section 2.2.2, this data is essential in order to do the energy and exergy balances. Consequently, no proper analysis of these components could be performed.

Secondly, temperature sensors and energy meters are not completely synchronized. When performing an hourly analysis, it is of great importance that the meters and the sensors are synchronized, so one can know at each time step how much energy was given to the compressors, for example. This resulted in a very unprecise analysis and illogical results, such as negative exergy destruction or negative efficiencies. On top of this, the stored data is the average of the measurements taken in a 1-hour interval. Thus, fluctuations in the measurements are not reflected, leading to average values that hide these fluctuations and which are unprecise. It leads to many errors, because many times the average from two sensors is closer than the real values were at the exact moment. For this reason, there is much variation in the temperatures, summed up to the uncertainty of the sensors.

Lastly, better sensor sensibility would make any further analysis more accurate. Therefore, it was concluded that the exergy analysis would not give enough relevant information about the components.

5.4 Summary of the Analysis Performed Based on Data from 2019

In order to better understand the data presented above, and to summarize, the most important points will be listed below:

1. The maximum heating load from the heat pump was 139 kW in 2019.
2. The maximum cooling demand was 207 kW in 2019.
3. The maximum heat delivered from the electric boiler was 121 kW in 2019.
4. The maximum heating demand from the building was 228 kW in 2019.
5. The maximum heat demand from the DHW was 28 kW in 2019.
6. No effective exergy analysis could be performed.

7. The average heat load extracted from data rooms was 27 kW, and the capacity of the cooling machine 55,7 kW (it works at 48,47 % capacity).

Chapter ***6***

Discussion

In this chapter, the presented results will be discussed. Problems encountered in the system will be brought forward and possible solutions proposed. Alternatives to the current system will be studied as well as a new energy system proposal.

6.1 Problems Encountered in the System

6.1.1 Size of the Main Energy System

After analysing the data from 2019, it is clear that the heat pump system is over-dimensioned. This becomes clear when looking at Figure 6.1 where the installed capacity is compared to the maximum and average values from 2019. With a capacity of 276 kW each of heating and 226 kW each of cooling, the system on the day that had the maximum heating load from the heat pump, worked at a 25,18 % of capacity (139 kW maximum heating load vs. 276 x 2 kW heating capacity). On average, the heating load from the heat pump was 14 kW - 2,53 % of the total installed capacity. In terms of cooling, the maximum cooling load was 207 kW - 45,79 % of the installed capacity. On average, the system required 24 kW of cooling, constituting only 5,3 % of the total capacity.

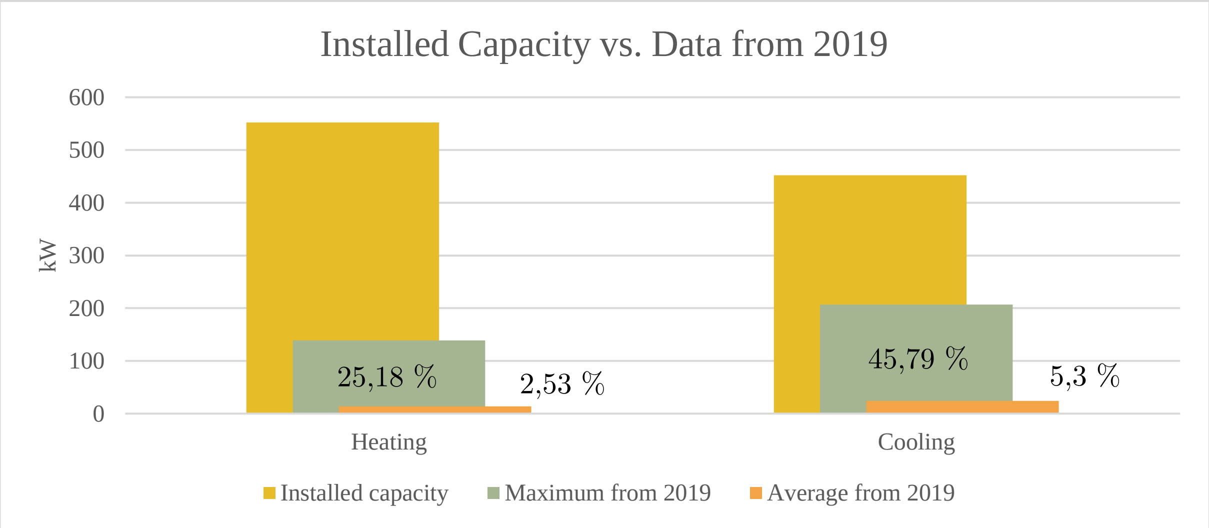


Figure 6.1 Comparison between the installed capacity of the heat pumps and the real use in 2019 [88].

Other parameters that indicate that the system is over-dimensioned, are the heating and cooling oversizing parameters (Eq. 2.33 and Eq. 2.34). Since the design loads are not available, the calculation will be performed with the maximum heating and cooling demands from 2019:

$$\alpha_h = \frac{P_{installed} - P_{max\ heating}}{P_{max\ heating}} = \frac{552\ kW - 139\ kW}{139\ kW} = 2{,}97$$

$$\alpha_c = \frac{P_{installed} - P_{max\ cooling}}{P_{max\ cooling}} = \frac{452\ kW - 207\ kW}{207\ kW} = 1{,}18$$

It can be seen in the results, that since the parameters α_h and α_c are higher than 0, the system is over-dimensioned. α_h is further from zero than α_c, which means that the system is more over-dimensioned regarding space heating.

Evaluation parameters will now be calculated based on the data gathered (Eq. 2.2 and Eq. 2.3):

$$COP_{heating + cooling\ (HP)} = \frac{Q_{out\ hot} + Q_{out\ cold}}{W_{in}} = \frac{99\ 709 + 65\ 216\ kWh}{57\ 465\ kWh} = 2{,}87$$

In order to see the effects that part load operation has on the system, the PLF factor defined in Section 2.3.2 (Eq. 2.35 and Eq. 2.36) will be calculated:

$$PLR_{heating\ HP} = \frac{P_{max}}{P_{design}} = \frac{139\ kW}{2 * 276\ kW} = 0{,}2518$$

$$PLF_{heating\ HP} = 1 - C_d(1 - PLR) = 1 - 0{,}25\ (1 - 0{,}2518) = 0{,}82$$

$$PLR_{cooling\ HP} = \frac{P_{max}}{P_{design}} = \frac{207\ kW}{2 * 226\ kW} = 0{,}4579$$

$$PLF_{cooling\ HP} = 1 - C_d(1 - PLR) = 1 - 0{,}25\ (1 - 0{,}4579) = 0{,}87$$

The parameter C_d will be set to 0,25 as stated in [60]. Taking the average of the two values, one can observe that the real COP of the system is in average 15% less due to part load operation.

$$COP_{heating + cooling\ (HP)\ REAL} = 2{,}87 * 0{,}845 = 2{,}42$$

It should be noted that both $PLR_{heating\ HP}$ and $PLR_{cooling\ HP}$ are calculated with the maximum heating and cooling demand, when the worst-case scenario would be if they were calculated with the average load, 14 kW and 24 kW respectively. This would lead to a PLF factor of 0,76, obtaining an even lower COP of 2,18 on average. Results show that an over-dimensioned system impacts directly the performance of the system.

It would also be of interest to calculated the Seasonal Performance Factor (SPF) defined in Eq. 2.5, in order to know the real performance of the system taking into account the electrical consumption of all auxiliary systems. To do this, it is required to know the electricity use of auxiliar systems to the heat pump such as fans and pumps. This information is not available to the student, so the SPF will not be calculated.

6.1.2 Size of the Cooling Machine

In order to see the effect of part load operation in the cooling machine as well, the PLF factor and the real COP will be calculated:

$$COP_{cooling\ machine\ (DR)} = \frac{Q_{out\ hot}}{W_{in}} = \frac{108\ 243 + 12\ 870 + 115\ 407\ kWh}{56\ 138\ kWh} = 4{,}21$$

$$PLR_{cooling\ machine} = \frac{P_{max}}{P_{design}} = \frac{27\ kW}{55{,}7\ kW} = 0{,}4847$$

$$PLF_{cooling\ machine} = 1 - C_d(1 - PLR) = 1 - 0{,}25\ (1 - 0{,}4847) = 0{,}87$$

$$COP_{cooling\ machine\ REAL} = 4{,}21 * 0{,}87 = 3{,}66$$

As can be observed, this system loses 13 % of the COP due to the fact that the cooling machine almost always operates at half capacity.

6.1.3 Control of the System

Looking at Figure 5.2 and Figure 5.6, one can see that the auxiliary system, the electric boiler, does not work when it should. This system should be on only in two scenarios: when the heating demand is so high that the heat pump units alone cannot cover them, and when the outside temperature is below -15 ºC (or in case of simultaneous failure of both of the units). The first scenario never happened during 2019, according to the gathered data and reflected in Figure 5.2. The second scenario only happened a few times during the end of January 2019.

A thorough analysis of the data shows that of the 485 h that the electric boiler was on in 2019, 443 h it should not have been on (because the heat pump was producing less than 60 kW at that hour and the outside temperature was higher than -14 ºC[5]). This means that 91 % of the time that the boiler was on, it should not have been. Even though this leads to higher energy consumption, heat from the electric boiler only represents an 8 % of the total space heating load, as seen in Table 5.1.

This shows, along with the information described in Section 3.4, that the sensor RT41 measured a temperature lower than the setpoint. This means that the heat pumps were not working properly at that time. Figure 6.2 shows that during that period, the heat pumps were working with a load of 15 kW, which is much lower than their capacity (2,71 % of the total heating capacity). This suggests that there is a problem with the control system of the heat pumps, which tells them to lower their capacity when the conditions are not met. The referred setpoint is unknown (although it is seen that is around 24 ºC), since it changes overtime and this information is not stored in GK's servers.

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[5] This value was taken as -14 ºC in the calculations, to give a 1 ºC margin of error to the sensor which measures the temperature outside.

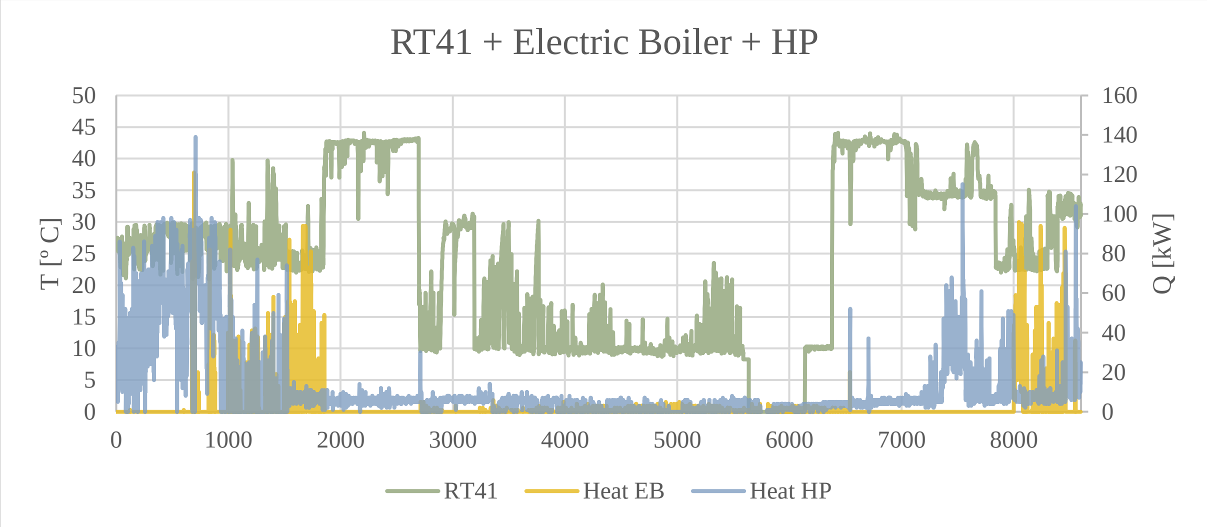

Figure 6.2 RT41 sensor data and the heating from electric boiler and heat pumps [88].

It can be seen in Figure 6.3, that the days when the electric boiler was used the most do not match the days when the heat pumps were most used. In fact, it can be clearly seen that most of the times when the electric boiler was turned on, the heat pumps did not deliver much energy.

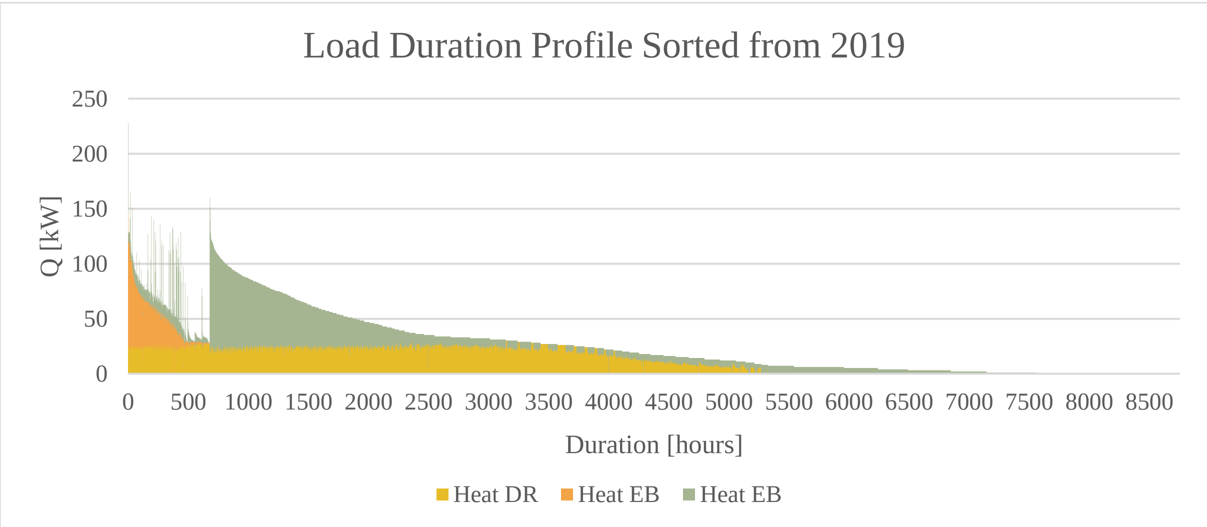

Figure 6.3 Load duration profile from 2019. Sorted from highest to smallest use of the electric boiler [88].

6.2 Possible Improvements Based on the Analysis and Current Technology

After a deep analysis of the system, several proposals will be given. First, changes that can be made in the current system, and second general improvements (as if the system was built nowadays from scratch).

6.2.1 Changes to the Current System

Control System

Due to the control system that is implemented now in the building (explained in Section 3.4), the heat pump units fluctuate from heating to cooling mode several times every hour. This is due to the system, that measures the temperature inside and regulates the heat pumps behaviour based on that. This fluctuation between modes is prejudicial because leads to worse indoor temperature quality (the system overheats and the cools down), in addition to an increase in the energy consumption. However, by using the strategies described in Section 2.5.3, one could anticipate to the changes in temperature by using historical data, weather data and mathematical models, avoiding unnecessary switching of modes. Since the company already has historical data stored from many years and the weather data is accessible, it should not be difficult to apply these structures. This new control structure could lead to an energy reduction, as well as a better indoor temperature quality and user satisfaction.

Variable Speed Pump and Motor Valves

Apart from the pumps distributing heat to the air handling units (JP42 and JP43 on the main energy system), no other pumps can vary their speed. However, due to the size of the system, which has already been concluded is too big, the system could benefit from the use of variable speed pumps. Especially the heat pump distribution pumps (JP40 and JP41), could allow a better adaption to the real energy needs and lead to energy savings, as well as to an improvement of the use of the system and a reduction in operational and maintenance costs.

In order to do this, some motorized valves should be installed before and after each of the heat pumps (see Figure 6.4), so they can operate individually. This could be performed if closing valves are to be found in the system.

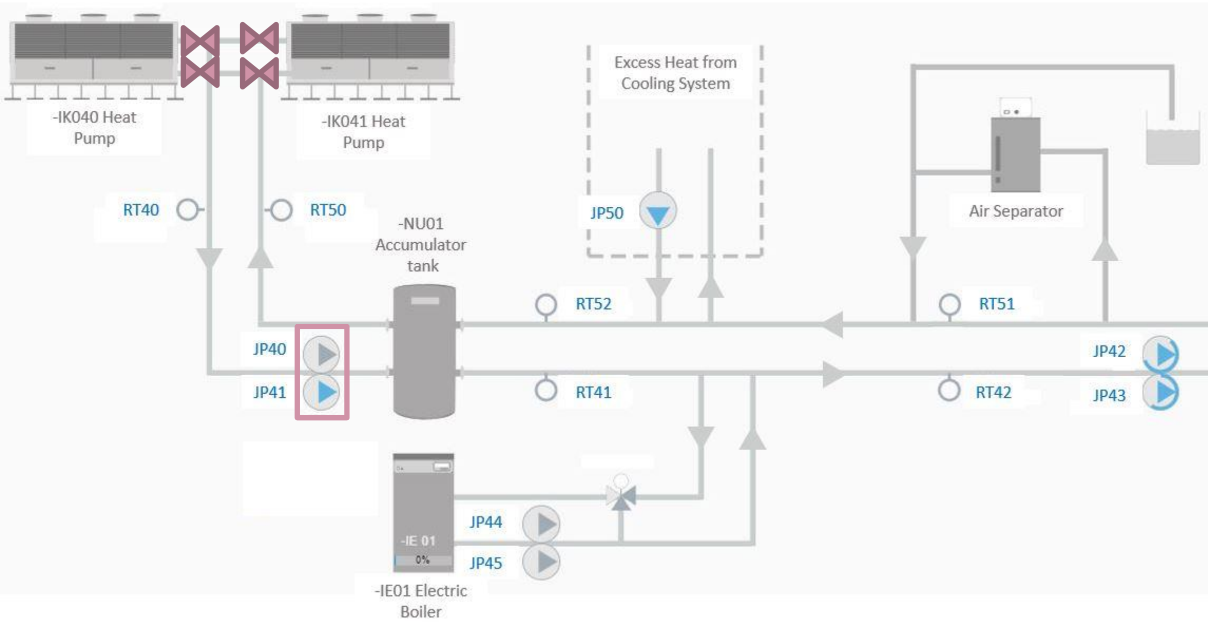

Figure 6.4 Main energy system of the building with the proposed changes. Marked in pink are the pumps and valves which should be changed.

6.2.2 General Improvements

Dimensioning of the Main Energy System

Before proposing a new dimensioning of the system, a brief study of other years before 2019 will be carried out, in order to see if the year 2019 was particularly warm or cold.

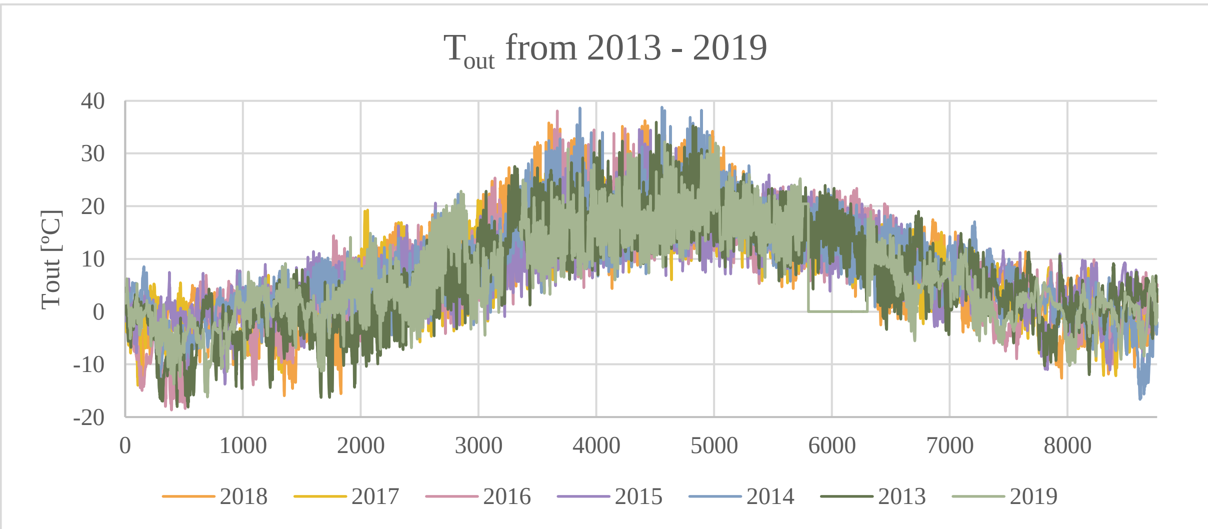

Figure 6.5 Outside temperature from the years 2019 to 2013 [88].

Figure 6.5 shows that the outside temperature in 2019 followed the pattern of previous years, not deviating from the general behaviour seen since 2013.

It is of interest to know how many hours of those years required heating and cooling to be able to quantify the difference between years. For this purpose, HDH and CDH will be calculated, which were previously explained in Section 2.2.3.

From a heating perspective, one can see in Figure 6.6 that 2019 had more heating degree hours than the rest of the analysed years. This means that 2019 (most likely) had the largest heating demand and that it would be wise to take these numbers as a reference. However, the average maximum heating load will be used for dimensioning, since all years have similar HDH, varying around 142 kW. The reference temperature for the calculations was set to 10 ºC (considering that it is a highly insulated building), but since all years where studied based on the same temperature, it is not very relevant.

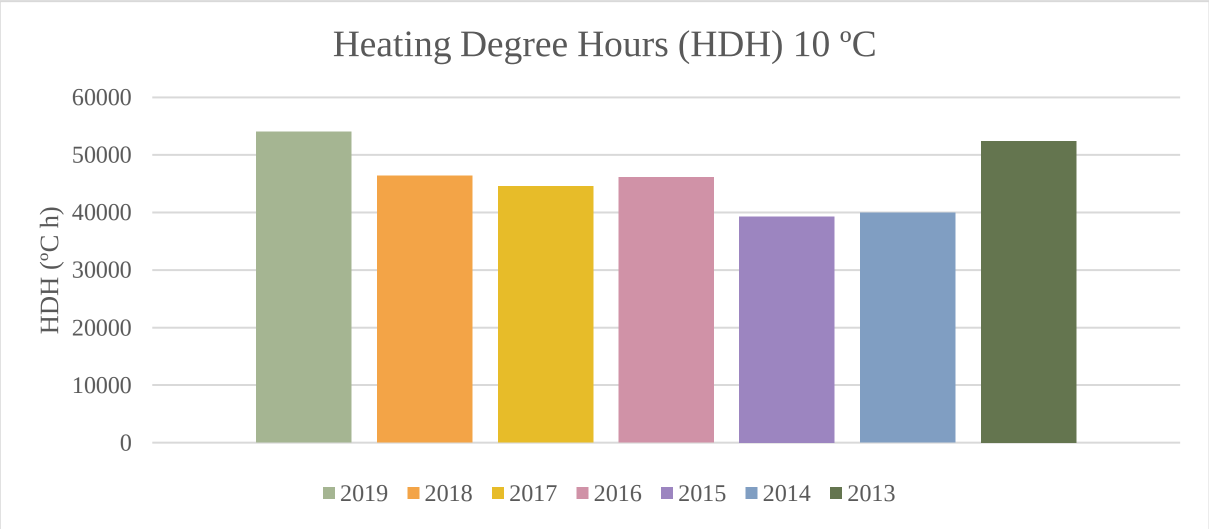

Figure 6.6 Heating degree hours from years 2013 to 2019 [88].

Nevertheless, one can see in Figure 6.7 that 2019 was not an especially warm year, exhibiting the third lowest cooling degree hours. Since the systems limitation is the cooling capacity, a brief analysis of the year with the most cooling degree hours will be made, to make sure that the chosen capacity of the heat pump would also cover warmer years.

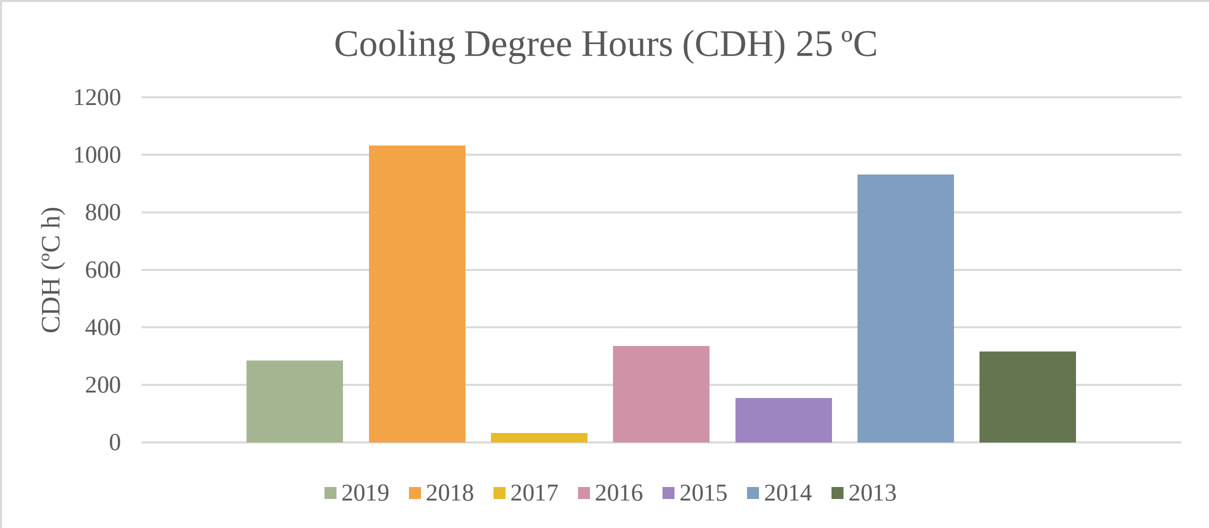

Figure 6.7 Cooling degree hours from years 2013 to 2019 [88].

When considering a new dimensioning of the system, it has to be taken into account that the limiting factor is the cooling loads, since they are entirely covered by the heat pump and are higher than the heating loads (heating loads are covered by heat pumps, data rooms and electric boiler). Bearing this in mind, at least 259 kW of cooling must be considered. According to Table 6.1, 2014 had the largest cooling load with 259 kW, which is 25 % more than the 207 kW of 2019. From now on, the limitation of cooling capacity will be set to 286 kW (a security margin of 10 % above the highest registered cooling load).

Table 6.1 Maximum cooling loads from years 2013 to 2019 [88].

Year	Maximum cooling load
2019	207 kW
2018	217 kW
2017	156 kW
2016	209 kW
2015	147 kW
2014	**259 kW**
2013	132 kW

At this point, two options are considered which are compared in Figure 6.8:

1) One heat pump which covers all cooling loads on its own.

2) A set of two smaller heat pumps which cover all cooling loads while working simultaneously, but alternate during heating season.

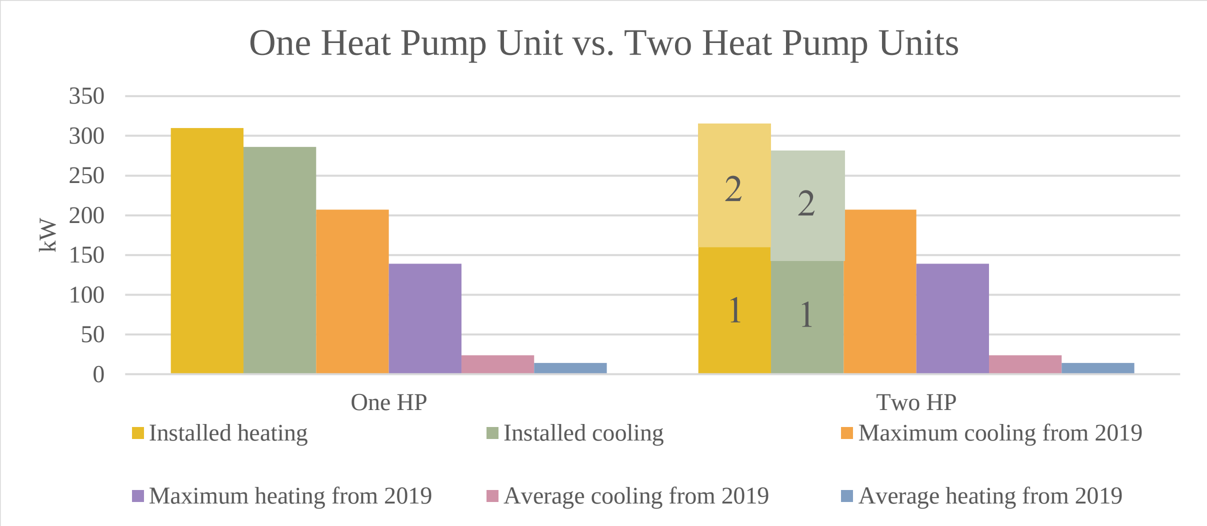

Figure 6.8 Comparison of the two proposed alternatives.

Option 1: One Heat Pump Unit

If option one was chosen, a heat pump with a capacity of 286 kW would be used. This would mean that during summer the heat pump would work almost at full capacity. If one assumes that the relationship between heating and cooling capacity is the same as in the one in the installed heat pumps, with 286 kW cooling capacity, a heating capacity of around 310 kW will be reached (usually in reversible heat pumps the heating capacity is around 8 - 10 % higher (from a comparison between several reversible heat pumps [91])). During heating season, using data from 2019, the system would be working at 44,83 % at maximum load and 4,51 % at average load. During cooling season, the system would work at 72,37 % at maximum load and 8,39 % at average load.

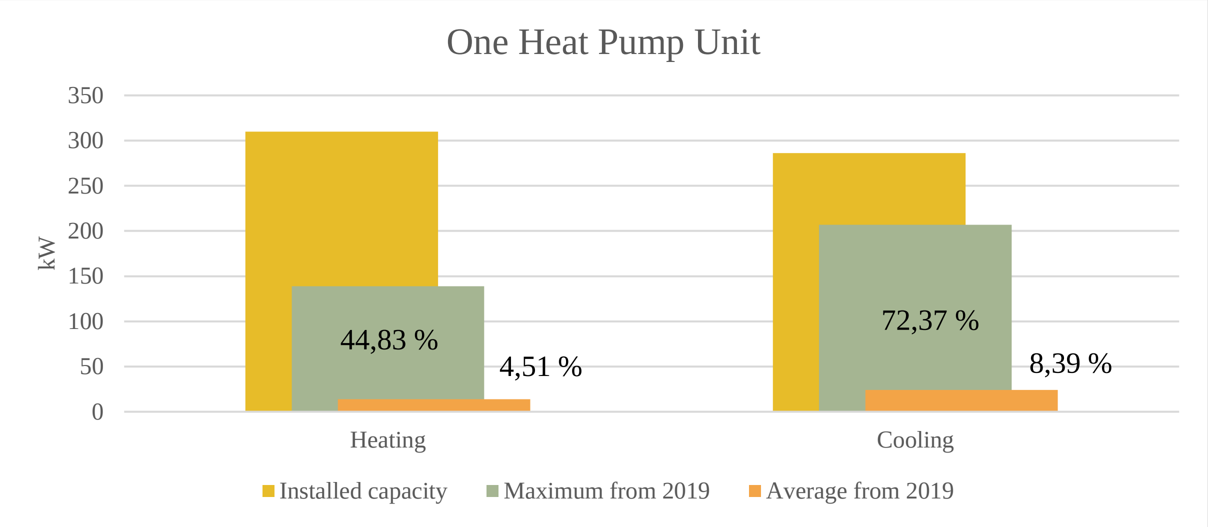

Figure 6.9 Option 1: One heat pump unit. Comparison between heating and cooling capacities[6].

To see the improvement from the original system, the equivalent operation time and other parameters will be calculated:

$$\tau_{heating} = \frac{Q_{HP-heating}}{P_{HP-dim-heating}} = \frac{225\,493\ kWh}{310\ kW} = 727{,}39\ h$$

$$\tau_{cooling} = \frac{Q_{HP-cooling}}{P_{HP-dim-cooling}} = \frac{65\,216\ kWh}{286\ kW} = 228{,}02\ h$$

$$PLR_{heating\ HP\ OP1} = \frac{P_{max}}{P_{design}} = \frac{139\ kW}{310\ kW} = 0{,}4483$$

$$PLF_{heating\ HP\ OP1} = 1 - C_d(1 - PLR) = 1 - 0{,}25\ (1 - 0{,}4483) = 0{,}87$$

$$PLR_{cooling\ HP\ OP1} = \frac{P_{max}}{P_{design}} = \frac{207\ kW}{286\ kW} = 0{,}7237$$

$$PLF_{cooling\ HP\ OP1} = 1 - C_d(1 - PLR) = 1 - 0{,}25\ (1 - 0{,}7237) = 0{,}931$$

Which on average would be a PLF factor of 0,9:

$$COP_{heating+cooling\ (HP)\ OP1\ REAL} = 2{,}87 * 0{,}9 = 2{,}58$$

This improvement would improve the COP in 6,6 %.

If the calculations were made with average load values (14 kW and 24 kW respectively), the PLF factor would be 0,7665, and the real COP would be 2,19, which is an improvement of 0,4 % from the reference case based also in average load values.

[6] The comparison will be made with data from 2019, although the dimensioning is done based on the highest stored cooling load.

Option 2: Two Heat Pump Units

If option two was chosen, two heat pumps with a capacity of 143 kW of cooling would be installed. This would mean that the heating capacity of these heat pumps would be 160 kW. During summer, both heat pumps would work simultaneously in order to cover all cooling loads, but if the cooling demands are lower, they could alternate. However, during heating season, they would alternate. The system would work with 86,87 % at maximum heating load, and 8,75 % at average heating load. Since heat from data rooms is available during the whole year and is approximately 27 kW, all heating loads would be covered with these two systems. However, an auxiliary system is needed for safety reasons (when heat pumps stop working) and for when the temperature outside is too low. If used properly, this system would need to have a capacity of 150 kW, to be able to provide the heat necessary when both heat pumps are off (heat from data rooms would still be available). Nonetheless, this is not very likely to happen and a smaller auxiliary system could be installed.

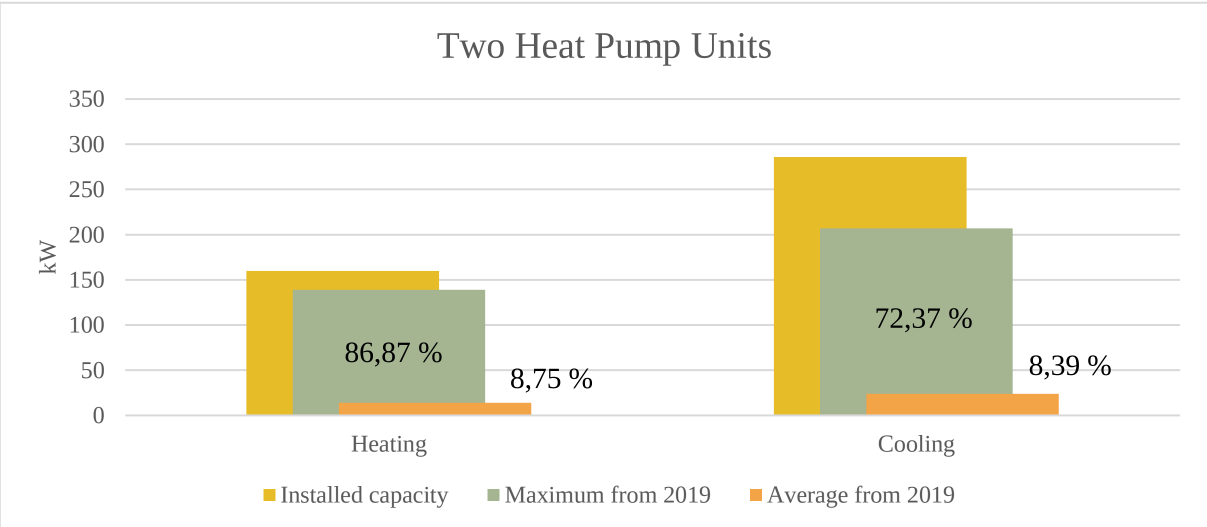

Figure 6.10 Option 2. Comparison between heating and cooling capacities. One heat pump working during heating season, two heat pumps working during cooling season.

In addition, it must also be noted that a heat pump unit that works full time, will require more maintenance, consequently having to stop the system more frequently.

If the equivalent operation time is calculated, using Eq. 2.31:

$$\tau_{heating} = \frac{Q_{HP-heating}}{P_{HP-dim-heating}} = \frac{225\,493\ kWh}{160\ kW} = 1409{,}33\ h$$

$$\tau_{cooling} = \frac{Q_{HP-cooling}}{P_{HP-dim-cooling}} = \frac{65\,216\ kWh}{2 * 143\ kW} = 228{,}02\ h$$

One can see that the equivalent hours are closer but still lower than the usual values (1 500 – 3 000 for buildings), as stated in [58].

$$PLR_{heating\ HP\ OP2} = \frac{P_{max}}{P_{design}} = \frac{139\ kW}{160\ kW} = 0{,}8687$$

$$PLF_{heating\ HP\ OP2} = 1 - C_d(1 - PLR) = 1 - 0{,}25\ (1 - 0{,}8687) = 0{,}968$$

$$PLR_{cooling\ HP\ OP2} = \frac{P_{max}}{P_{design}} = \frac{207\ kW}{286\ kW} = 0{,}7237$$

$$PLF_{cooling\ HP\ OP2} = 1 - C_d(1 - PLR) = 1 - 0{,}25\ (1 - 0{,}7237) = 0{,}931$$

Which on average would be a PLF factor of 0,95:

$$COP_{heating + cooling\ (HP)\ OP2\ REAL} = 2{,}87 * 0{,}95 = 2{,}73$$

Which is 12,8 % improvement on the COP.

If the calculations were made with average load values (14 kW and 24 kW respectively), the PLF factor would be 0,771, and the real COP would be 2,21, which is an improvement of 1,37 % from the reference case based also in average load values. A summary of the improvements in the COP of both options can be seen in Table 6.2. It can be observed, that there is no significant improvement when average loads are considered.

Table 6.2 Summary of the different proposed options for the heat pump system.

	Reference Case	Reference Case average	Option 1 max load	Option 1 average	Option 2 max load	Option 2 average
COP	2,87	2,87	2,87	2,87	2,87	2,87
PLF	0,845	0,77	0,90	0,77	0,95	0,77
COP_{REAL}	2,42	2,18	2,58	2,19	2,73	2,21
Improvement	--	--	6,6 %	0,4 %	12,8 %	1,37 %

In both proposed options, the system would work at less than 10 % capacity during average loads. After seeing that a system that covers all heating and cooling loads would be over-dimensioned most of the time, and it is not reasonable, a third option will be considered.

Option 3: Two Smaller Heat Pump Units and a Peak Load System

To propose and size a peak load system, it must be known, how much percent of the load it must cover. It has been seen, that the previous options were too over-dimensioned for the average loads. In order to see how the loads are distributed and to propose a sizing for the peak load system, both cooling and heating loads will be sorted in 10-kW intervals.

In Figure 6.11 can be seen, that 73,41 % of the hours that the heat pump was on heating mode, the load was lower than 10 kW. This is relevant when sizing the heat pumps, because even though the maximum heating load is 139 kW, only 0,10 % of the time the heat pumps work with a load higher than 100 kW.

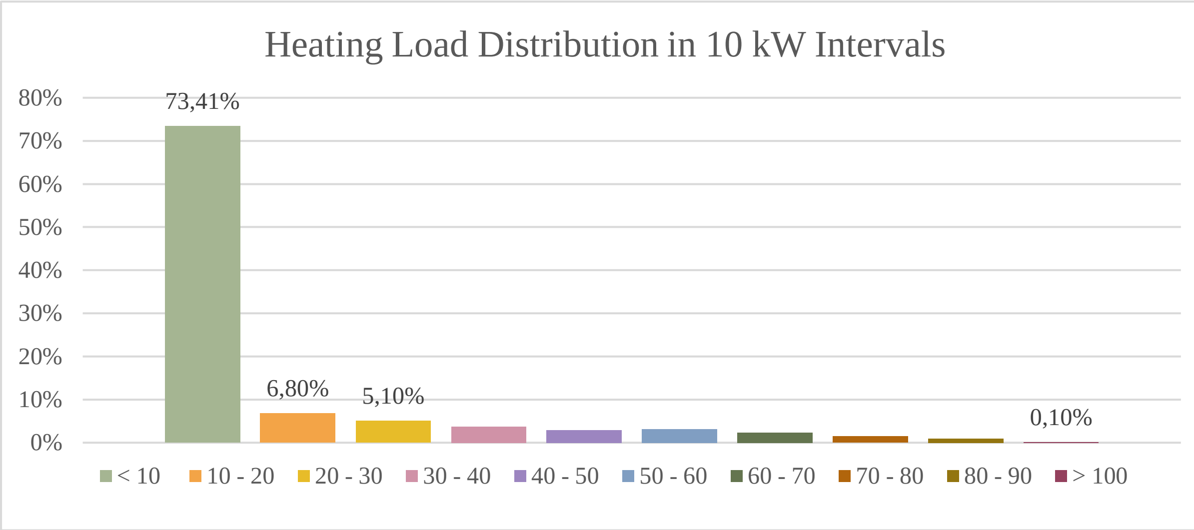

Figure 6.11 Heating load distribution in 10 kW intervals [88].

In Figure 6.12, the same distribution but with cooling loads can be seen. In this case, the heat pump worked almost 4 % of the time with loads higher than 100 kW, but it is still a too low percentage.

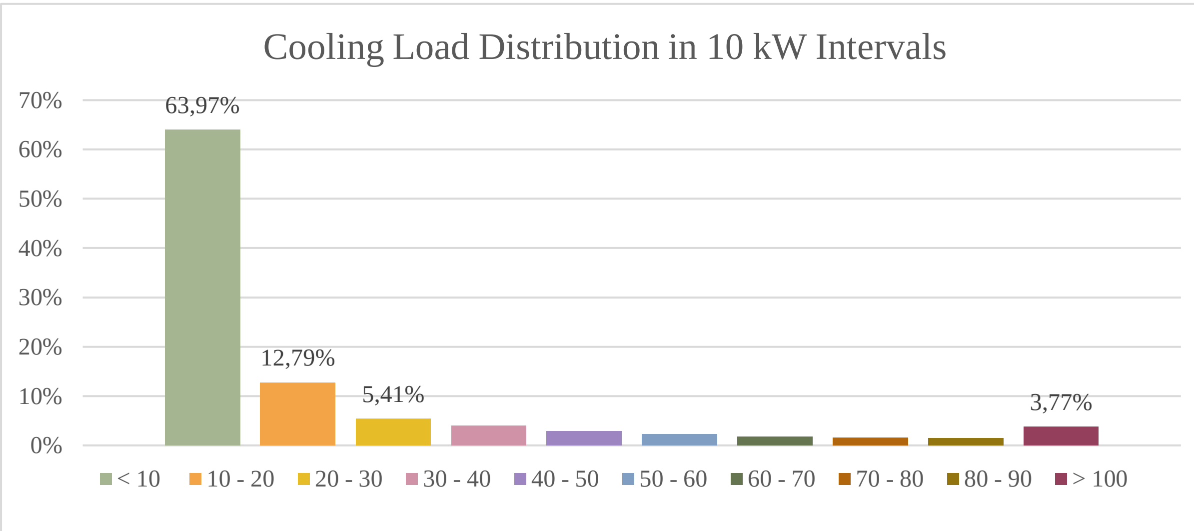

Figure 6.12 Cooling load distribution in 10 kW intervals [88].

Based on the data recovered from Figure 6.11 and Figure 6.12, it can be said with certainty that a peak load system would be beneficial for this system.

If the selected peak load system, had a heating capacity of 20 kW, and a cooling capacity of 18 kW, 80 % of the time that the system is on heating mode and 76 % of the time it is on cooling mode would be covered. It has been previously seen the benefits of having two units instead of one, to be able to alternate them. In terms of heating, if both heat pumps are working at the same time, the peak load system would have to have at least 120 kW capacity to cover all peak loads. In terms of cooling, and auxiliary system should cover 250 kW. This can be done with a CTES as explained in Section 2.5.2.

In Figure 6.13, the composition of Option 3 can be seen. In heating mode, as mentioned earlier, one heat pump unit could be working 80 % of the time, alternating between the two available. An additional 8 % of the time (88 % in total), the heat pump system alone could cover the heating demands, if both heat pump units work simultaneously. This means that the peak load system would work 12 % of the time. In cooling mode, around 76 % of the time, one heat pump unit would be enough to cover the cooling demands, increasing up to 84 % if both heat pump units work simultaneously. The thermal storage system would have to cover 16 % of the cooling time.

In terms of capacity, during heating mode, one heat pump unit would work at 70 % with average loads (14 kW). During cooling mode, one heat pump unit only would not cover the average cooling load, but if both of them were working at the same time, they would work at 66,66 % capacity. During both of the modes, the units would work at 100 % with maximum loads.

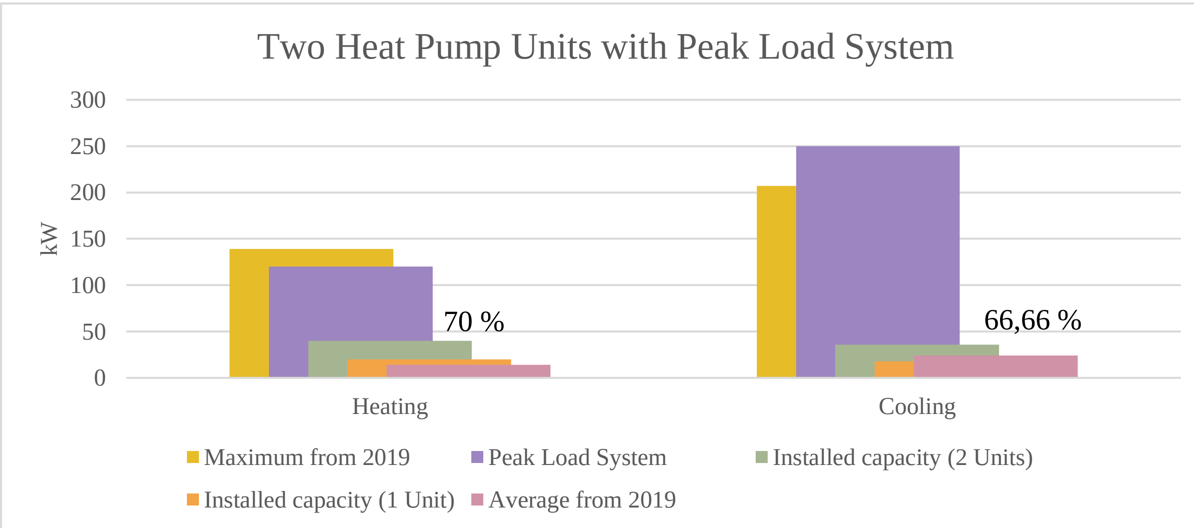

Figure 6.13 Option 3. Comparison between heating and cooling capacities and the addition of a peak load system, one for heating and one cooling.

Now, the previously calculated parameters will be calculated for this option, this time based on average values:

$$PLR_{heating\ HP\ OP3} = \frac{P_{max}}{P_{design}} = \frac{14\ kW}{20\ kW} = 0,7$$

$$PLF_{heating\ HP\ OP3} = 1 - C_d(1 - PLR) = 1 - 0,25\ (1 - 0,7) = 0,925$$

$$PLR_{cooling\ HP\ OP3} = \frac{P_{max}}{P_{design}} = \frac{24\ kW}{36\ kW} = 0,667$$

$$PLF_{cooling\ HP\ OP3} = 1 - C_d(1 - PLR) = 1 - 0,25\ (1 - 0,667) = 0,916$$

Which on average would be a PLF factor of 0,92:

$$COP_{heating\ +\ cooling\ (HP)\ OP3\ REAL} = 2,87 * 0,92 = 2,64$$

Which is 21,1 % improvement on the COP on average values. Since the PLF factor for maximum loads is 1 (there would be no part load operation), the COP would be 2,87, which translates to 18,59 % improvement. In Table 6.3 a summary of the improvements based on Option 3 can be seen.

Table 6.3 Comparison between the reference case and Option 3.

	Reference Case	Reference Case average	Option 3 max. load	Option 3 average loads
COP	2,87	2,87	2,87	2,87
PLF	0,845	0,77	1	0,92
COP_{REAL}	2,42	2,18	2,87	2,64
Improvement	--	--	18,59 %	21,1 %

Dimensioning of the Cooling Machine

As explained before, the cooling machine is almost double required size. However, it works at a “fixed” demand, so there is not a lot of variation in the load. If there is no plan on expanding the data rooms in the future, it is better to have the right size heat pump. It has been already calculated that the part load operation lowers the COP by 13 %.

If the system had 36 kW capacity, which would allow the addition of an extra data room cooler, the evaluation parameters would be:

$$\tau_{cooling\ machine\ NEW} = \frac{Q_{cooling\ machine\ NEW}}{P_{cooling\ mcahine\ NEW}} = \frac{108\ 243 + 12\ 870 + 115\ 407\ kWh}{36\ kW}$$
$$= 6\ 570\ h$$

$$PLR_{heating\ HP\ OP2} = \frac{P_{max}}{P_{design}} = \frac{27\ kW}{36\ kW} = 0{,}75$$

$$PLF_{heating\ HP\ OP2} = 1 - C_d(1 - PLR) = 1 - 0{,}25\,(1 - 0{,}75) = 0{,}9375$$

$$COP_{heating + cooling\ (HP)\ OP2\ REAL} = 4{,}21 * 0{,}9375 = 3{,}95$$

Which is an improvement of 7,92 % in the COP. So changing the machine to one that has a lower capacity would increase the COP and subsequently reduce the electrical consumption.

Table 6.4 Summary of the proposed improvement for the cooling machine.

	Reference Case	New Case
COP	4,21	4,21
PLF	0,87	0,9375
COP_{REAL}	3,66	3,95
Improvement	--	7,92 %

Economic analysis of the proposed alternatives

A brief economic analysis can be performed to see how much money is lost only due to the part load operation of the heat pumps, to see how much money could be saved if any of the proposed alternatives is chosen, due to the improvement of the COP.

The part load operation of the current heat pumps (reference case), supposed an increase in electricity consumption of 10 685 kWh in 2019, compared to what it would have been if the system worked at full load. The price for electricity during the first quarter of 2021 was in average 0,494[7] NOK/kWh for commercial buildings according to [92] (https://www.ssb.no/en/energi-og-industri/statistikker/elkraftpris), so that increase in electricity consumption entailed an increase of 5 278 NOK (compared to a full load operation scenario).

In the cooling machine, the part load operation increased the electricity consumption by 8 485 kWh per year, which had a cost of 4 192 NOK/year.

[7] To carry out a more precise analysis, the exact tariff of the company should be used, since some have their own deal. For this calculation, a generic value will be used.

To see how much the proposed improvements save in terms of electricity and costs, the same procedure will be applied in Table 6.5:

Table 6.5 Economic impact of the part load operation and the proposed improvements.

	Ref. case	Ref. case avg	Opt. 1 max load	Opt. 1 avg	Opt. 2 max load	Opt. 2 avg	Opt. 3 max load	Opt. 3 avg	CM ref. case	CM
COP_{REAL}	2,42	2,18	2,58	2,19	2,73	2,21	**2,87**	**2,64**	3,66	3,95
Improvement	--	--	6,6 %	0,4 %	12,8 %	1,37 %	**18,59 %**	**21,1 %**	--	7,92 %
El. [kWh/year]	68 149	75 654	63 924	75 308	60 412	74 627	**57 465**	**62 472**	8 485	59 878
Costs (↑)/ Savings (↓) [NOK/year]	↑ 5 278	↑ 10 497	↓ 2 087	↓ 171	↓ 3 822	↓ 507	↓ **5 278**	↓ **6 512**	↑ 4 191	↓ 2 344

It can be seen, that the most profitable option would be option 3, both with maximum and average loads. However, in this analysis were not taken into account other factors such as the operational and maintenance costs or the investment costs.

Lower Cooling Demands

Since the limiting factor when dimensioning the heat pump system are the cooling loads, lowering these loads will allow a smaller system and a better use of it. Several possibilities to do this are presented below:

Night Cooling

As explained in Section 2.5, night cooling or ventilation is an easy and cheap way to lower the building's temperature during the night. In this building in particular, the use of an auxiliary system to lower cooling demands could balance heating and cooling demands, leading to a better fit of the energy system. However, a thorough analysis is required to see if this measure is actually beneficial.

Cooling Thermal Energy Storage (CTES)

A more complex system, which requires a higher investment, is installing a CTES technology. While night cooling is limited to night hours and is used during the next few hours, CTES has the advantage of storing high quantities of cold for a longer period of time. This system could help lower the cooling demand by shaving the peaks, and would be charged in times where the demand is not as high (night times for example).

Different Heat Pump System for DHW

As seen in Figure 4.6 and Figure 4.7, electrical heating and distribution makes up for 14,75 % and 85,25 % of the electricity consumption, respectively. In Section 3.3 the different systems that make up the energy central were explained and the DHW system was studied. To sum up, it consists of three 400 L tanks, of which one is electrified to provide the extra heat required. However, the efficiency of this system can be 100 % maximum. A possible way to reduce electricity consumption, could be to install a separate heat pump system for heating of DHW.

Small CO_2 Heat Pump

CO_2 has been used for high temperature water heating due to its good characteristics, reaching water temperatures up to 90 ºC (see Section 2.1.2).

The DHW system recovers heat from data rooms in addition to using heat from the electric resistance in one of the tanks. The total energy consumption of the system was seen in Figure 6.14. The maximum load was 28 kW in March 2019. If 5% security margin is used, a 30-kW CO_2 heat pump should be installed to cover all DHW demands. Since these kinds of systems can have a COP as high as 4,7 [93], there would be a substantial electricity reduction. Taking into account that in 2019, 39 510 kWh of heat were transferred in total to the DHW system, a COP of 4,7 gives an electricity demand of 8 406 kWh, approximately 70 % less than the current electric tank (the electricity used to extract heat from data rooms is neglected in this calculation, since it is also used to space heating).

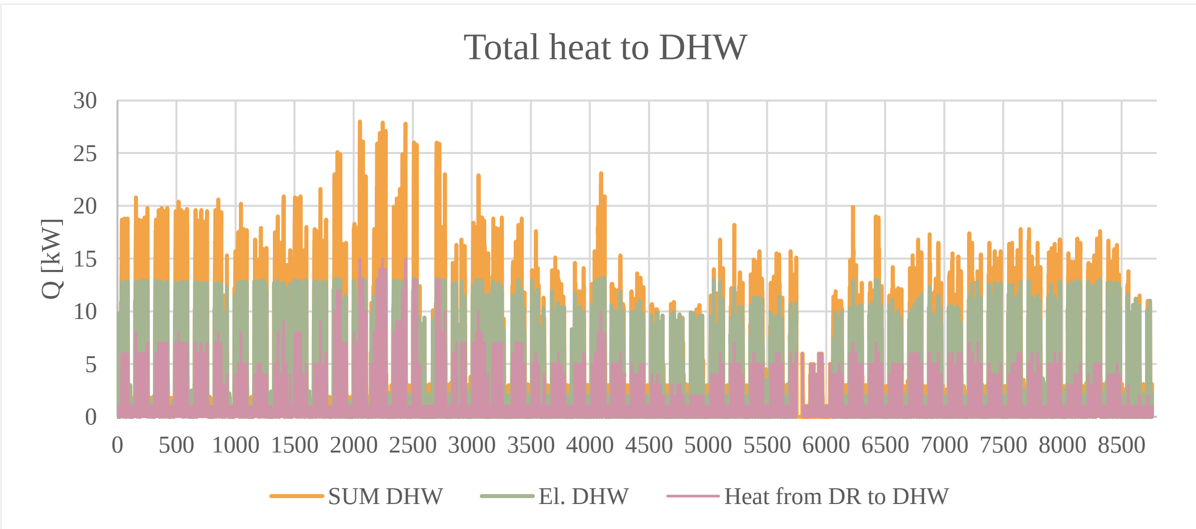

Figure 6.14 Total heat distribution to DHW [88].

It can also be studied the possibility to continue using heat from data rooms and only substitute the electrical tank. In this case the needed heat pump would be smaller, as the maximum load covered by the tank is 13 kW. Some companies [94] offer 15 kW CO_2 heat pumps, with a COP of 3,2 and a 1 000 L water storage (divided in two 500 L tanks). Therefore, a CO_2 heat pump could be used also in this scenario.

Heat Pump Cycle Improvements

Since the heat pump units are bought from an independent company, no change in the current units can be made, as no insight of the cycle is available to be analysed. However, in some cases it is possible to upgrade the cycle by adding or changing some components. If this would be the case, some improvements to the cycle are explained in Appendix A.

Other Refrigerants

The potential of natural refrigerants is previously studied in Section 2.1.2. For this building in particular, the use of ammonia as a refrigerant could be a better option than the current R410a. Since the system has inherent leaks, refrigerant has to be available to refill during operation. However, due to restrictions in Europe, it will be more difficult and more expensive to get a hold of non-environmentally friendly refrigerants.

Ammonia (R717)

An advantage when using a NH_3 heat pump, is that due to its properties, it can deliver heat at a high temperature, and because there is a big temperature glide in the condenser, heat can be delivered to both DHW and space heating. In order to have 90 ºC temperatures at the condenser, a two-stage ammonia heat pump would be needed [50].

By having two units of NH_3 heat pumps, of 160 kW of heating capacity in total, it would be enough to cover both heating/cooling demands and the DHW demands (which are approx. 13 kW). Thanks to its high enthalpy of evaporation, very low mas flow rate is required and therefore smaller pipes, compressors and devices in general. This would further reduce the cost of using such a system.

Propane (R290)

Another option would be to use propane as a refrigerant. Although it has good thermodynamical properties for the heat pump cycle, it is often not used due to its flammability and security restrictions [95]. However, the units could be placed outside to limit these hazards.

A reversible propane heat pump could be used for space heating and cooling purposes, with a capacity of 280 kW heating and 260 kW cooling if one unit was to be chosen or 145 kW heating, 136 kW cooling if two units were chosen and if no peak load system is desired [96].

Carbon Dioxide (R744)

As mentioned in the previous section CO_2 has a magnificent performance when high temperatures are needed. This could be the case when only one heat pump is desired to cover all loads: heating/cooling and DHW.

In Figure 6.15 a possible configuration of a CO_2 heat pump that covers both space heating and DHW demands is shown.

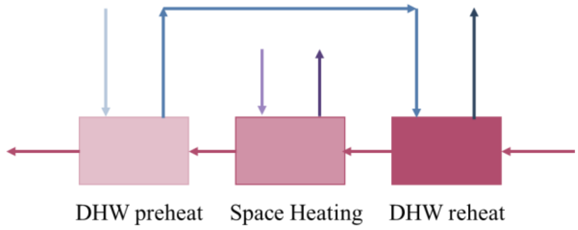

Figure 6.15 CO_2 system used for both space heating and DHW heating [97].

Dual Heat Pump Systems

PVT combined with a heat pump system

As explained in Section 2.5, one of the new technologies regarding solar thermal energy are the PVT, which, apart from producing electricity and improving the electricity production, are able to recover heat from the sun. Since the building has an interest in capturing electricity via PV panels, it is proposed that instead of having just PV panels, PVT panels could be installed to recover heat as well and improve the electricity production and use it to, for example, defreeze the evaporator when needed.

6.2.3 Proposal for an Energy System for Miljøhuset

Finally, after having proposed several improvements to the system, a final energy system will be proposed for Miljøhuset, having in mind the analysis performed in the previous section.

It has been seen that one of the main limitations of the system is the high cooling demands and the control of the auxiliary system, so this proposal will be focused on solving those problems.

Type of Heat Pump

The building is located in Ryen, Oslo. As it can be seen in Figure 6.16, the building is far from any water source, either sea water or fresh water. For this reason, a water-source heat pump is discarded for this case study.

Figure 6.16 Location of Miljøhuset in Oslo (Google Maps).

According to [89] and looking at Figure 6.17, previous evaluation of the site discarded any ground-source heat pumps, due to the 60 m of loose mass that there is until reaching proper ground. For the above-mentioned reasons, an air-source heat pump is selected.

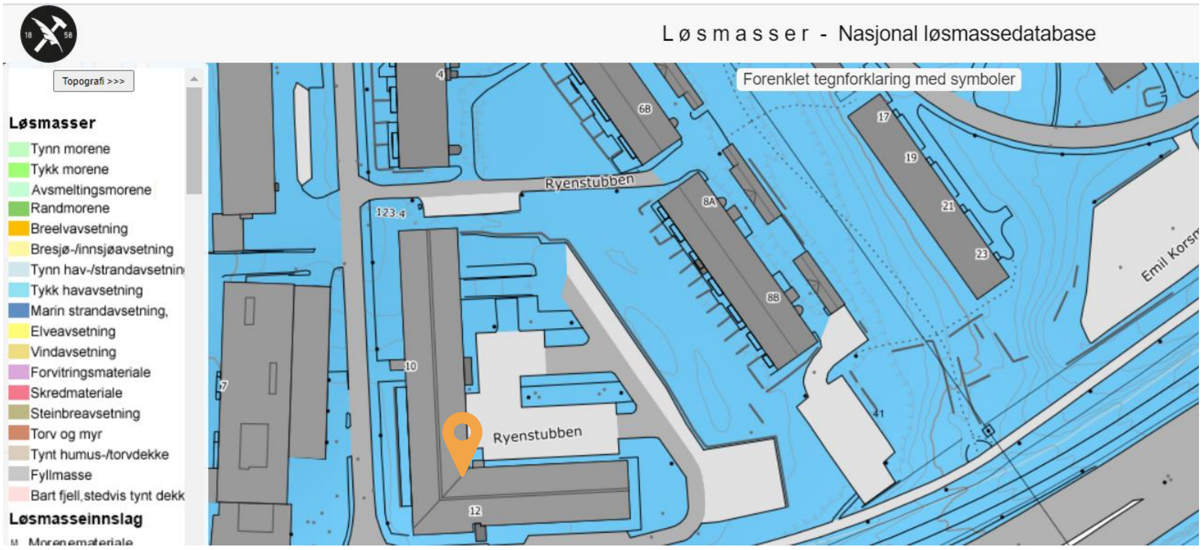

Figure 6.17 Geological map of the site. The legend indicates that the soil is mainly formed of gravel and sand [98].

Refrigerant Selection

A natural refrigerant heat pump is selected due to the many factors explained in Section 2.1.2. The selection of the refrigerant is based on the needs of the system. Propane cannot be used for DHW applications, so if a combined system is sought, it must be discarded. The use of ammonia, although very efficient in terms of energy, comes with several disadvantages, where security issues are the most prominent. Since the size of the system is not too big, it is not recommended to use ammonia. The final options are either propane or CO_2.

Dimensioning of the System

As discussed in Section 6.2.1, having two smaller units benefit the COP, while lowering the operational and maintenance costs. This justifies the declination towards having two units. Based on the analysis performed in Section 5.1, a new dimensioning of the system was proposed in Option 3: heating capacity of 20 kW, alternating the use of the two units, and a cooling capacity of 18 kW, using both units simultaneously if needed. Since this system would cover average loads, a peak load system is needed to cover maximum loads. This would lead to a better use of the heat pumps capacity and lower electricity consumption.

In order to be able to adjust to the needs of the building, avoiding on/off losses of the system, compressors with VSD are proposed. This would allow regulation of power of the units instead of alternation between on/off to reach the desired temperature. Another advantage of using VSD, is that the accumulator tank might be unnecessary, reducing costs.

It is also proposed a change in the current control system. Since large amount of data is available, a feedback/feedforward control system (see Section 2.5.3) based on this data and weather data, might avoid unnecessary work from the heat pumps, saving both energy and money.

There are two options now regarding the DHW system. There could be a heat pump system that could cover all space heating/cooling and DHW demands, that would run on CO_2 and would have to be bigger than the previously proposed sizing. Another option, would be to have a propane heat pump cover the space heating and cooling loads, and a separate 30-kW CO_2 heat pump to cover DHW demands. This would be a better option, since CO_2 systems are more suited to high temperature applications, and do not work as good if the main demand is space heating [99].

Peak Load System/Auxiliary System

If the chosen sizing of the heat pumps would be the one proposed in option 3, (see Section 6.2.2), to better fit the average loads, the building must be coupled to a peak load system to cover peak loads, so the district heating network could be an option. Using district heating would cover heating peak loads, but the system would still be limited by the cooling loads. Seeing that the cooling loads are so much higher than the heating loads, it seems like there is little room for benefit in this option if a thermal storage system is not used to reduce cooling demands. Since a very small heat pump would be needed to profit from the district heating coupling, it would not cover the cooling demands. Therefore, a thermal storage system is proposed to be able to have a small-sized heat pump.

Thermal Storage

Since the cooling demands are the ones limiting the sizing of the heat pumps, and do not have a back-up/auxiliary system, a cooling thermal storage is suggested. PCM tanks could lower the cooling loads as well as act as an auxiliary cooling system. The tanks would be charged during the night (by freezing the PCM), and used (by melting the PCM) in the central hours of the day when the cooling demands are higher. The capacity of these tanks can reach up to 1600 kW and work for 7 h (11200 kWh), as it is done in the University of Bergen [100]. In this case, such high capacity is not needed. This system would give more flexibility when choosing the size of the heat pump, since the cooling loads are the ones limiting its size.

The final energy system is presented below in Table 6.6.

Table 6.6 Final energy system proposed for Miljøhuset.

Main Energy System	Air-to-water reversible heat pump DR heat still used for space heating and DHW
Refrigerant	Propane
Dimensioning HP	Heating: 2 x 20 kW Cooling: 2 x 18 kW
Peak Load System	District heating Min. Capacity: 120 kW
Back-up System	District heating
DHW	Water-to-water heat pump, using heat from DR as a source CO_2 30 kW
Thermal Storage	PCM tanks (salt hydrate) Min. Capacity: 250 kW Charged during the night
Cooling Machine	Water-to-water heat pump Capacity: 36 kW Refrigerant: Propane

Chapter 7

Conclusion

In this thesis, the performance of the energy central of a large office building was studied. For this, a theoretical background was given as well as a description of the latest technology used in high insulated buildings. The described case was the Miljøhuset building from GK in Oslo, Norway.

The building's main heat source is a two-unit reversible heat pump system, connected to a heat exchanger which transfers heat extracted from data rooms. The size of these heat pumps is 226 kW each of cooling and 276 kW each of heating, with a total of 452 kW and 552 kW respectively. As an auxiliary system, the building has an electric boiler of around 130 kW. Heat is extracted from data rooms by three 9-kW data coolers and transferred to the main energy system by a 55,7-kW cooling machine. After a deep analysis of the data recovered from the servers from 2019 the following discoveries were made:

1. **The system is over-dimensioned**

The data shows a heating peak load of 139 kW and a cooling peak load of 207 kW in 2019. Historical data was reviewed and a maximum cooling peak load of 259 kW was found in 2014. In average terms, the average heating load was 14 kW, and the average cooling load 24 kW.

This means that the system works at 25,18 % for heating and 45,79 % for cooling during maximum loads. However, these percentages drop to 2,53 % and 5,30 % when talking about average loads. Due to this partial load operation, the COP was between 15 – 25 % lower than what it would be if the system were operating at full load (depending if the maximum loads or the average loads are considered).

An alternative dimensioning of the system was proposed, with two options: one or two-unit heat pump systems. It was taken into account that the limiting factor for the dimensioning of the heat pumps were the cooling loads. Option 1 consists of a single-unit heat pump with a capacity of 310 kW for heating and 286 kW for cooling, capable of covering all heating and cooling loads. This unit would work at 44,83 % and 72,37 % for heating and cooling respectively for maximum loads, and at 4,51 % and 8,39 % for average loads. The improvement in the COP was 6,6 % for maximum loads and 0,4 % for average loads.

Option 2 consists of two units of 160 kW heating and 143 kW cooling capacity each. During heating season only one heat pump would be working at a time, but both of them would be on during cooling season to be able to provide all the cooling necessary. In this scenario, the heat pumps would work at 86,87 % for heating and 72,37 % for cooling for maximum loads, and 8,75 % and 8,39 % respectively for average loads. This will result in 12,8 % improvement in the COP for maximum loads and 1,37 % for average loads. Other benefits related to having two heat pump units instead of only one, is the alternation between units, which lowers the operation time and therefore the maintenance.

Option 3 sizes a heat pump system which covers average loads, but also a peak load system and a thermal energy storage to cover maximum heating and cooling loads. In this option, to heat pump units are chosen as well, since it was seen that it was beneficial. Two 20 kW heating and 18 kW cooling heat pumps were selected, as well as a 120-kW peak load system and a 250-kW thermal energy storage. This system increased the COP in 18,59 % at maximum loads and 21,1 % at average loads. This was concluded to be the best option of all three proposed ones.

2. **Problems in the controller of the heat pumps, which causes a misuse of the electric boiler.**

The electric boiler should only work when either the temperature outside is lower than 15 ºC or one of the heat pumps fails or is not enough to cover heating loads. The analysis performed showed that from the 485 h that the electric boiler was on in 2019, 91 % of the time it should not have been on. Since the electric boiler is controlled using the RT41 sensor, it is suspected that the problem relies on the controller of the heat pumps. During the periods when the electric boiler was on, the heat pumps were working at very low load.

A change in the control system is proposed, changing the current one, which used the indoor temperature. The proposed system would use weather data, historical data and a mathematical model to control the energy system. This would also avoid unnecessary change between modes of the heat pump.

3. **Cooling machine over-dimensioned**

Part of the heat (48 %) delivered to the building comes from the cooling of data rooms. This is done by means of three 9 kW data coolers, with a total 27 kW cooling capacity. Nevertheless, the cooling machine that transfers that heat to the building, has a capacity of 55,7 kW. This over-dimensioning would only make sense if an expansion of the data rooms is expected/planned. With a proper size (36 kW to be able to expand the system to one more data room), the COP would be 7,92 % higher.

Several changes were proposed to adapt better to the building's characteristics. However, since some of the alternatives cannot be implemented simultaneously, a full new energy system was proposed in Section 6.2.3, summarized in Table 6.6.

Chapter ***8***

Further work

This thesis is focused on evaluating the energy system as it is nowadays, and on proposing alternatives to the current system to either solve current problems or improve the performance of the building. However, due to the amount of data gathered and studied, the cost and feasibility evaluation of those proposals could not be carried out. Since some of the proposals are for the current building, and some for an ideal building which would be built nowadays, an analysis of this kind would shed some light on what is actually economically wise.

In order to better evaluate the different proposals, a SIMIEN simulation of the building would be of use. A deep research should be carried out in order to determine all the parameters missing for the simulation. It would also be of interest, in order to develop the feasibility study, a SIMIEN simulation of the different proposals, combining them to obtain different configurations and compare them.

If more data could be recovered, it would be interesting to develop a simulation model, in EES or Dymola (based on Modellica). Missing data could be supposed after a thorough research. These models would help characterise the system and would be of use when proposing improvements of the cycle.

It would be also interesting to investigate the building further by interviewing the workers that took part in the design or by recovering more documents. The source of some problems was not discovered (such as the overuse of electricity for the DHW distribution system, or the control problems between the heat pumps and the electric boiler), and it might be helpful for further analysis.

As mentioned previously, one of the objectives of the student was to study the exergy destruction and exergy efficiencies, to offer a further analysis of the components. Nevertheless, due to many difficulties explained in Section 5.3, this analysis could not be performed. It would be of interest to analyse the provided data, and treat it, so meaningful results can be obtained.

Chapter 9

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Appendix ***A***

Vapor Compression Cycle Improvements

Sometimes, the designed cycle does not fit the requirements of the process, or the performance obtained is not enough, some changes/additions to the cycle can be made to further benefit from it.

Subcooling

In order to lower the temperature of the refrigerant even more after having gone through the condenser, a subcooling cycle can be installed. This modification can lead to an improvement of the performance because the temperature at which the fluid is expanded lowers, so the entropy difference is smaller. When heat rejected to the medium is not enough, or when different spaces/mediums want to be heated at different temperatures, an addition of a subcooler and even a gas-cooler is a good choice. For example, the condenser could take 80 % of the heat, and the subcooler 20 %.

However, in cold climates, too low condenser exit temperature can be an issue. This can be solved by bypassing the subcooler when needed.

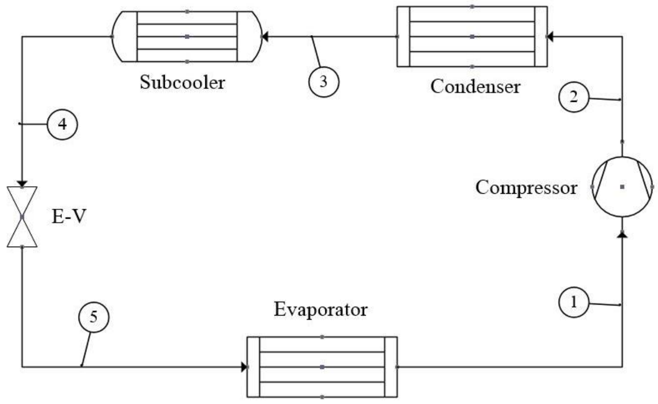

Figure (Appendix) 1 Subcooler in a vapor compression cycle. Here, only the refrigerant cycle has been represented

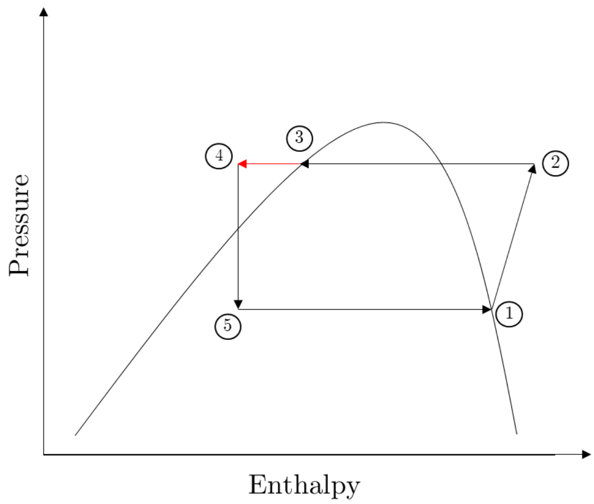

Figure (Appendix) 2 P-h diagram of the vapor compression cycle with subcooling (marked in red)

Internal Heat Exchanger

In order to both subcool the fluid at the condenser exit and superheat the fluid at the evaporator exit, an internal heat exchanger can be installed. By doing this, the expansion loss will be reduced. However, this is not always a benefit for the system, because the inlet temperature of the compressor will increase, increasing its specific volume and thus the compressor work.

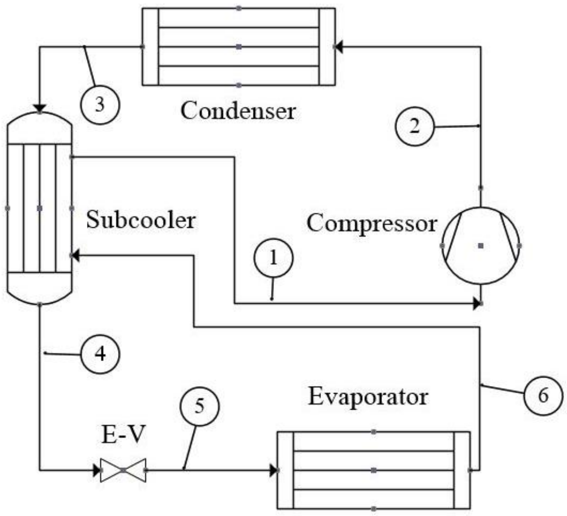

Figure (Appendix) 3 Internal Heat Exchanger.

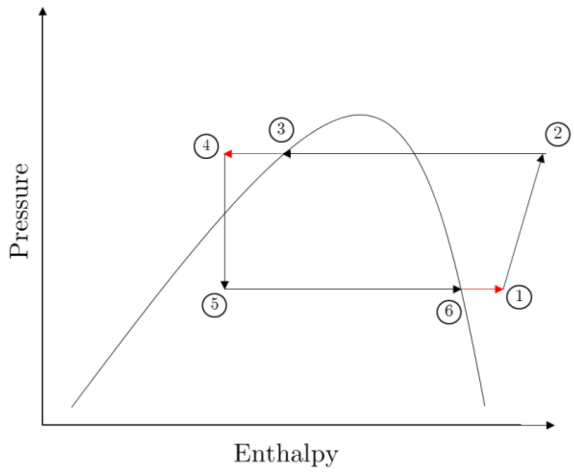

Figure (Appendix) 4 4 Internal Heat Exchanger cycle. Marked in red the subcooling and superheating influence

Staged Compression

Other problems that heat pumps may encounter, is excessive temperature at the exit of the compressor. This occurs mainly when using refrigerants with lower molar capacity. High temperatures after compression, can lead to damages in the lubricating oil and eventually, a failure of the device [36]. To avoid this, several compressors can be installed in series. Refrigerant would enter the low temperature (LT) compressor, then into a de-superheater in order to lower its temperature. Then it would enter the high temperature (HT) compressor to further get compressed until the desired pressure. This kind of systems are broadly used for refrigeration purposes because of the possibility of having two different temperature evaporators (one for chilling and one for freezing, for example).

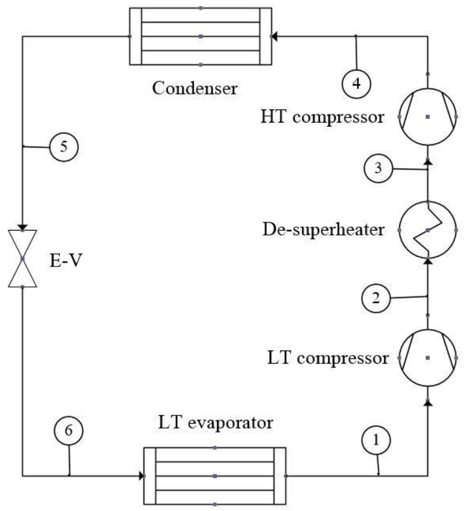

Figure (Appendix) 5 Staged compression in a vapor compression cycle

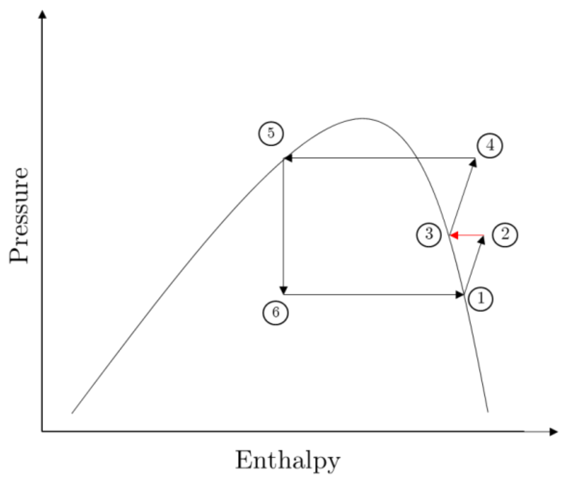

Figure (Appendix) 6 Staged compression P-h diagram. Marked in red the LT de-superheater.

Ejector System

In order to recover part of the work lost in expansion valves, an ejector system can be used. This device allows to trade high-pressure for high-speed, lowering its pressure below the one of the flows being introduced from the evaporator. By doing so, it creates a sucking effect, and after mixing with the main fluid, it increases its pressure. This procedure reduces the amount of work required of the compressor, lowering the electric energy demand.

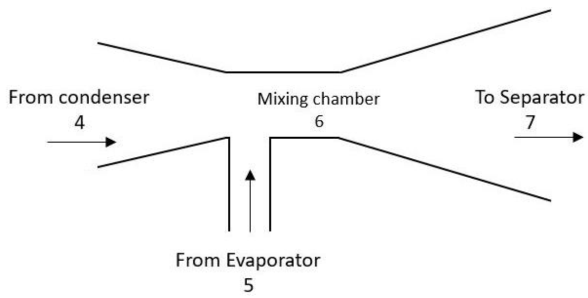

Figure (Appendix) 7 Simplified ejector diagram.

In Figure (Appendix) 7 refrigerant from the condenser enters the ejector (4), sucks gas from the evaporator (5) and gets mixed while increasing its pressure (7). An auxiliary compressor could be added in parallel in order to make the most of the pressure recovered with the ejector.

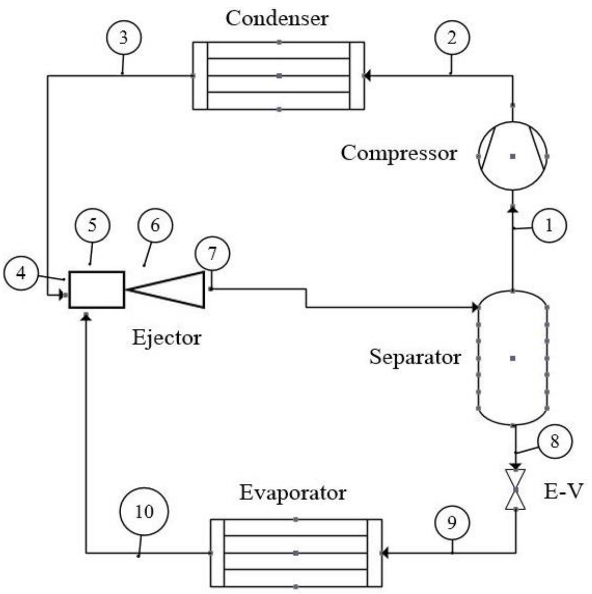

Figure (Appendix) 8 Ejector system diagram. Points 4,5,6 and 7 correspond to inlet from condenser and evaporator, mixing chamber and exit respectfully.

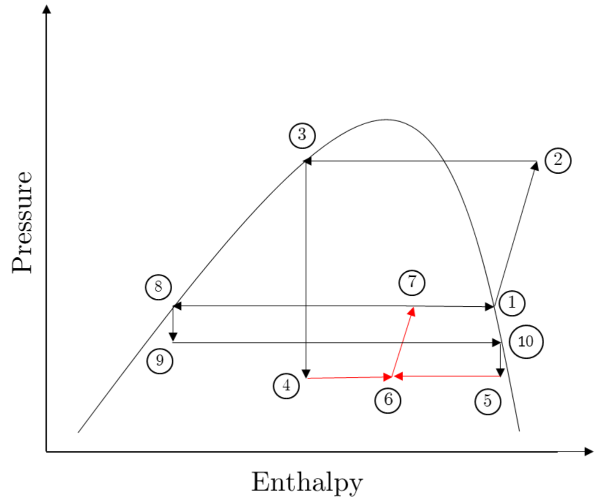

Figure (Appendix) 9 Ejector cycle diagram. Marked in red is the ejector's influence.

Economizer

One can take advantage of the expansion work lost in the expansion valve, by preheating the flow between the two compressors. By increasing its temperature, its pressure will increase as well. This is done by using a subcooler at the exit of the condenser.

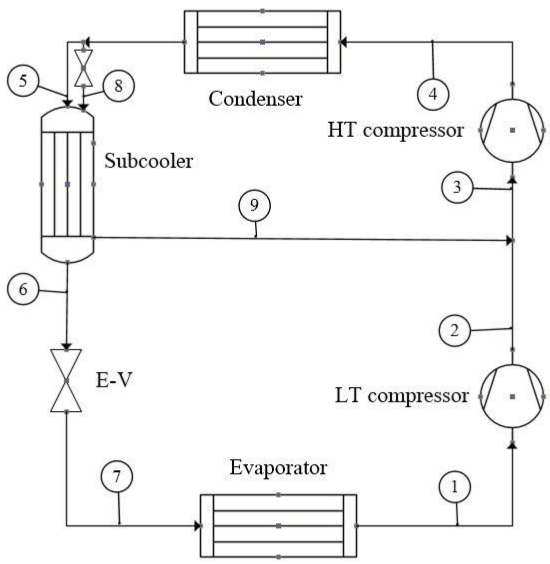

Figure (Appendix) 10 Economizer solution in a two-stage compression cycle

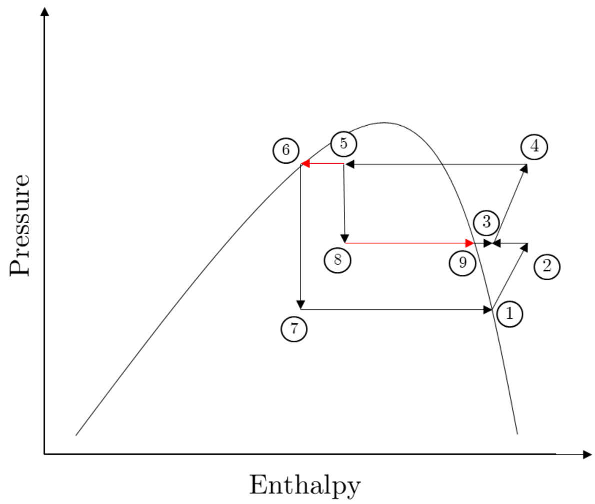

Figure (Appendix) 11 Economizer cycle. Marked in red, the economizer's influence

Table (Appendix) 1 Summary of the different upgrades.

Upgrade	**Main features**	**Consequences**
Subcooler	Further condensation/cooling of the refrigerant	Lower return temperature → Better COP
Internal Heat Exchanger	Superheat fluid after evaporator	Lower expansion loss Higher specific volume → higher compression work
Staged Compression	Lower temperature after compression	Less problems with lubricating oil Lower total compression work
Economizer	Maintain high pressure before expansion	Lower expansion loss
Ejector	Use the expansion work after the condenser	Less compression work needed

Appendix **B**

Useful Graphs from 2019

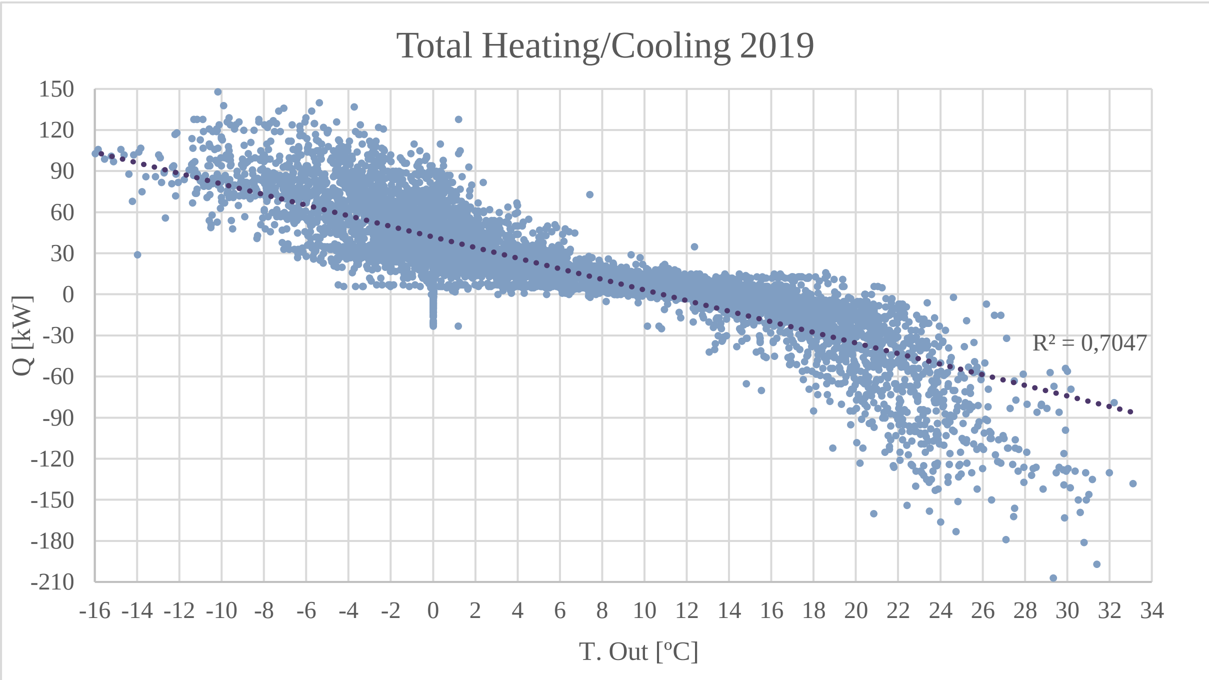

Figure (Appendix) 12 Total heating and cooling load from 2019 vs. ambient temperature. In dark blue, the trendline from the points [88].

From Figure (Appendix) 12 one can see that the correlation between total heating/cooling and outside temperature follows a linear equation which is acceptable and expected. When the outside temperature is lower, a higher heating load is expected, and when the temperature outside is higher, a higher cooling load.

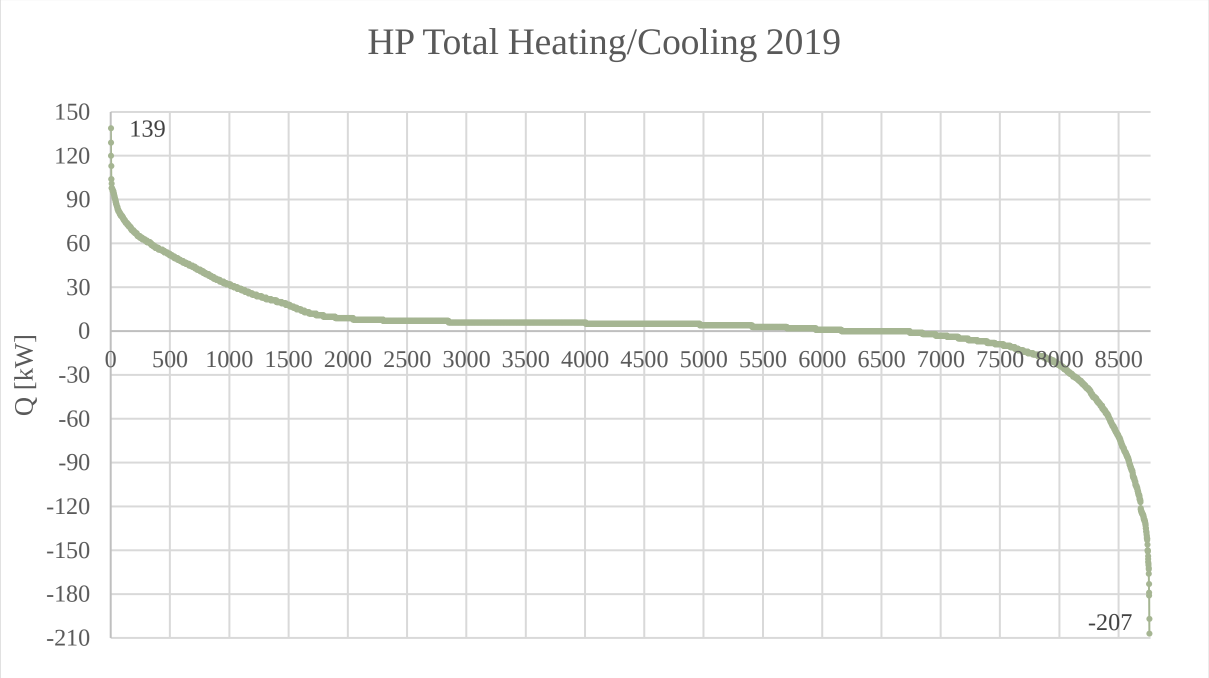

Figure (Appendix) 13 Total heating and cooling provided by the heat pump [88].

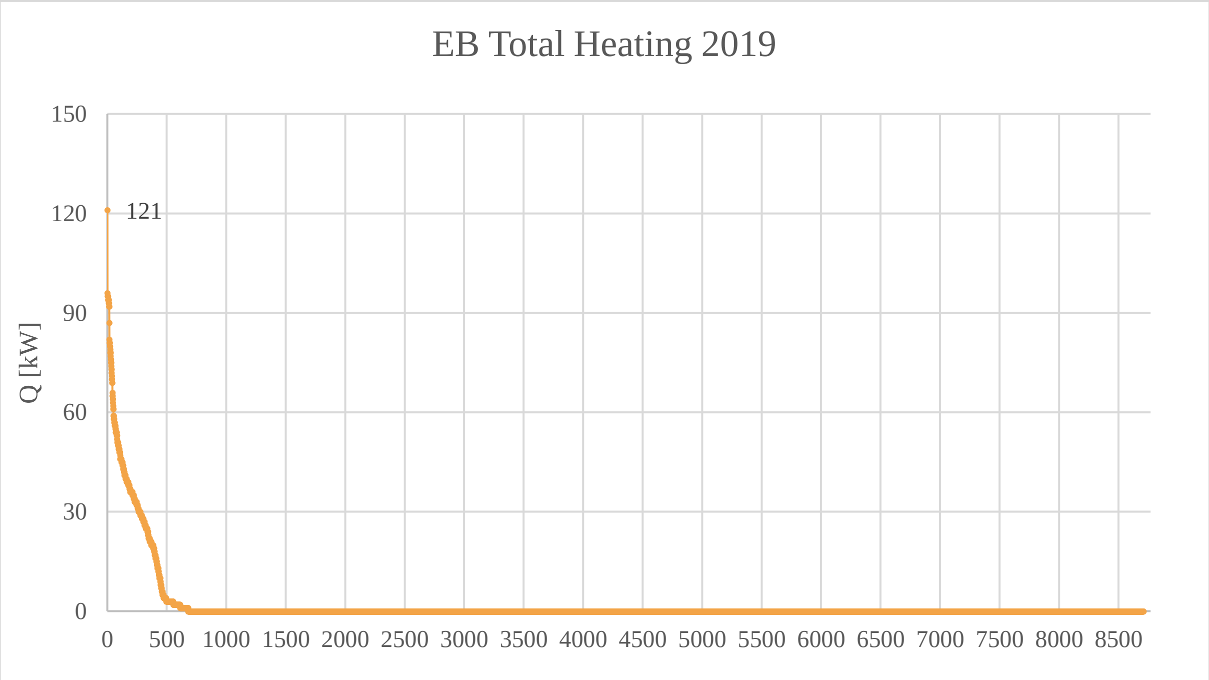

Figure (Appendix) 14 Total heating provided by the electrical boiler [88].

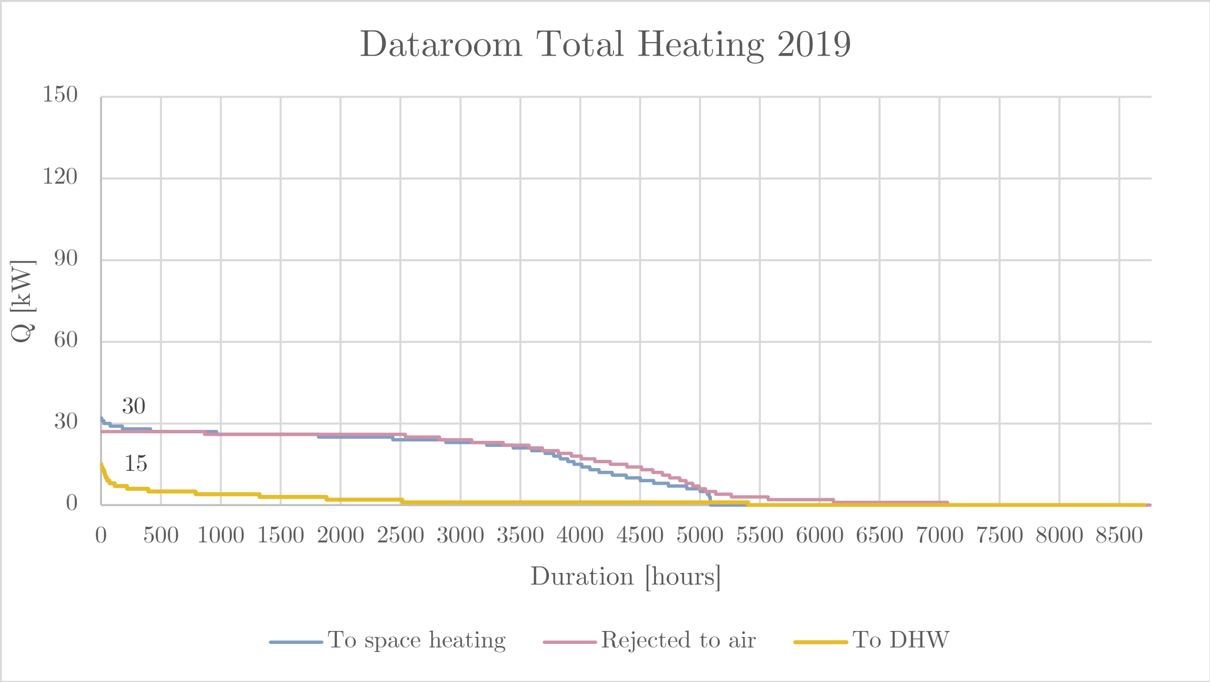

Figure (Appendix) 15 Total space heating provided by the heat extracted from data rooms, heat rejected to air and heat provided to the DHW system [88].

Appendix ***C***

Temperature Sensor data from 2019

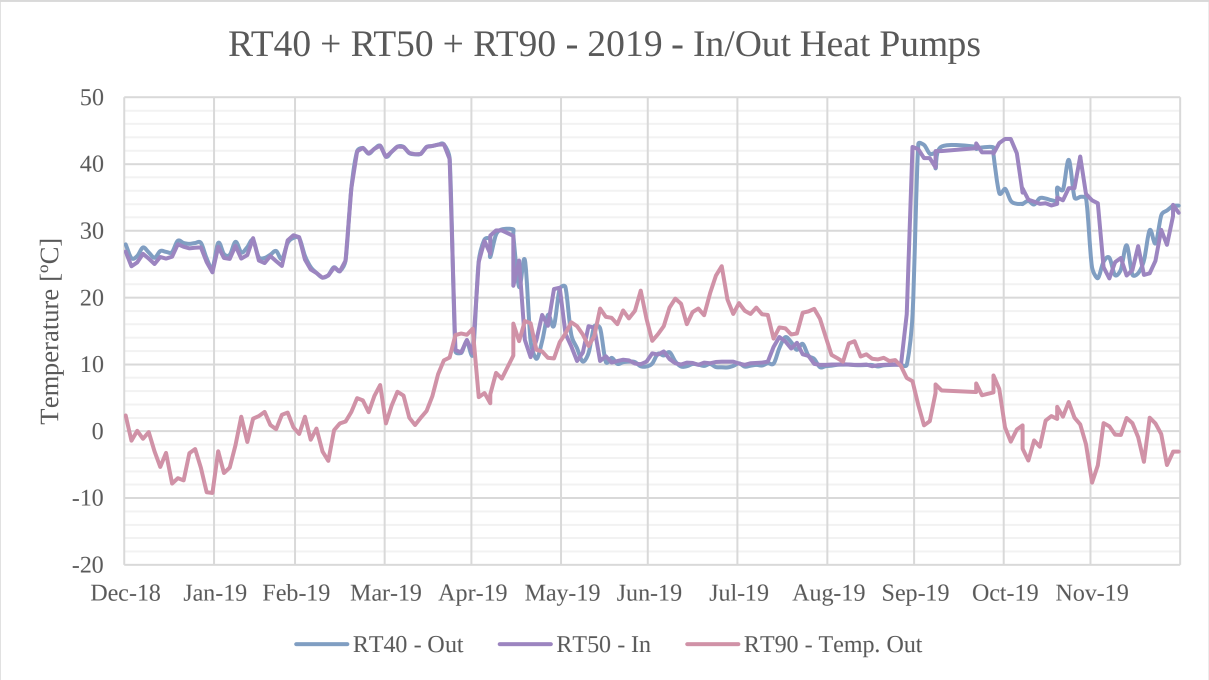

Figure (Appendix) 16 Comparison between the temperature outside and the temperatures in and out of the heat pumps [88].

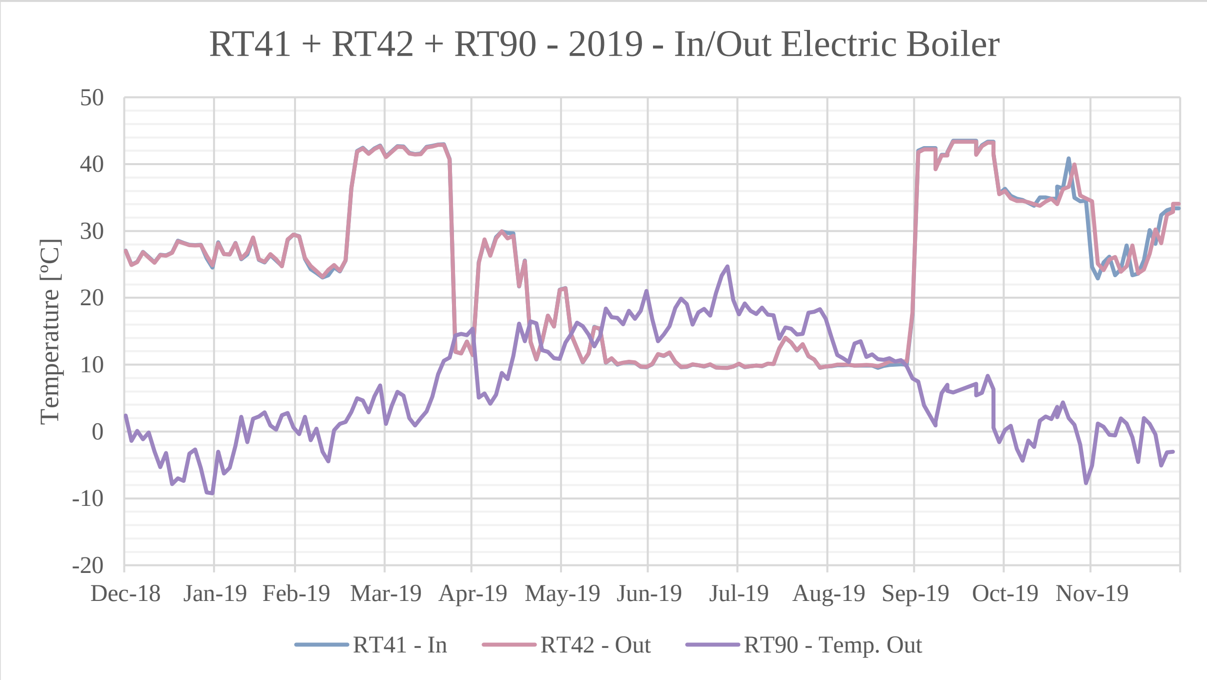

Figure (Appendix) 17 Comparison between the temperature outside and the temperatures in/out of the EB [88].

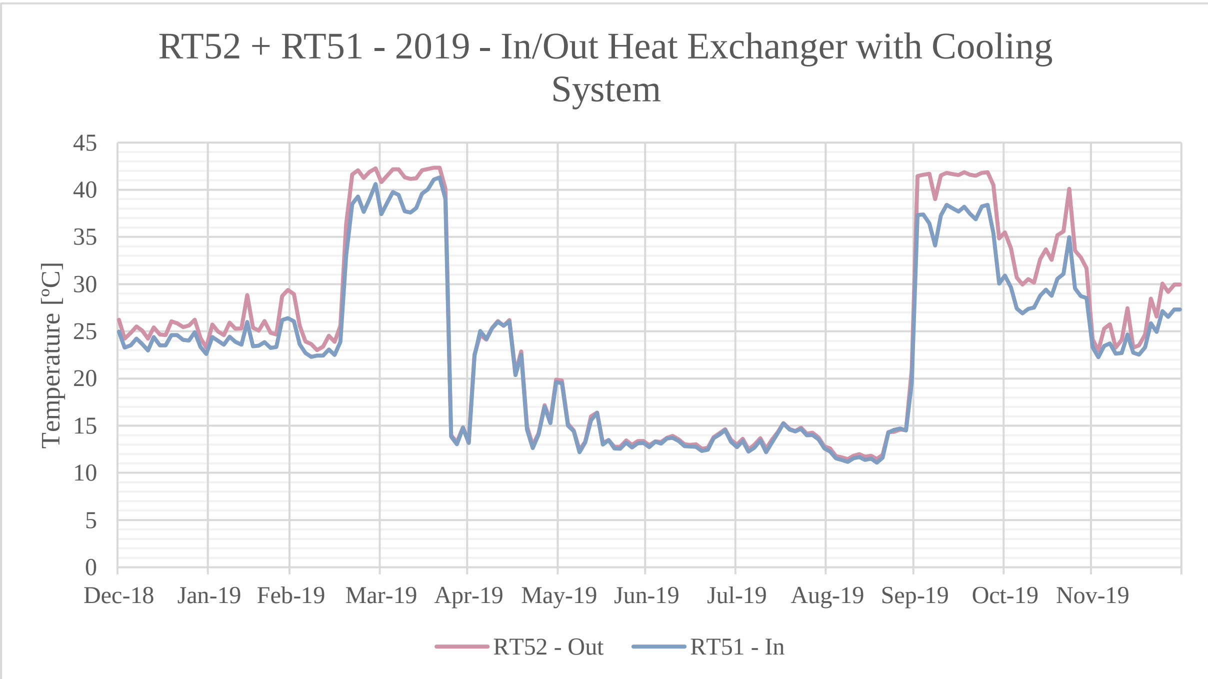

Figure (Appendix) 18 Comparison between the temperatures in and out of the heat exchanger that connects with the cooling system [88].

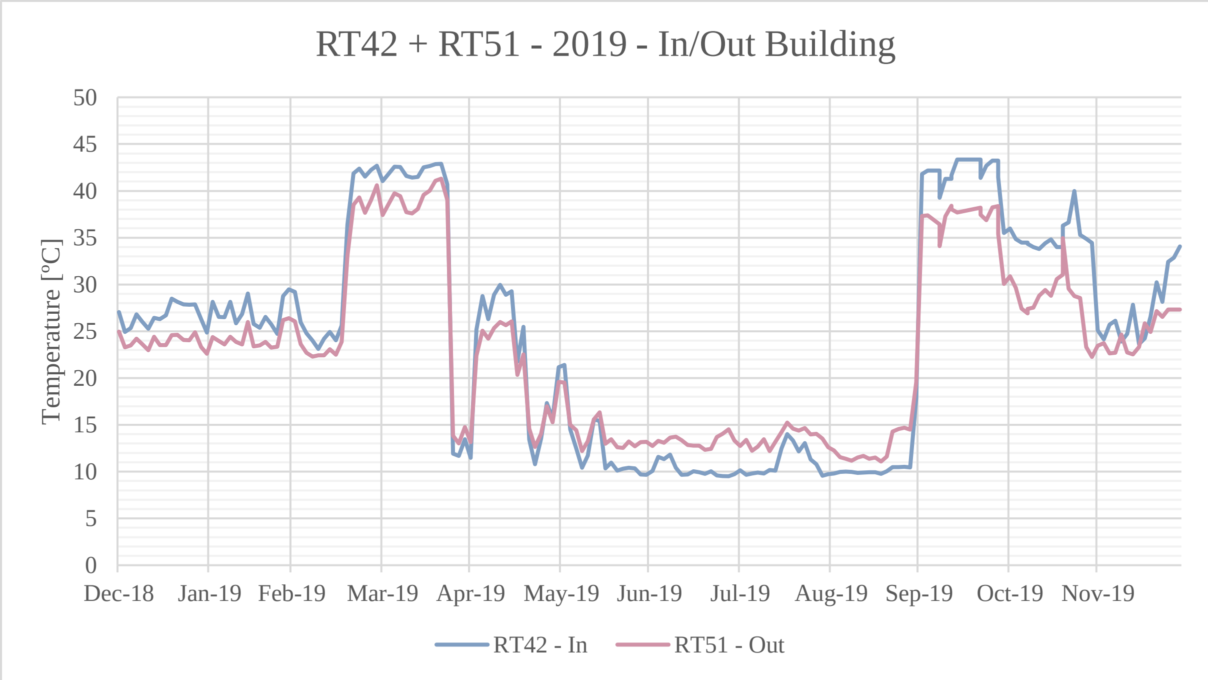

Figure (Appendix) 19 Comparison between the temperatures to and from the building [88].

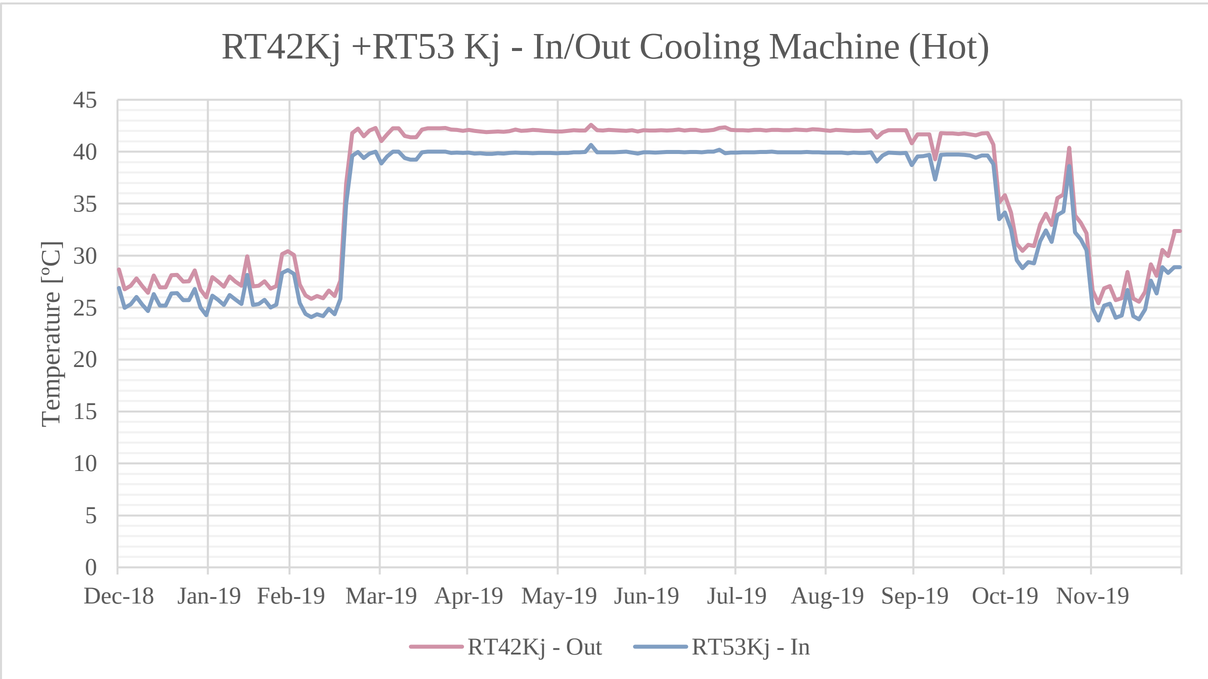

Figure (Appendix) 20 Comparison between the temperatures in and out of the system in the hot side [88].

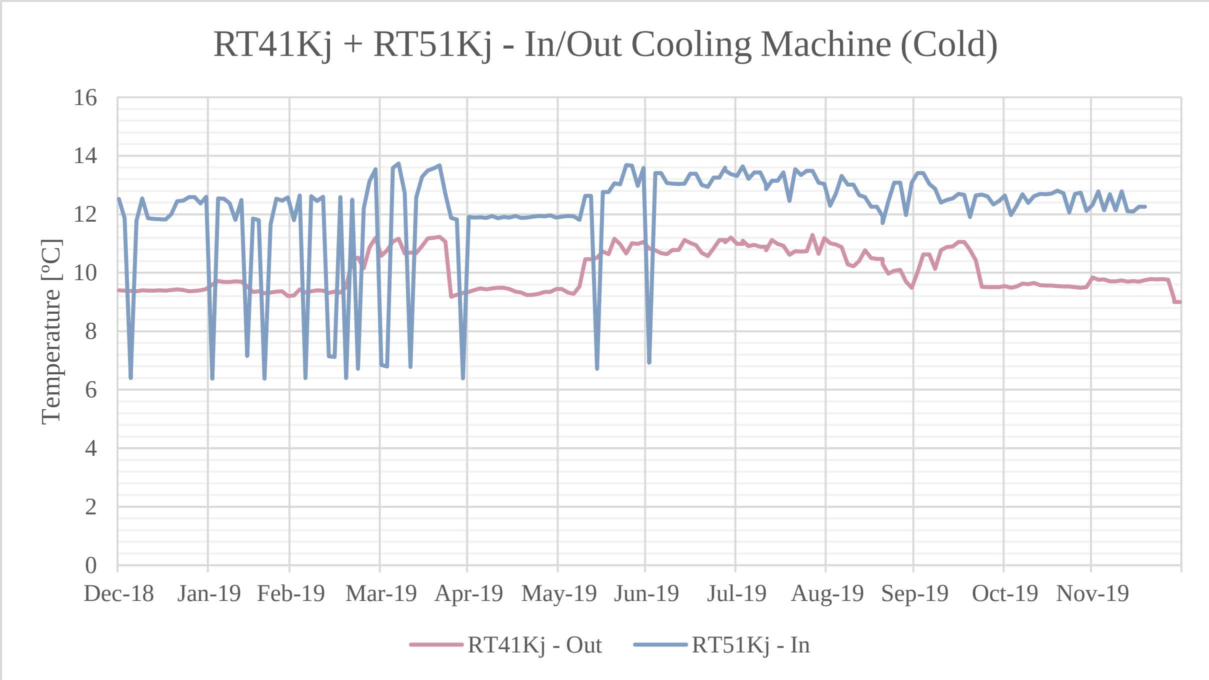

Figure (Appendix) 21 Comparison between the temperatures in and out of the system in the cold side [88].

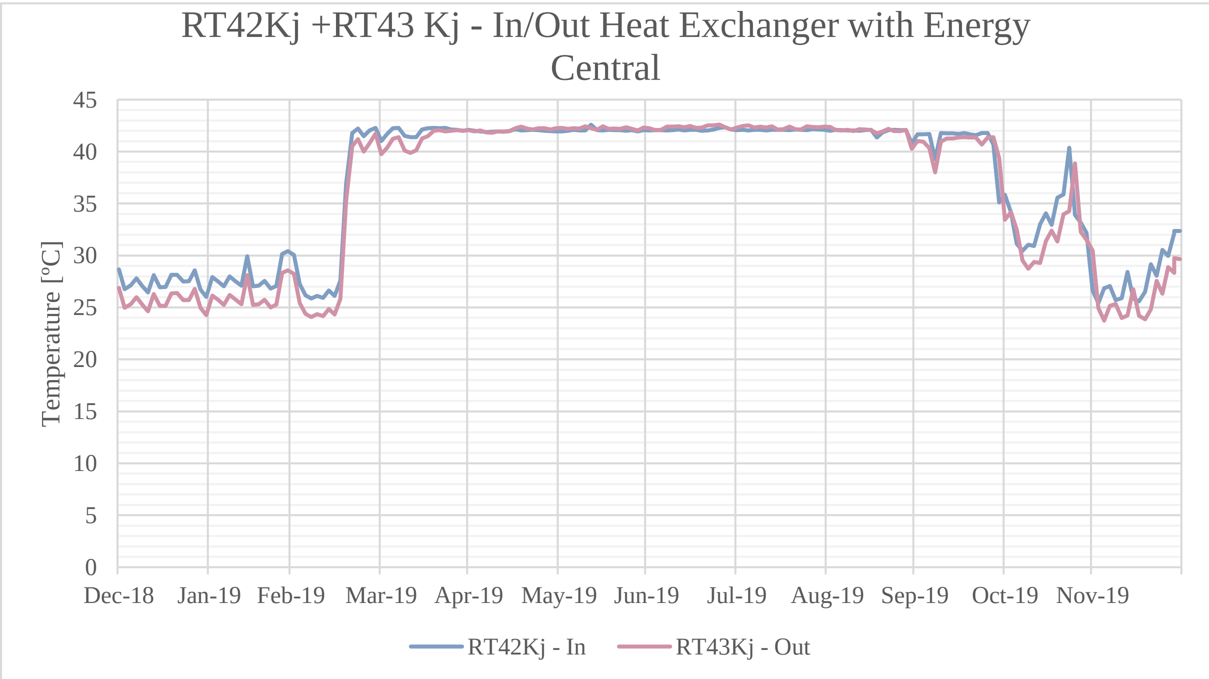

Figure (Appendix) 22 Comparison between the temperatures in and out of the heat exchanger that connects with the energy central [88].

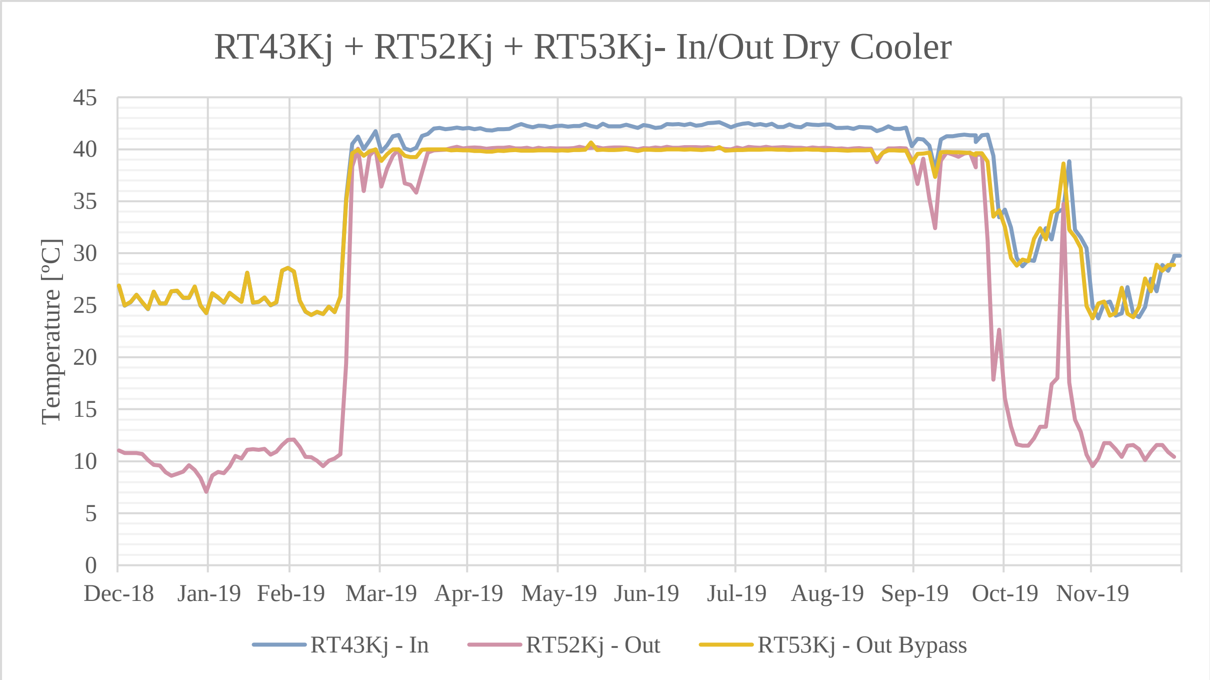

Figure (Appendix) 23 Comparison between the temperatures in and out of the dry cooler [88].

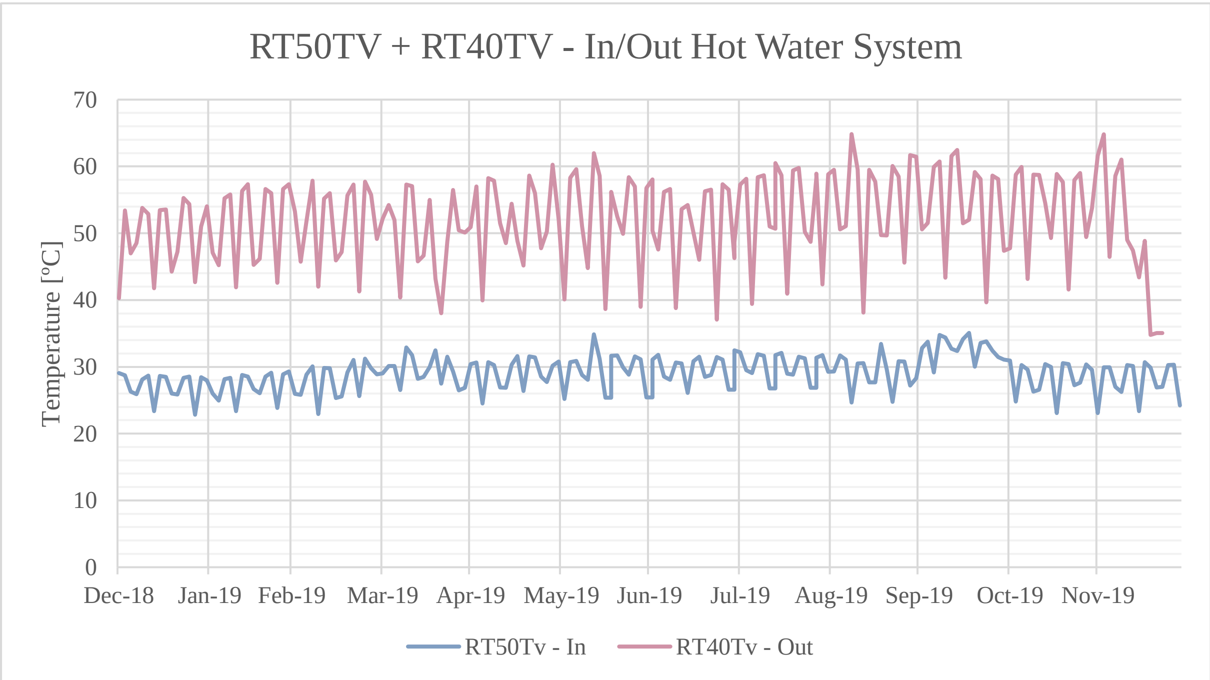

Figure (Appendix) 24 Comparison between the temperatures in and out of the hot water system [88].

Appendix D

Hourly analysis from 2019

The daily analysis of the four randomly chosen days shows several points worth mentioning.

Hourly analysis, regular winter day (27/02/2019)

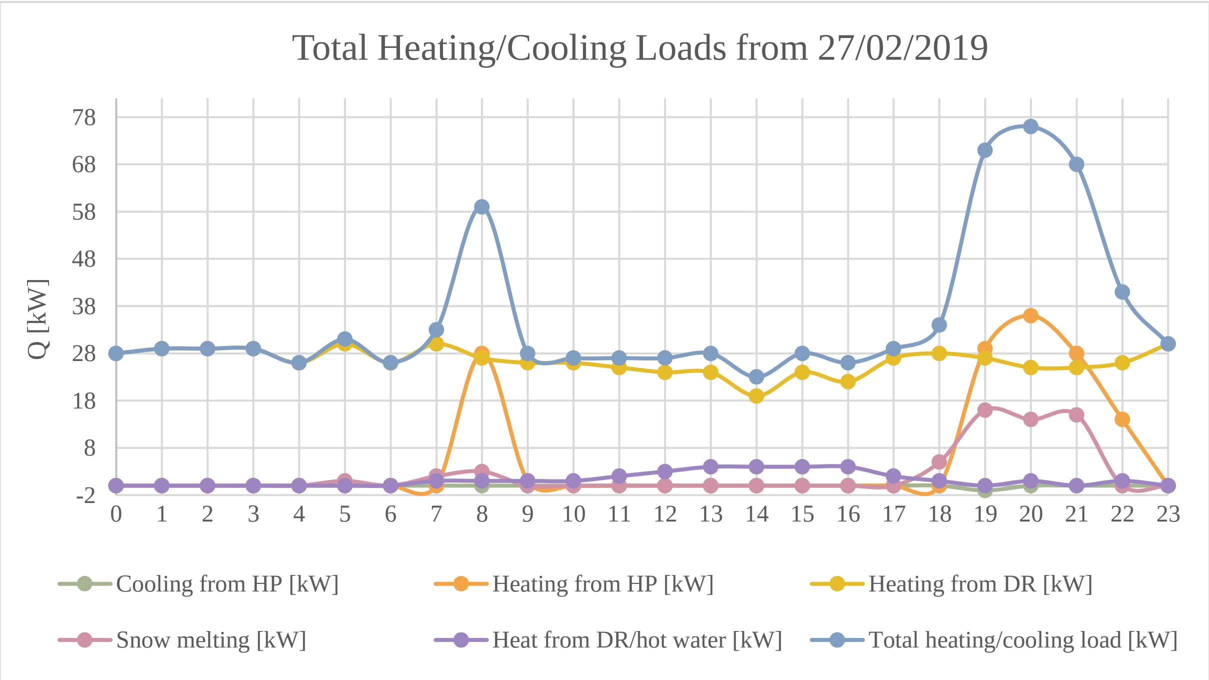

Figure (Appendix) 25 Hourly heating/cooling loads from a regular winter day in 2019 (27/02/2019) [88].

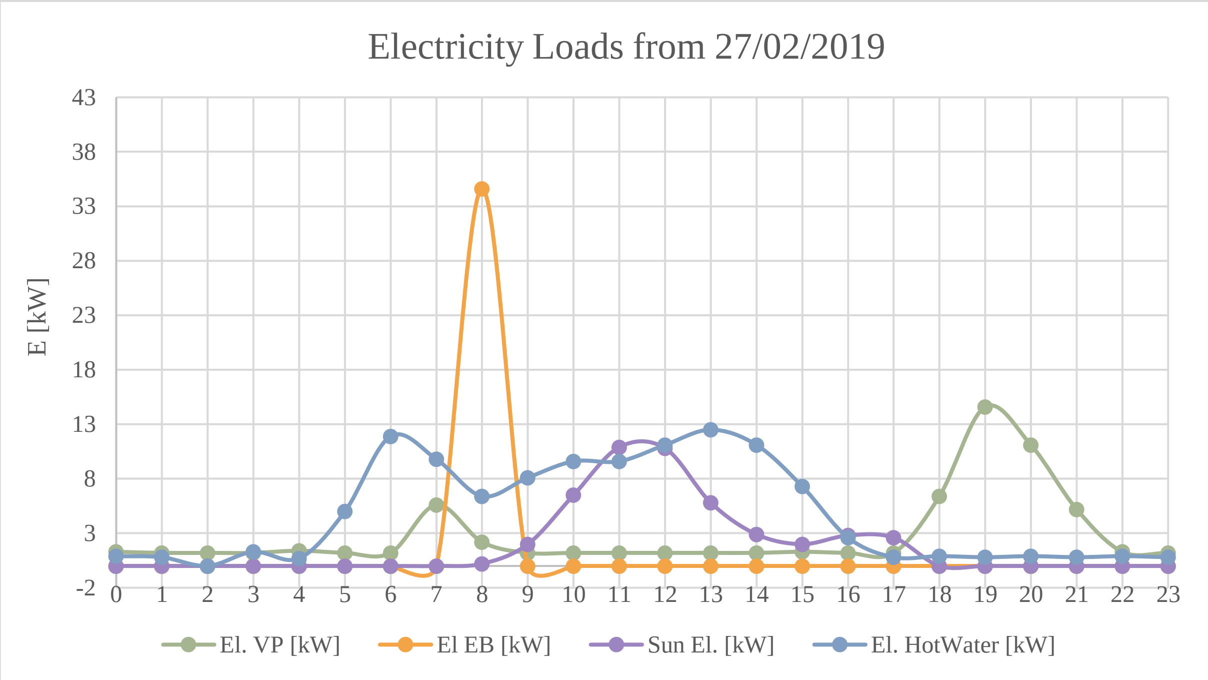

Figure (Appendix) 26 Hourly electricity load from the heating system from a regular winter day in 2019 (27/02/2019) [88].

The first studied day was the 27th of February of 2019, which with an average temperature of 1 ºC, is the coldest of the four days studied. In Figure (Appendix) 25 one can see that the base heating load (~30 kW) is covered by heat recovered from the data rooms. At 8.00 and at 20.00 an extra heat addition was required (up to 58 kW at 8.00 and 78 kW at 20.00) and therefore were the heat pumps switched on, as well as the electric boiler with 35 kW (Figure (Appendix) 26). When heat from the data rooms was not needed in the building, it was rejected to hot water (a total of 30 kWh/day). At the same period as when the extra heat was needed, was some heat delivered to the snow melting system. During the day, there was no cooling needed in the building. The total heat transfer from data rooms to the building, hot water system and snow melting system was 718 kWh/day. The electricity used to heat DHW, was around 8 kW during the period when the building was occupied.

Hourly analysis, regular spring day (24/04/2019)

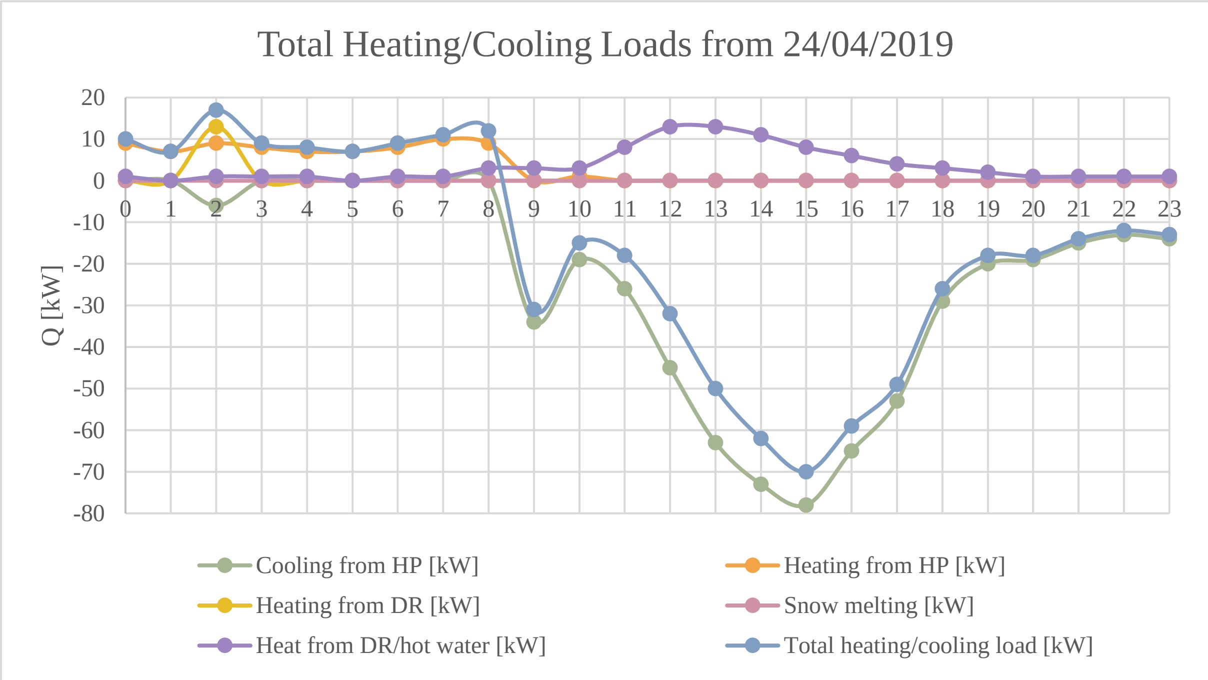

Figure (Appendix) 27 Hourly heating/cooling loads from a regular spring day in 2019 (24/04/2019) [88]

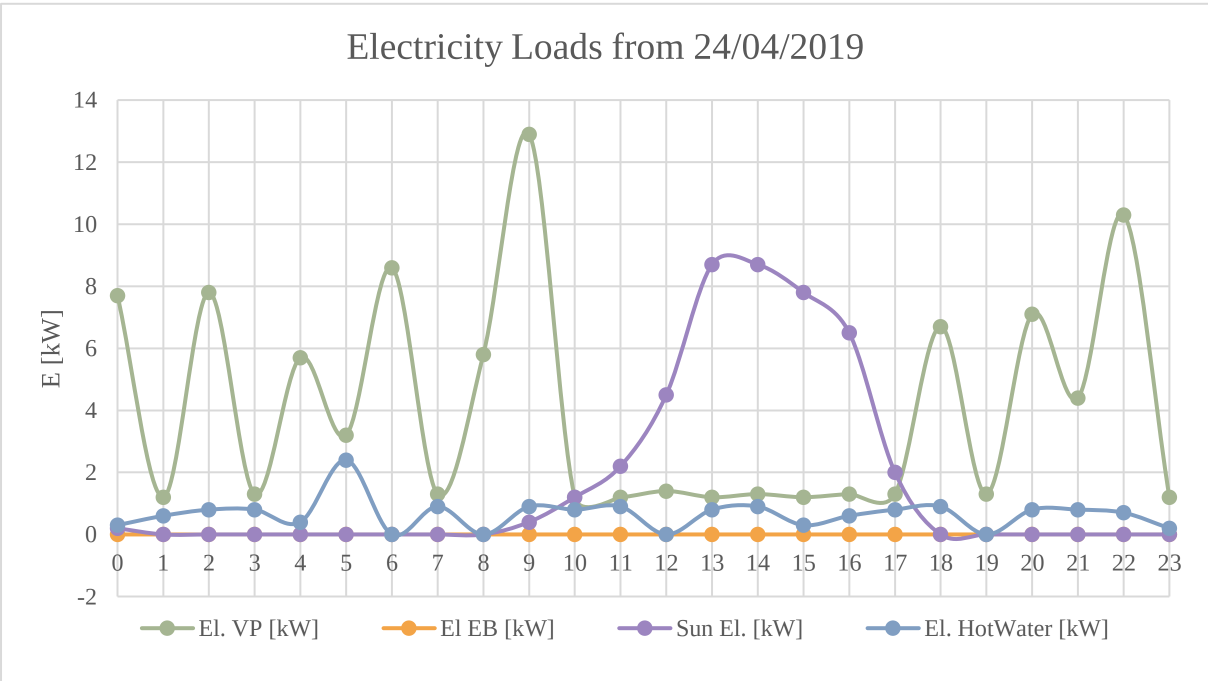

Figure (Appendix) 28 Hourly electricity loads from the heating system from a regular spring day in 2019 (24/04/2019) [88].

The second studied day was the 24th of April of 2019. The average temperature was almost 14 ºC, reaching 20 ºC at 14.00. It comes as no surprise, that the highest cooling load is around this time (see Figure (Appendix) 27). However, in the early hours of the day, some heating was required. Surprisingly was heat from the data rooms not used for this purpose, but instead where the heat pumps switched on. Heat from data rooms was rejected to ambient air through the dry cooler (temperature sensor RT52Kj shows a temperature of ~40 ºC during that day). This heat could have been instead used for heating the building or preheating water, so some electricity on the heating tank could have been saved (~116 kWh in electricity during the day used for heating water). The total heat transfer from the data rooms to the building (13 kWh/day) and hot water system (87 kWh/day) was 100 kWh/day. During this period, the electric boiler was not turned on. The electricity used to heat DHW is surprisingly low, being it a regular day with regular loads.

Hourly analysis, regular summer day (26/06/2019)

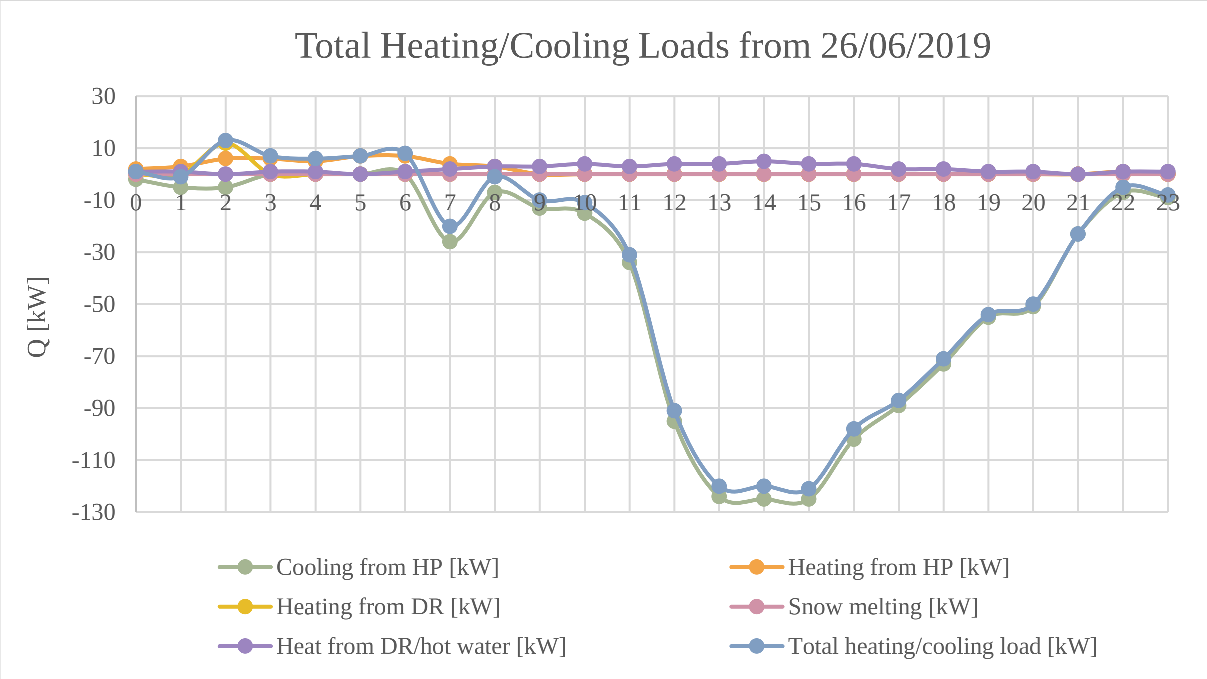

Figure (Appendix) 29 Hourly heating/cooling loads from a regular summer day in 2019 (26/06/2019) [88]

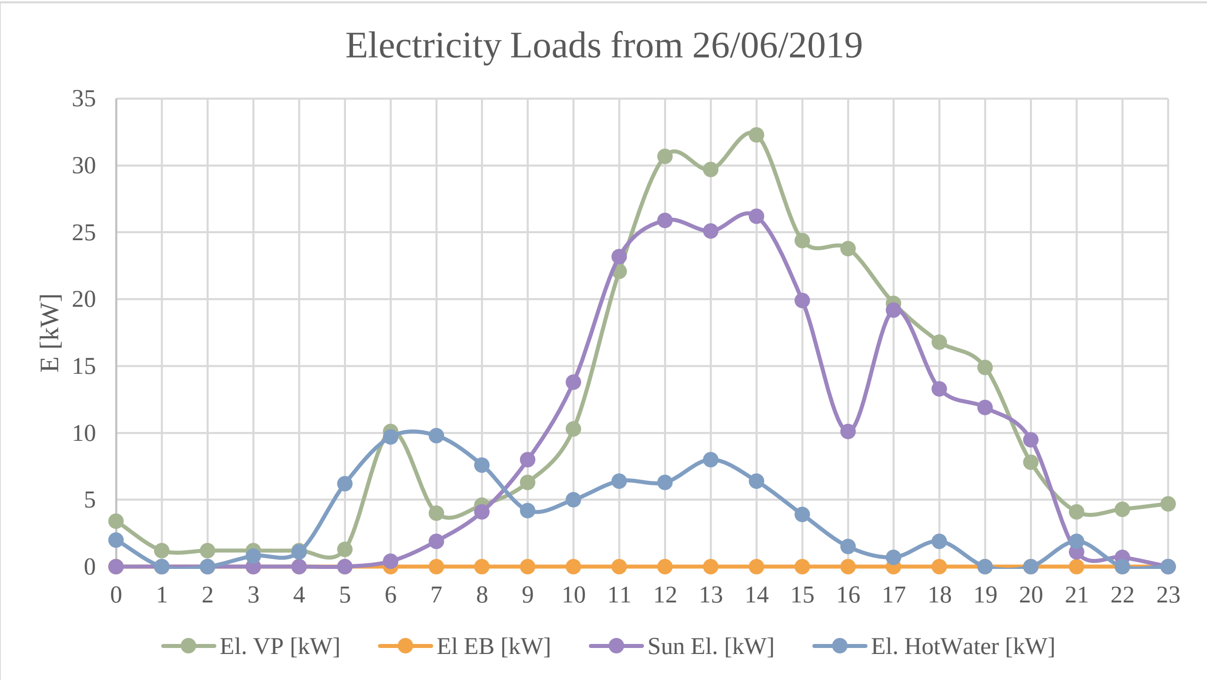

Figure (Appendix) 30 Hourly electricity loads from the heating system from a regular summer day in 2019 (26/06/2019) [88].

The third day studied was the 26th of June of 2019. The average temperature of the day was 17,5 ºC with a peak of 25 ºC around 19.00. From Figure (Appendix) 29 one can see that the behaviour of the energy central is similar to the one in April. Nevertheless, less electricity was used in the heating of water (~83 kWh/day), although it was not because heat from data rooms was used (just 49 kWh/day). Most likely was the hot water demand lower that the one in April and therefore was less heat needed. The dry cooler was on this day as well and all heat from the data rooms was rejected through this equipment. The total heat transferred from the data rooms to the building was 12 kWh/day and 49 kWh/day to the hot water system. One can see in Figure (Appendix) 30 that the solar production almost covered the electricity used for the heat pumps. The electricity used to heat DHW is similar to the pattern seen in Figure (Appendix) 26.

Hourly analysis, regular autumn day (9/10/2019)

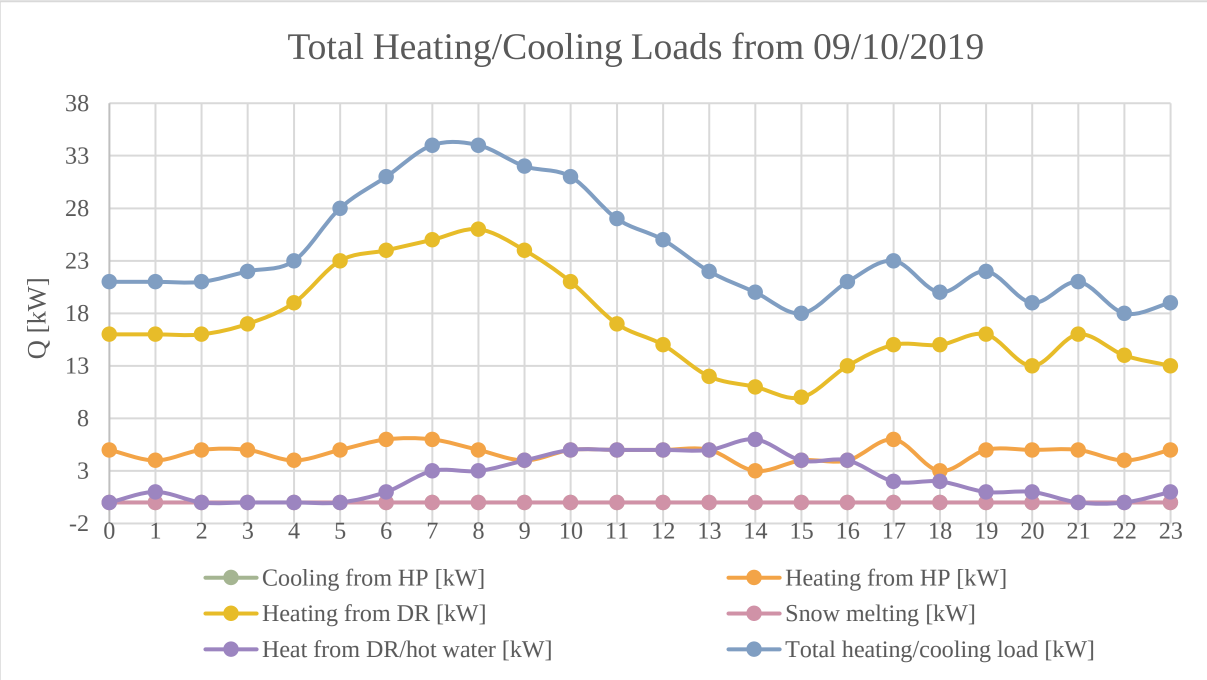

Figure (Appendix) 31 Hourly heating/cooling loads from a regular autumn day (09/10/2019) [88].

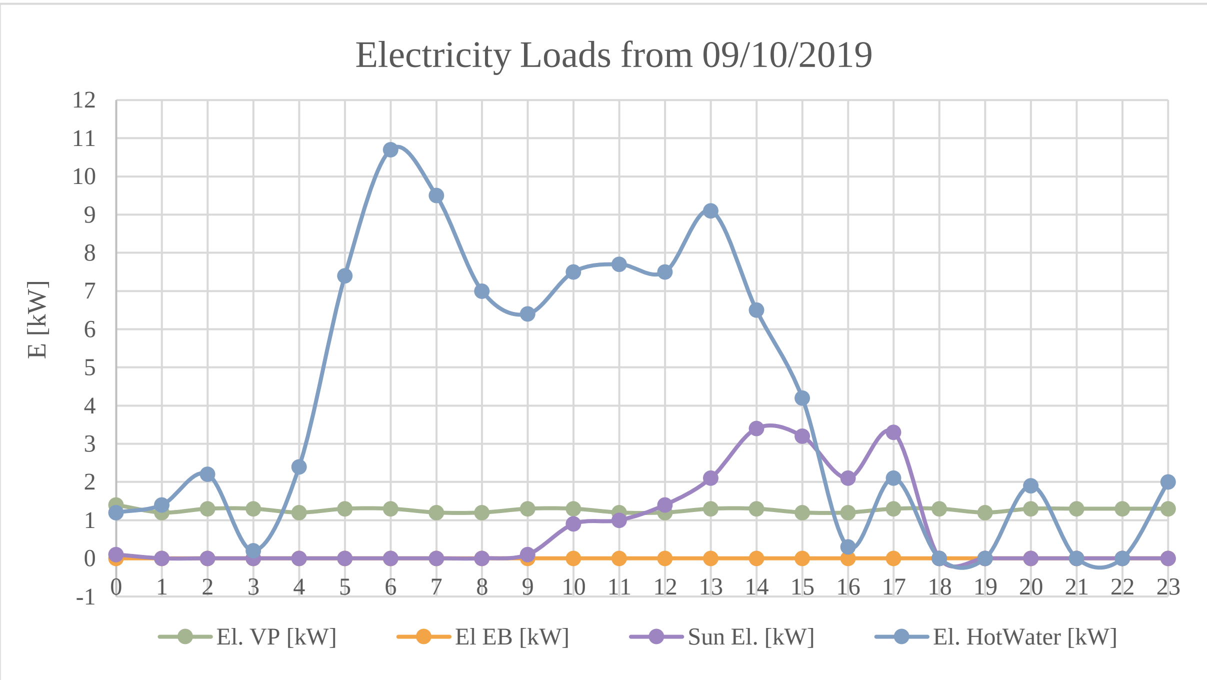

Figure (Appendix) 32 Hourly electricity loads from the heating system from a regular autumn day (09/10/2019) [88].

The last day analysed was the 9th of October of 2019. The average temperature was around 4 ºC, more or less constant during the day. Heat from data rooms was used throughout the day to heat the building, as well as heat pumps (see Figure (Appendix) 31). The heating load was a bit higher during peak hours in the morning, but in general it was more or less constant. The heat contribution of heat pumps is also fairly constant. The total heat recovered from data rooms was 460 kWh/day (407 kWh/day for building heating and 53 kWh/day for water preheating). Some heat was rejected as well through the dry cooler. The electricity used to heat DHW is more or less the same as in Figure (Appendix) 26 and Figure (Appendix) 30, being slightly higher and with peaks during the night.

That one may deeper understand the function of the system, an analysis of both the two coldest and the two warmest days of the year will be carried out.

Hourly analysis, coldest two days of the year (29-30/01/2019)

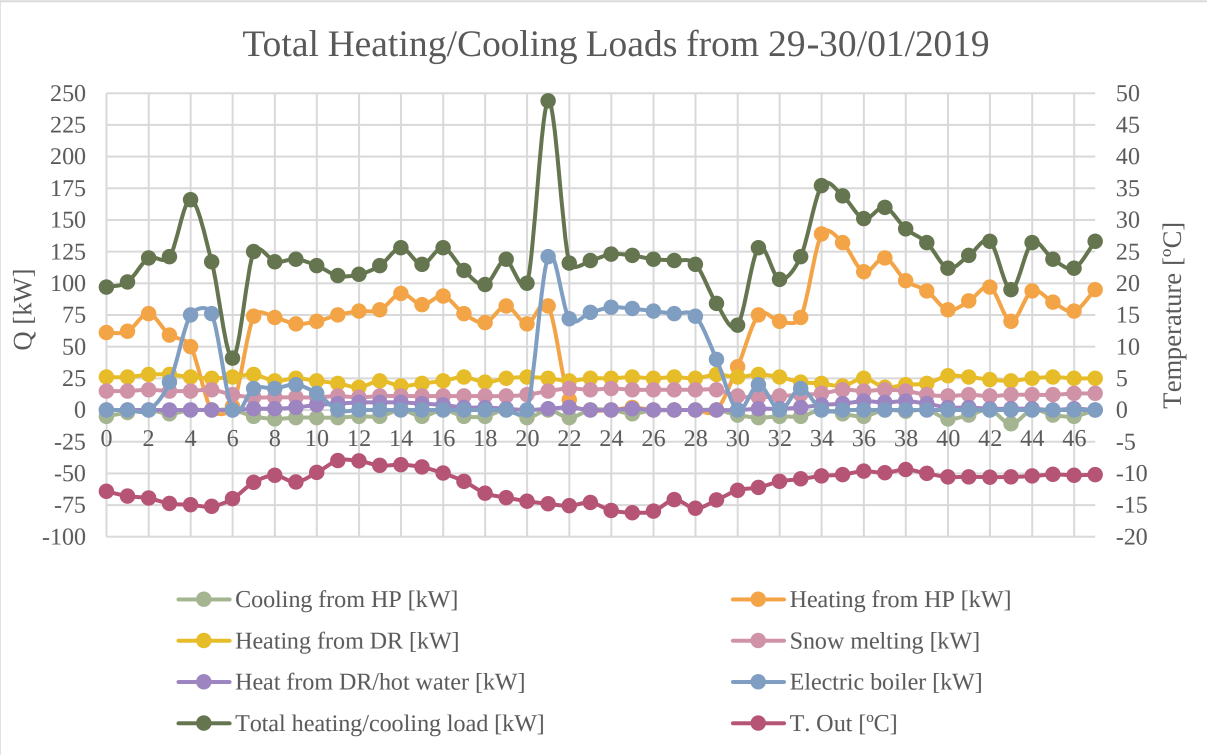

Figure (Appendix) 33 Hourly heating and cooling loads from the coldest days of 2019 (29-30/01/2019) [88].

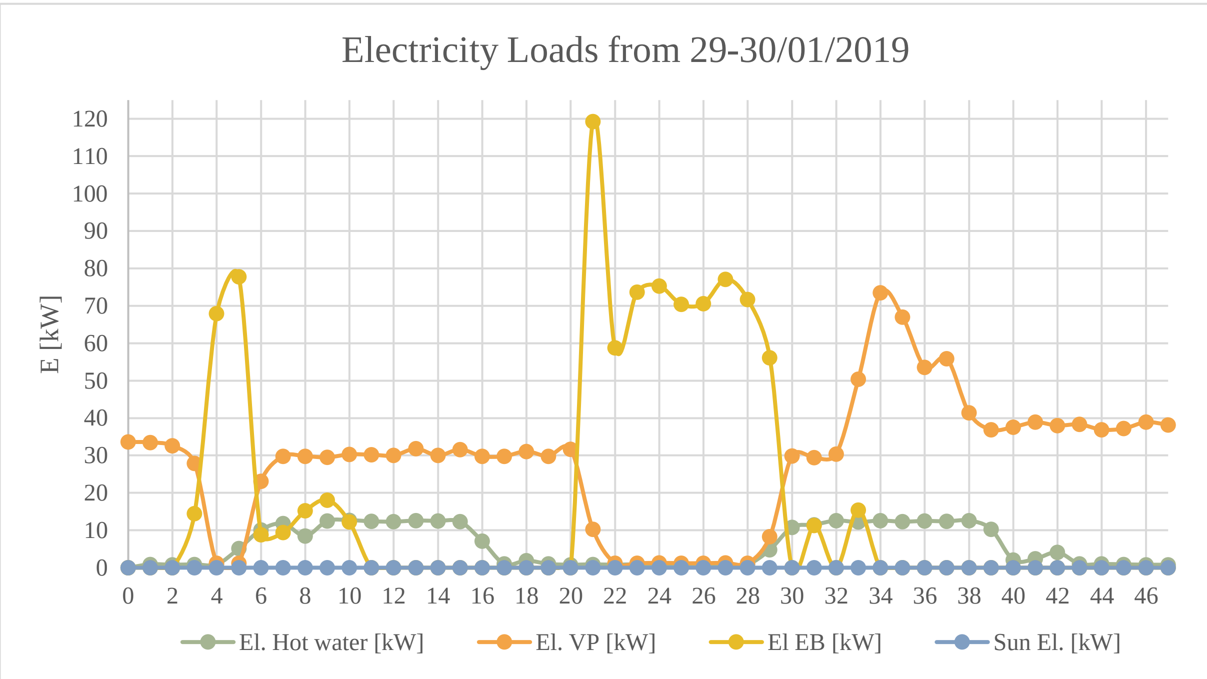

Figure (Appendix) 34 Hourly electricity loads from the heating system from the two coldest days of 2019 (29-30/01/2019) [88].

In Figure (Appendix) 33 one can see the heating and cooling loads from the two coldest days of 2019, 29th and 30th of January. The average temperature for these two days was -12 ºC, with peaks of -16,2 ºC. During this period, there was two intervals where the outdoor temperature was below -15 ºC, on the 29th from 3.00 until 5.00 and from 22.00 on the 29th until 4.00 on the 30th. One can see that during these periods the heat pumps were switched off, due to poor efficiency. Instead, the electric boiler was turned on to cover the heating demand (see Figure (Appendix) 34). In order to cover peak loads during early hours of the work schedule, both heat pumps and electric boiler were on. Heat from data rooms was used constantly (~25 kW). Some heat was also used for melting snow and for preheating water, although electricity was mostly used for this purpose (280 kWh in total).

Hourly analysis, warmest two days of the year (28-29/07/2019)

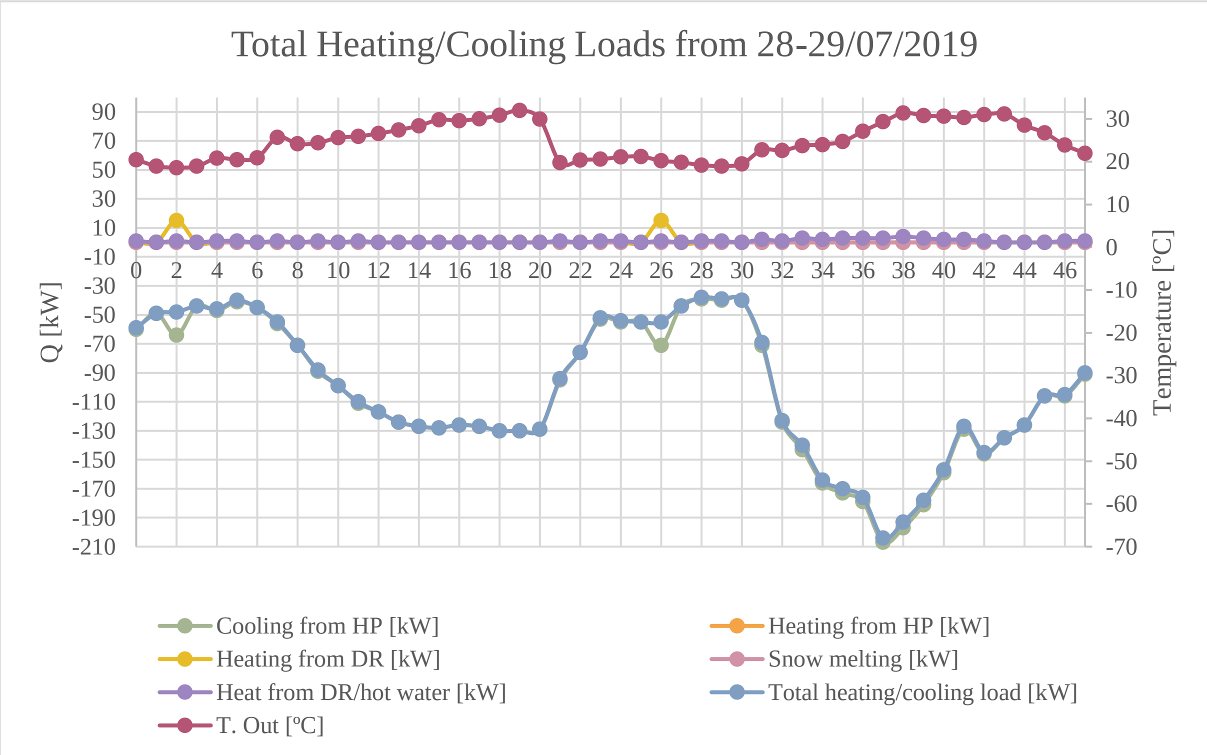

Figure (Appendix) 35 Hourly heating/cooling loads from the two warmest days of 2019 (28-29/07/2019) [88].

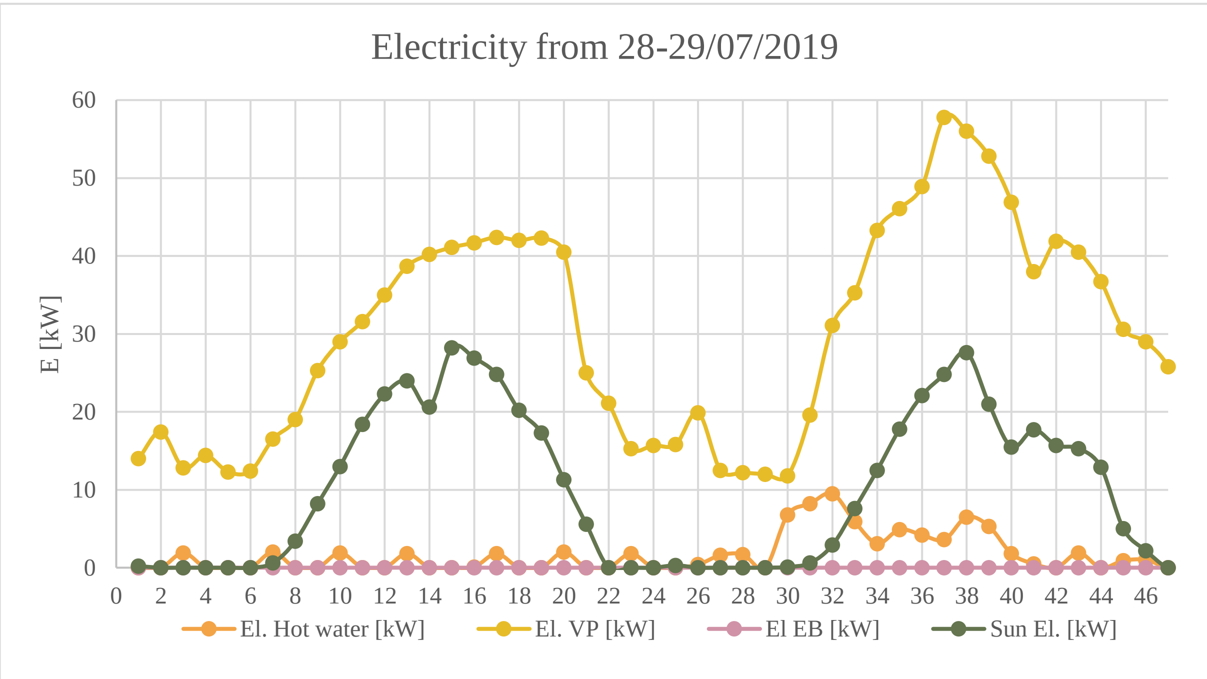

Figure (Appendix) 36 Hourly electricity loads from the two warmest days of 2019 (28-29/07/2019) [88].

Figure (Appendix) 35 shows the two warmest days of 2019, the 28th and 29th of July of 2019. The average temperature was almost 25 ºC, with periods of over 31 ºC. Almost all heat from data rooms was rejected to the ambient with the exception of one heating hour each day (15 kW) and some heat which preheated hot water (44 kWh in total). Hot water was heated by electric boilers with a load of 81,1 kWh in total, which comes as no surprise because hot water demand during this period is supposed to be very low, due to holidays. Both heat pumps worked fully in summer mode during this period. It can be seen in Figure (Appendix) 36 that the solar production follows the pattern of the electrical consumption of the heat pumps.

*Appendix **E***

Characteristics of the components

Dry Cooler

RCWORLD 6.3

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TEAM.MATE.DC.A M.56

KAPASITET	kW	72,2
TØRRKJØLEBATTERI		
Vanntemperatur (inn/ut)	**°C**	**44/38,4**
Vannmengde	m³/h	12
Trykkfall	kPa	55
Frostsikring	%	Eth. Glycol 30%
AKSIALVIFTER	n.	2
Luftmengde	m³/h	19100
Utetemperatur	**°C**	**28**
Effektforbruk	kW	1,4
Maks. strømforbruk FLA	A	6,8
NETTSPENNING	V-ph-Hz	230-1-50
LYDTRYKKNIVÅ Lp 1 m (ISO3744)	dB(A)	69,2
DIMENSJONER		
Lengde	mm	2500
Bredde	mm	658 (H) - 1175 (V)
Høyde	mm	1135 (H) - 1172 (V)
NETTO VEKT	kg	185

TILBEHØR

511 - Bein for vertikal luftstrøm (sett); Automatsikringer;

Main Heat Pump Units IK40/IK41

rcworld 7.3
rcgroup products selection software
rcgroupairconditioning
ISO 9001 • Cert. 18/6
rcgroup.it

EAGLE HP T.240.P4.Y2 D

EUROVENT CERTIFIED PERFORMANCE

SUMMER WORKING MODE		
COOLING CAPACITY	**kW**	**226**
Compressors power input	kW	73,4
Compressors electric absorption	A	130
Water temperature (in/out)	**°C**	**12/7**
Water flow rate	m³/h	43
Pressure drop	kPa	57
Ambient temperature	**°C**	**35**
WINTER WORKING MODE		
HEATING CAPACITY	**kW**	**276**
Compressors power input	kW	58,2
Compressors electric absorption	A	113
Water temperature (in/out)	**°C**	**23,8/30**
Ambient temperature	**°C**	**2,5**
COMPRESSORS		scroll
Quantity	n.	4
Max electric absorption FLA	A	194
Starting current LRA	A	418
Capacity steps	n.	4
Velocità di rotazione	cps	
PLANT WATER	n.	1
Water volume	l	20,3
Max water flow rate	m³/h	55,6
Antifreezing	%	Eth. Glycol 30%
Fouling factor	m² °C/kW	0,043
AXIAL FANS	n.	6
Airflow	m³/h	132000
Power input	kW	11,64
Max electric absorption FLA	A	23,4
REFRIGERANT		R410A
Total refrigerant charge	kg	67,5
Gas circuits	n.	2
POWER SUPPLY	V-ph-Hz	400/3/50
ENERGY EFFICIENCY INDEXES		
EER = Energy Efficiency Ratio	kW/kW	2,66
COP = Coefficient of Performances	kW/kW	3,95
ESEER = Eurovent Standard		4,02
IPLV = ARI Standard 550/590		4,46
SPL @ 1 m in free field conditions (ISO3744)	dB(A)	82,2
Sound power level ISO EN 9614-2	dB(A)	102
DIMENSIONS		
Length	mm	4056
Width	mm	2310
Height	mm	2490
NET WEIGTH	kg	2850
ACCESSORIES		
175 - Victaulic connections;		

Data Room Coolers

RCWORLD 6.4

rcgroup.it

neXt DW.U.S* 015.Z1.H2

ACCESSORIES

EXTRA-CIRCUIT		
Total cooling cap.	kW	9,05
Sensible cooling cap.	kW	9,05
Water temperature (in/out)	°C	10/15
Inlet air temperature	°C	22
Relative humidity	%	50
Water flow rate	m^3/h	1,56
Coil + valve pressure drop	kPa	7
Antifreezing	%	0
NET WEIGTH	kg	271

Cooling Machine IK42

RCWORLD 6.3

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MANTA T.48.P2.D.J7

KJØLEKAPASITET	**kW**	**55,7**
Kompressorer effekt	kW	15,5
Kompressorer strømforbruk	A	29,2
FORDAMPER		
Vanntemperatur (inn/ut)	**°C**	**15/10**
Vannmengde	m³/h	9,57
Trykkfall	kPa	29
CONDENSER		
Vanntemperatur (inn/ut)	**°C**	**37/43**
Vannmengde	m³/h	11,4
Trykkfall	kPa	37
KOMPRESSORER		scroll
Antall	n.	2
Maks. strømforbruk FLA	A	44
Startstrøm LRA	A	140
Capacity steps	n.	2
FORDAMPER	n.	1
Vannvolum	l	4,3
Maks. vannmengde	m³/h	12,8
Frostsikring	%	0
Tilsmussingsfaktor	m² °C/kW	0,043
CONDENSER	n.	1
Vannvolum	l	4,3
Maks. vannmengde	m³/h	15,9
Frostsikring	%	Eth. Glycol 30%
Tilsmussingsfaktor	m² °C/kW	0,043
KJØLEMEDIE		R410A
Total refrigerant charge	kg	4,8
Gas circuits	n.	2
NETTSPENNING	V-ph-Hz	400/3/50
ENERGY EFFICIENCY INDEXES		
EER = Energy Efficiency Ratio	kW/kW	3,59
ESEER = Eurovent Standard		6,47
IPLV = ARI Standard 550/590		6,88
SPL @ 1 m i fritt felt forhold (ISO3744)	dB(A)	53
Potenza sonora Lw secondo ISO EN 9614-2	dB(A)	68,9
DIMENSJONER		
Lengde	mm	1200
Bredde	mm	750
Høyde	mm	1700
NETTO VEKT	kg	455

TILBEHØR

172 - Gummi fester (sett); 919 - Klokkekort; 923 - RC-Com MBUS/JBUS Seriekort; CO2 avgift; Potfrie signaler;

NTNU

Norwegian University of
Science and Technology